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February 10, 1995
CON/ERW/95-027

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Contracting Officer

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Office of Acquisition and Grants Services, RAD-30
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Subj: Draft Final Report – "NUCARS Modeling in Support of Freight Car Tolerance Side Bearing Clearance Test"

Refr: Contract DTFR53-93-C-00001, Task Order No. 101, Modification No. 2

Dear Mr. Spratling:

Here is the subject report for FRA review and comment. It is AAR's intent to furnish the *final* version within ten (10) days from receipt of FRA's comments. Please fax any changes to me on 719.584.0711 and we'll make a *best efforts* attempt to meet our contractual obligation submission date of February 28, 1995 for the *final* version.

Sincerely,

ASSOCIATION OF AMERICAN RAILROADS

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U.S. Department of Transportation

Federal Railroad Administration

# NUCARS MODELING IN SUPPORT OF FREIGHT CAR TOLERANCE SIDE BEARING CLEARANCE TEST

Office of Research and Development
Washington D.C. 20590


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**Pueblo, Co 81001**



DOT/FRA/ORD-

February 1995

[redacted]

**DRAFT**

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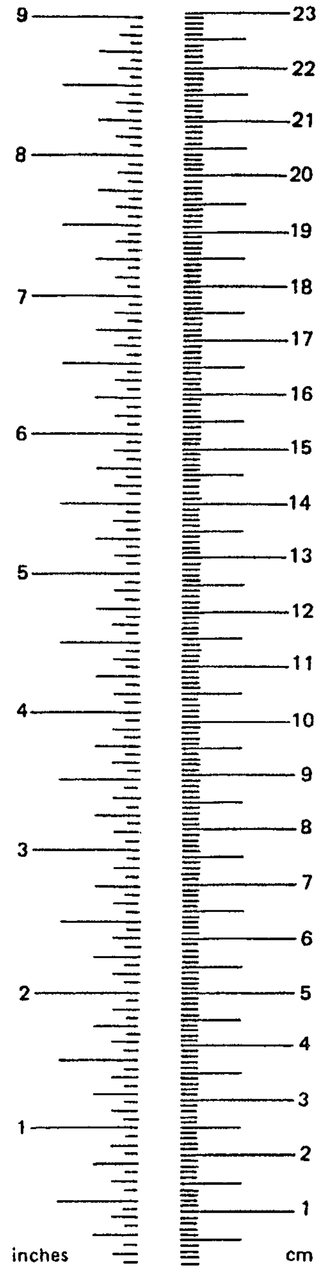
# METRIC CONVERSION FACTORS

## Approximate Conversions to Metric Measures

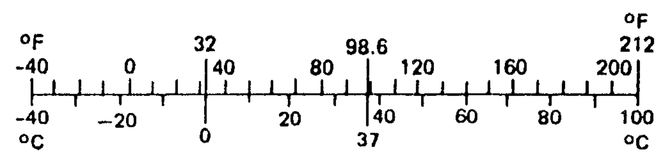
| Symbol | When You Know | Multiply by | To Find | Symbol |
|---|---|---|---|---|
| | | **LENGTH** | | |
| in | inches | *2.50 | centimeters | cm |
| ft | feet | 30.00 | centimeters | cm |
| yd | yards | 0.90 | meters | m |
| mi | miles | 1.60 | kilometers | km |
| | | **AREA** | | |
| $in^2$ | square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

* 1 in. = 2.54 cm (exactly)




## Approximate Conversions from Metric Measures

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



## Table of Contents

## 1.0 INTRODUCTION

The Federal Railroad Association (FRA) sponsered a research project with the Association of American Railroads' (AAR) managed Transportation Technology Center (TTC), Pueblo, Colorado, to investigate the effects of variations in side bearing clearance on the vechicle dynamics of a loaded freight car with a relatively high center of gravity. AAR's computer program NUCARS was used to simulate the car's dynamic behavior.

This investigation was a continuation of another project (TO101,1993), which also tested a loaded freight car with a high center of gravity (c.g.), but on the Vibration Test Unit in TTC's Rail Dynamics Laboratory and on track.[1] These early comparisons, between on-track tests and limited computer simulations, found in the analysis performed by Volpe National Transportation Systems Center (VNTSC) provided a basis for the studies included in this report.[2]

## 2.0 OBJECTIVE

The objective of this investigation was to support previous (1993) on-track testing and analytical modeling to further determine the effect of side bearing clearance in the presence of various secondary suspension configurations on high c.g. cars. Further parameters investigated were track gage, wheel/rail coefficient of friction, secondary spring stiffness, and the effect of hydraulic damping. All vehicle configurations were simulated using various track input amplitudes.

Prior to modeling, a preliminary review of all test data was conducted to obtain a better understanding of the overall test results and to help in the direction of the modeling effort.

## 3.0 PROCEDURE

The modeling effort involved various model configurations including side bearing clearance, wheel/rail coefficients of friction, car suspension, track input amplitude and track gage. The configurations modeled were based on comments from FRA and discussions with VNTSC personnel.[3] Review of the test plan resulted in modifications of the test matrix. These changes were made to better suit the overall objectives of the test program. All configurations were modeled on mathematically generated 39-foot staggered joints using standard wheel/rail profiles on a high c.g. covered hopper car.

### 3.1 NUCARS DESCRIPTION

NUCARS is a generalized vehicle dynamics program used to aid in vehicle design and testing. Vehicles are simulated in NUCARS as an assembly of body masses linked by connections representing actual suspension components. The flexible structure of the model allows the user to simulate the response of any vehicle to any track input. Simulation output includes predicted body and wheel set displacements, suspension forces and strokes, wheel/rail interaction forces, and wheel and rail wear indices.

The required input to accurately describe a specific vehicle includes masses and dimensions of major body components, moments of inertia, and detailed information on all suspension locations, stiffness, damping characteristics and the c.g. height.

The model simulated the NUCARS mode of predicting forces and car body response resulting from the dynamics of the rolling wheel on rail.

## 3.2 MODEL DEVELOPMENT

Development of the initial NUCARS model for this application was performed by personnel from VNTSC. This information was used as a baseline for the TTC modeling effort to eliminate duplication of effort.

The following changes were made to the NUCARS system file provided by VNTSC for the D5 spring configuration. All changes were agreed to by VNTSC personnel and TTC engineers.

### 3.2.1 System File

- .003 inch was added to the desired side bearing clearance due to the center plate spring stiffness (ie. .250 to .253), which then allowed proper settling of the car body center bowl.
- The side frame wedge angle on the Barber truck was changed from 37.5 degrees to 35 degrees, appropriate for this truck design.
- The vertical spring rate for the bolster spring nest was changed to a standard 100-ton spring rate for the D-5 simulations.

## 3.3 MODELING MATRIX

Table 1 summarizes the test matrix configurations used for the NUCARS modeling.

**Table 1. Modeling Configurations**

| Parameter | Configuration |
|---|---|
| Classification | Covered Hopper |
| Loaded Car Body Weight | 266,000 lbs |
| C.G Height | 93.0 inches |
| Side Bearings | Roller 688B |
| Truck Type | Barber, Variable Damped S-2-C |
| Truck Spacing | 40.5′ |
| Side Bearing Clearance | 1/4" and 3/8" |
| Truck Suspension | D5, D5 w/Hydraulic Damping, and D7 |
| Speed | 10 to 20 and 20 to 10 mph<br>10 to 30 and 30 to 10 mph |
| Track Gage | 56, 56.5, and 57 inches |
| Wheel Profile | Standard AAR-1B |
| Rail Profile | AREA136 |
| Input Amplitudes | 0.75, 1.0 and 1.25 inches |
| Wheel/Rail Coefficient of Friction | 0.1, 0.3 and 0.5 |

The basis for this model is the AAR102 covered hopper car. This car has a history of testing dating back to when it was part of the Norfolk Southern Car Rocker Test Facility for testing trucks and snubbing devices. AAR102 is used by AAR/TTC for the conduct of specification testing. The car is currently loaded with packed cement powder.

## 3.4 NUCARS INPUT FILE PARAMETERS

### 3.4.1 Input File

Track inputs based on a rectified sine waveform, shown in Figure 1, were used during all of the modeling cases. Three input amplitudes (0.75, 1.0 and 1.25) were selected based on the test data review and consultation with VNTSC personnel. The length of the input was sufficient to clearly identify resonant speeds, generally 1250 to 2500 feet. The distance-based rate of the input was set at 0.0085 mph/foot of distance traveled. This is equivalent to a 10 mph change in velocity over 1170 feet.

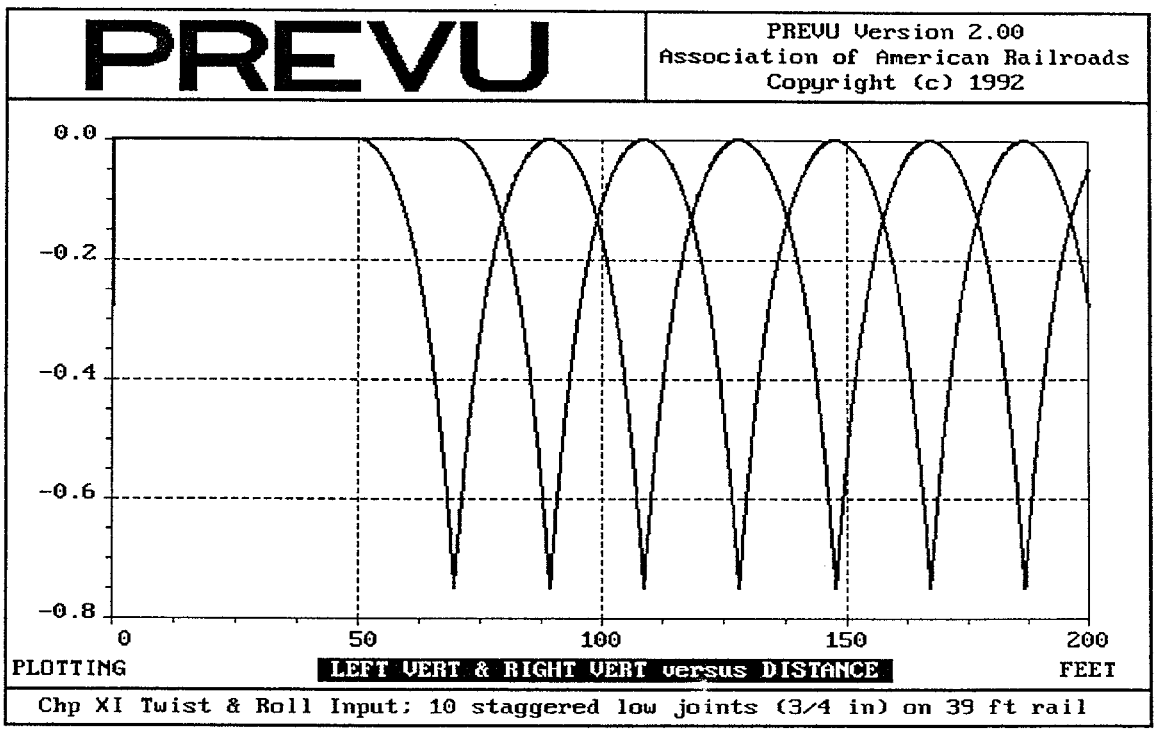



Figure 1. NUCARS Input File -- 3/4"

### 3.4.2 Wheel/Rail Geometry File

A description of the contact between the wheels and rails is required to run a railroad track simulation using NUCARS. This is contained in the wheel/rail geometry (WRG) file. The WRG file used for all modeling cases consisted of a standard AAR 1B wheel profile on an AREA 136-pound rail. Track gage variation in the WRG file included standard, narrow, and wide gage configurations. Figure 2 shows a NUCARS plot of rolling radius change with wheel set lateral position for this combination of wheel and rail at standard gage. The 1:20 tapered tread region is apparent during tread contact. The transition up the flange is fairly continuous until two point contact occurs.

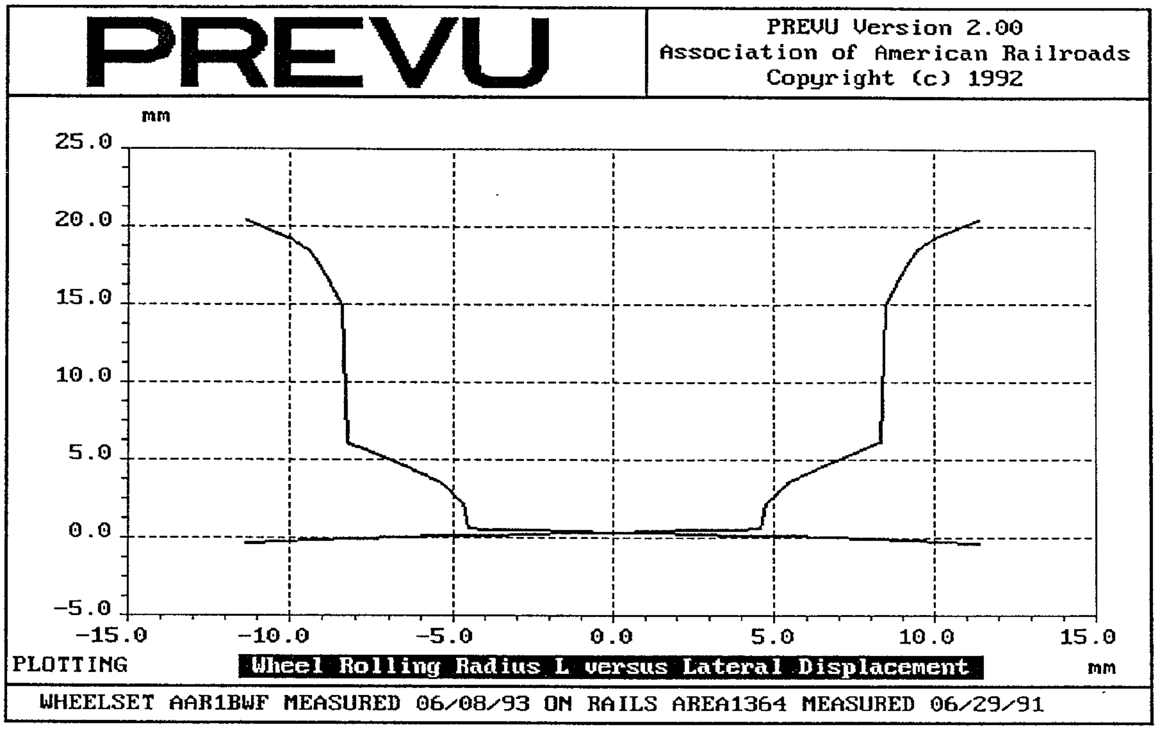



Figure 2. **Wheel Rolling Radius versus Lateral Displacement**

### 3.4.3 Data File

Table 2 lists the primary measurements which were evaluated using the NUCARS model. Additional measurements were also collected for diagnostic purposes during preliminary analysis. All data channels collected during simulations were high-pass filtered at 15 Hz.

Table 2. VTU Test Measurements Predicted by NUCARS

| Description | Location |
|---|---|
| Car Body Roll Rotation | Center |
| Vehicle Velocity | n/a |
| Vertical Wheel Force | Axle 1,2,3 & 4 |
| Axle Roll Rotation | Axle 1,2,3 & 4 |
| Bolster Roll Rotation | A & B End |

## 4.0 RESULTS

Simulations were performed based upon the model and input criteria described in Section 3.0 of this report. The primary mode of data reduction was conducted by combining the peak-to-peak roll angles with speed for each major run configuration.

Figures 3 and 4 show typical NUCARS plots of car body roll; the peak-to-peak roll angle of 5.5 degrees occurred at 12 mph. Figure 5 shows a typical history plot of speed versus distance during a downward sweep from 20 to 10 mph.

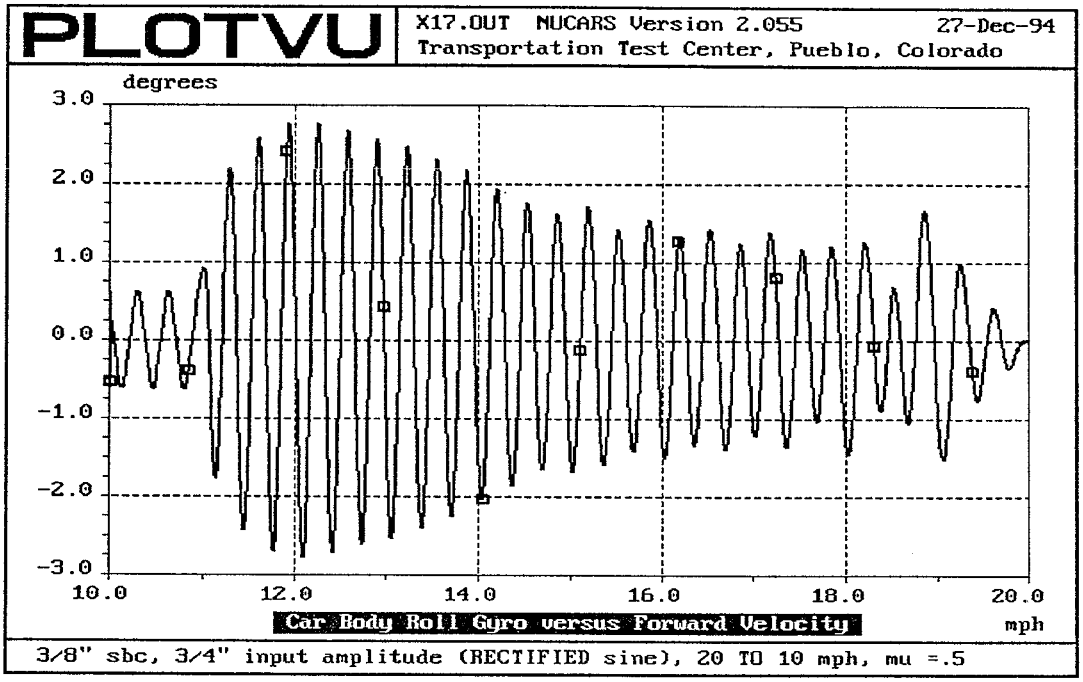



Figure 3. Time History -- Car Body Roll Versus Speed

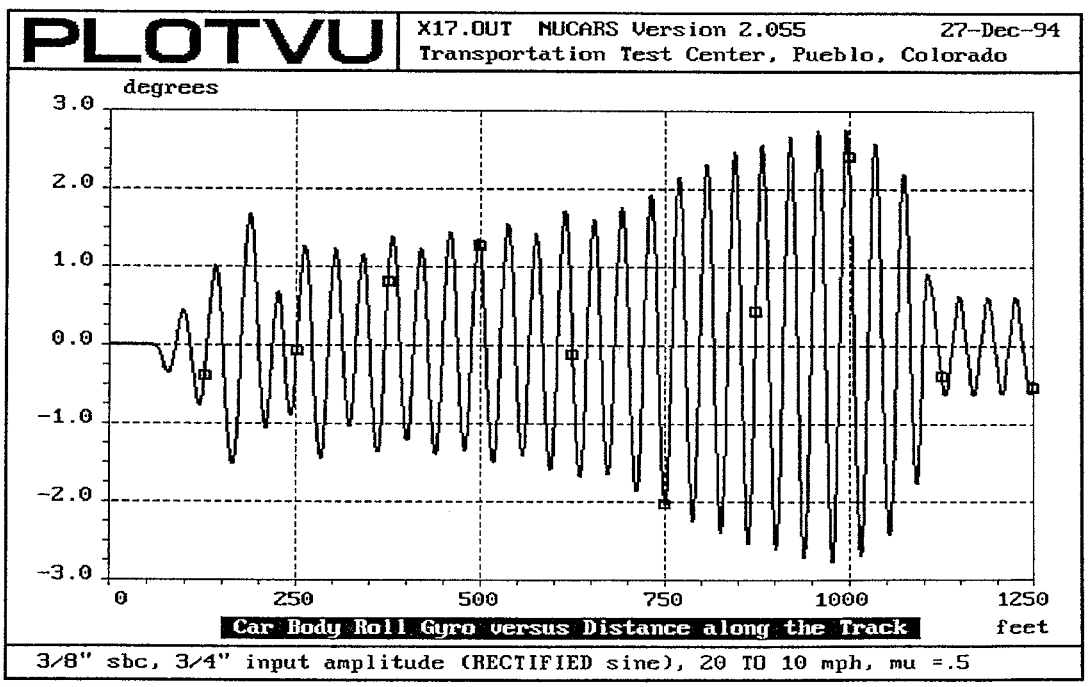



Figure 4. Time History -- Car Body Roll Versus Distance

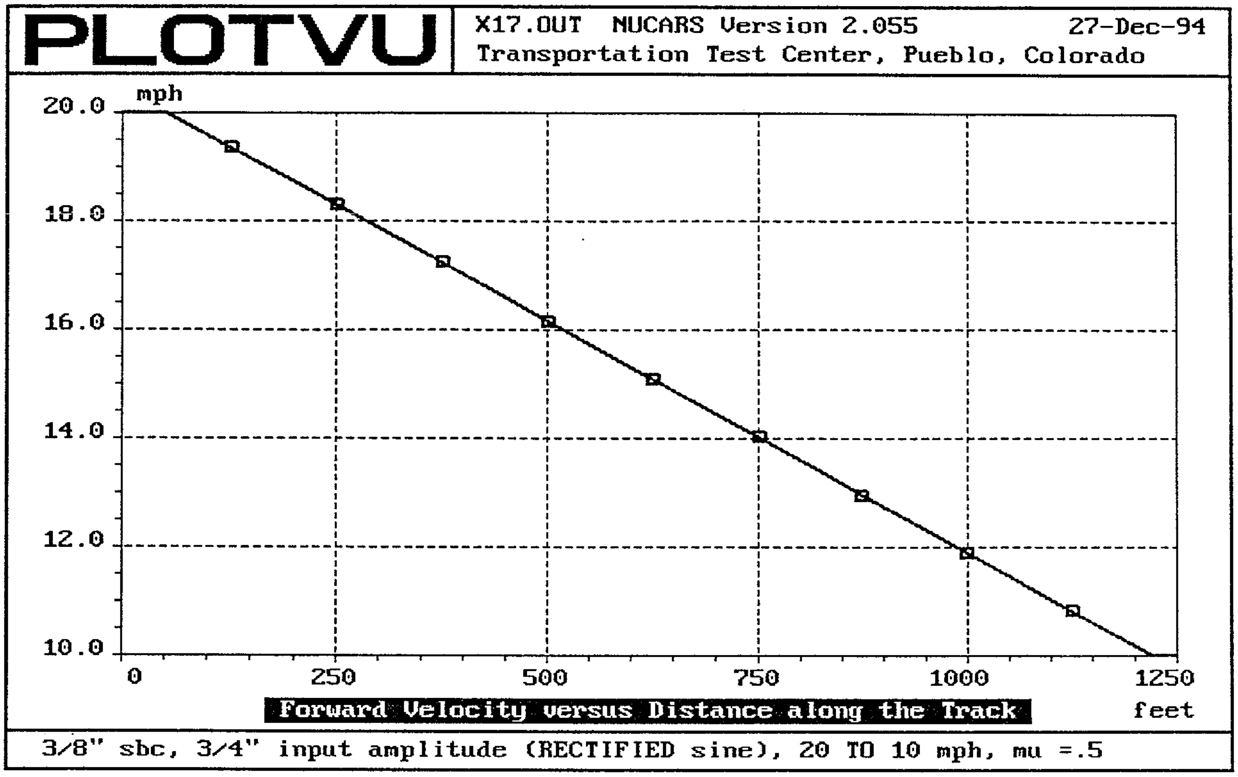



Figure 5. Time History -- Speed Sweep Versus Distance

## 4.1 MODELING ANALYSIS

Two performance measures were selected to analyze roll response from this modeling effort. The first is the maximum peak-to-peak roll angle. AAR freight certification standards place a limit of 6 degrees on this value. The second criterion is the minimum vertical wheel load expressed as a percentage of the static wheel load. AAR standards state that this value should not fall below 10 percent at any speed up to the maximum operating speed.[4] Values below this threshold identify potential loss of wheel/rail contact and guidance. The following sections contain summary plots from reduced data showing peak-to-peak car body roll versus speed of each model configuration.

### 4.1.1 D5 Configuration

Figures 6 through 8 show plots from the D5 truck spring configuration. Each figure includes side bearing clearances of 1/4- and 3/8-inch side bearing clearance. The figures are presented in order from 0.75, 1.00 and 1.25 track input amplitudes respectively.

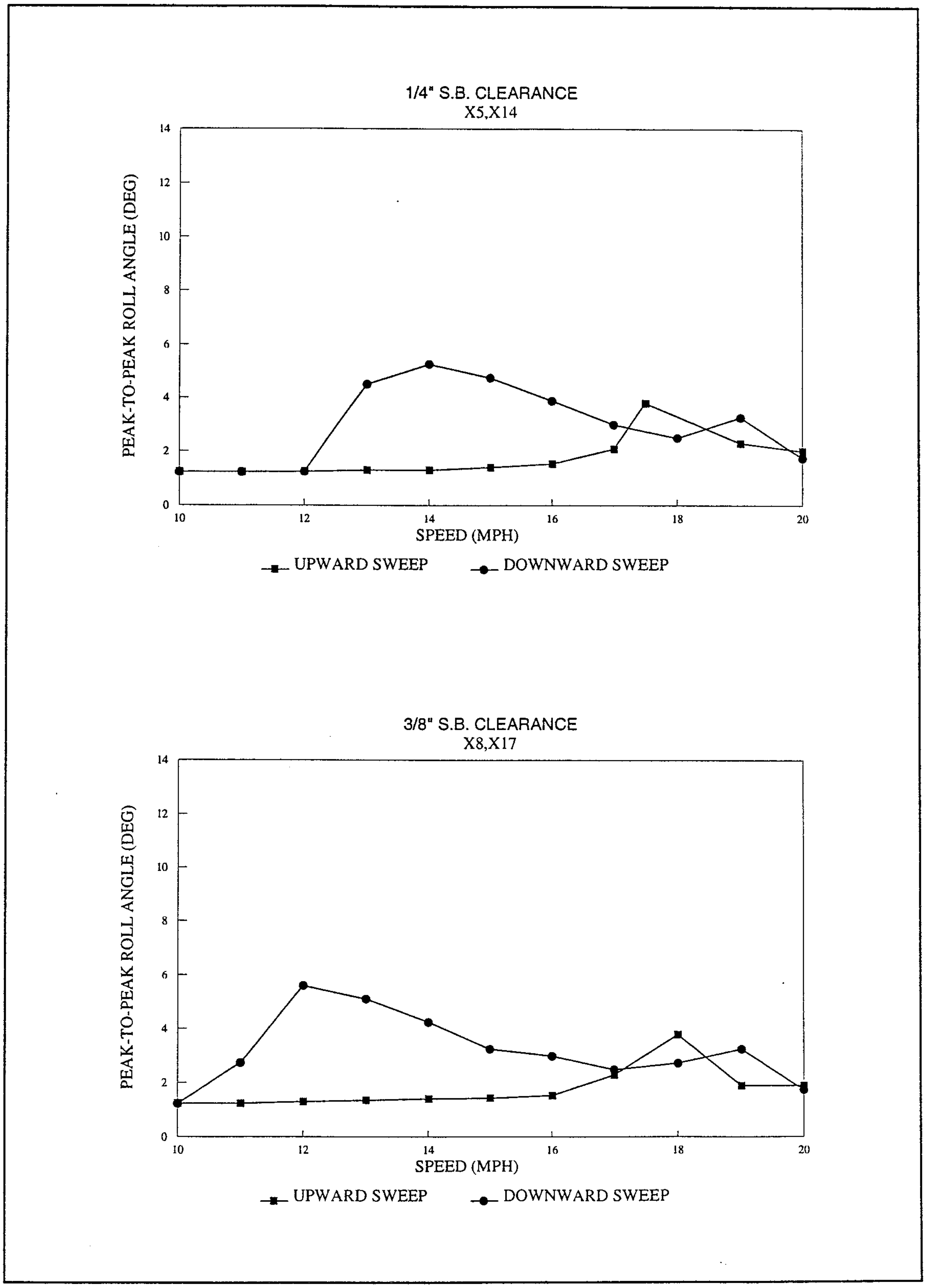
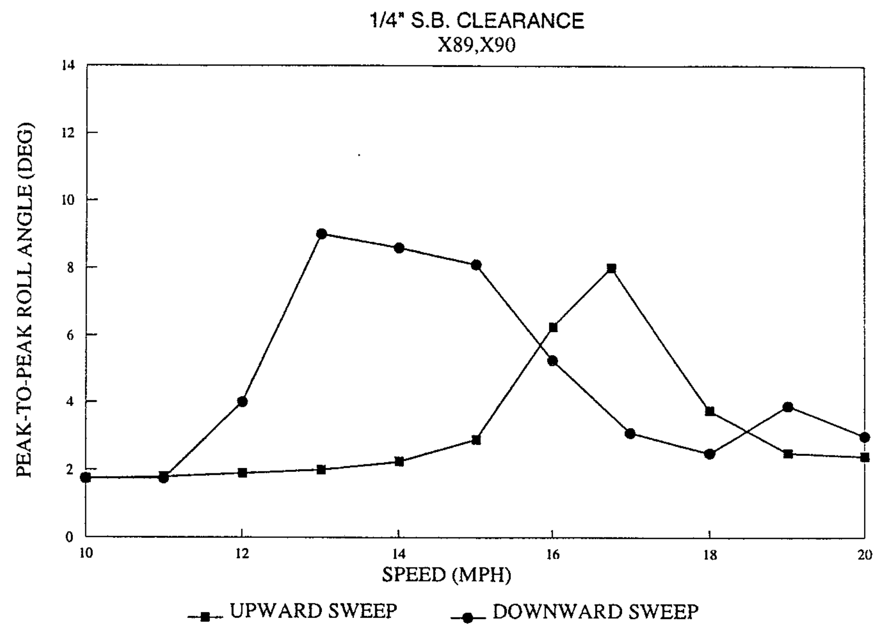



**Figure 6. Car Body Roll Versus Speed 3/4" Input**



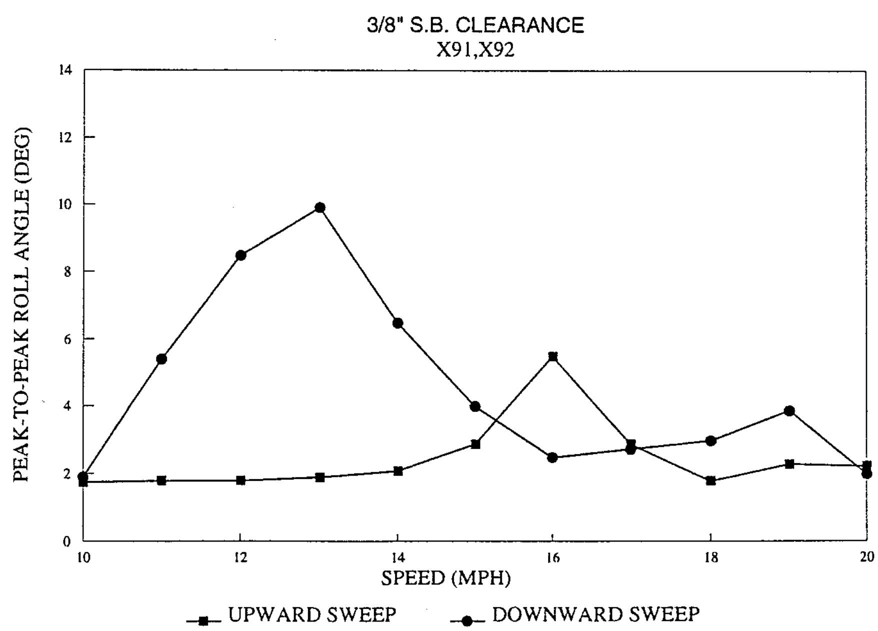
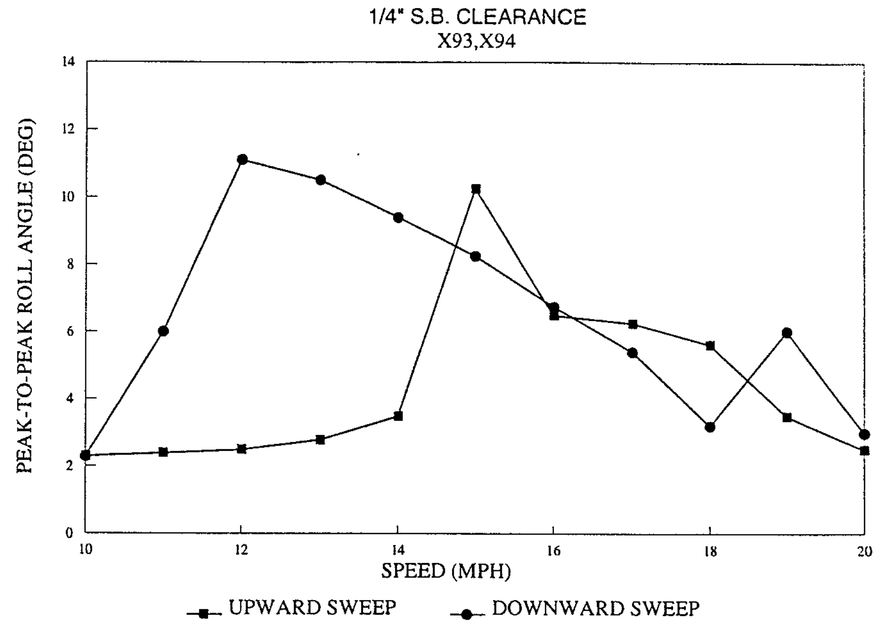




**Figure 7. Car Body Roll Versus Speed 1.0" Input**



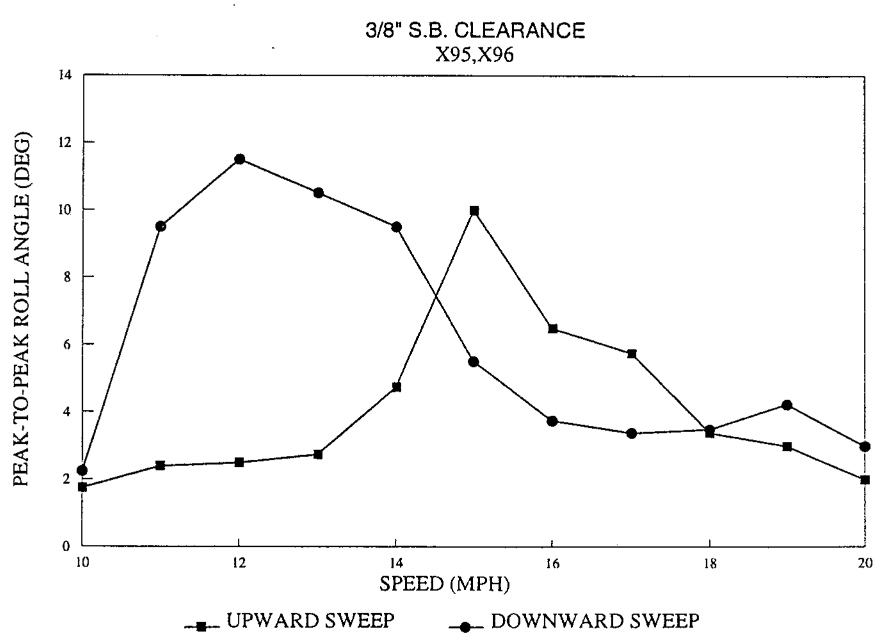




**Figure 8. Car Body Roll Versus Speed 1.25" Input**

### 4.1.2 D5 Configuration With Suspension Damping

Figures 9 through 11 contain plots from the D5 with hydraulic damping truck spring configuration. Each figure includes side bearing clearances of 1/4- and 3/8-inch side bearing clearance. The figures are presented in order from 0.75, 1.00 and 1.25 track input amplitudes respectively.

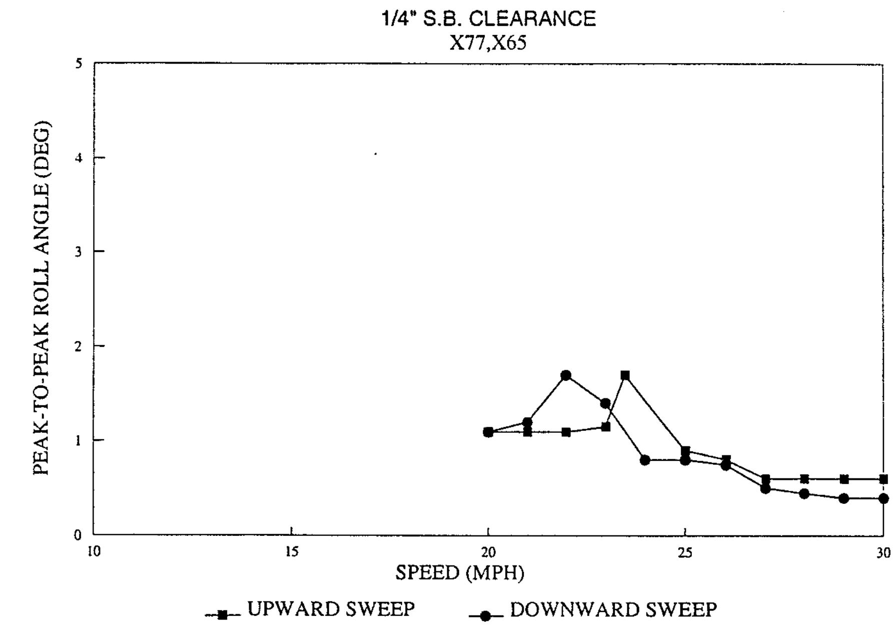
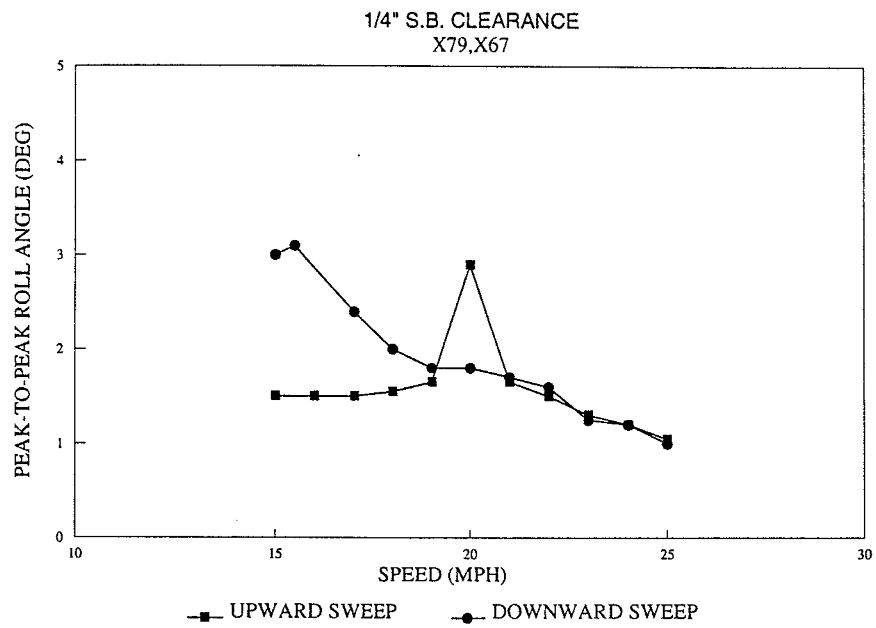



**Figure 9. Car Body Roll Versus Speed 3/4" Input**



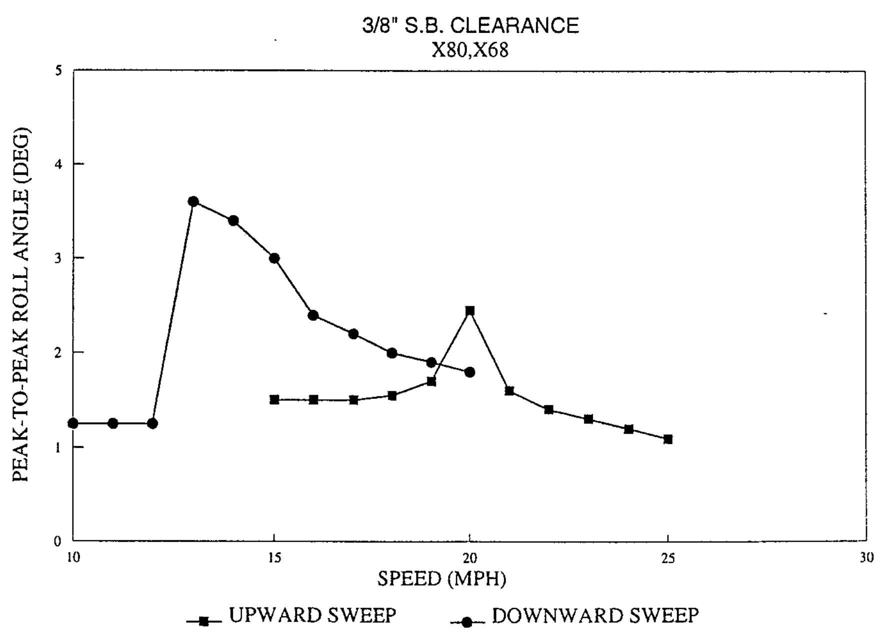




**Figure 10. Car Body Roll Versus Speed 1.0" Input**

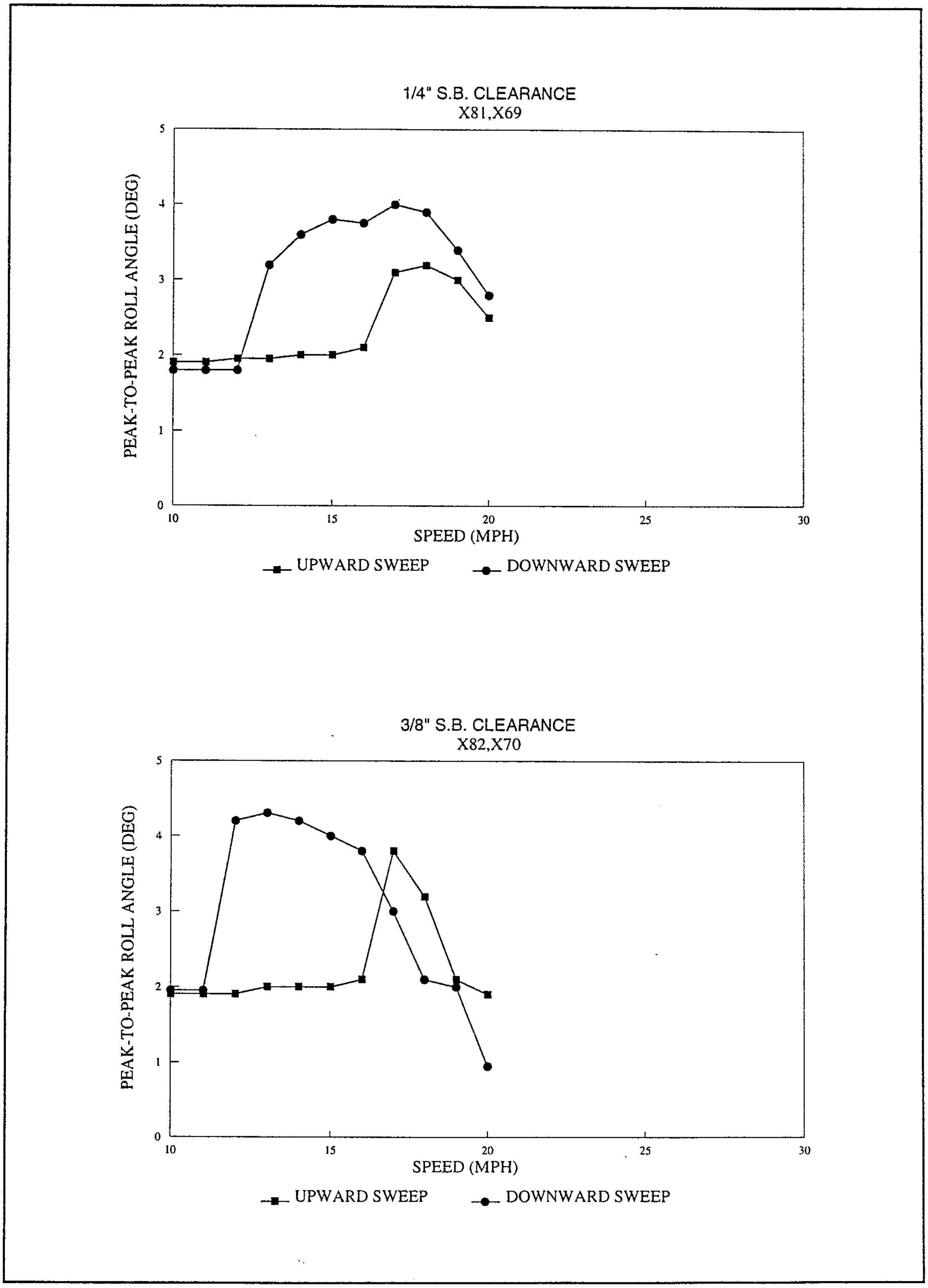



**Figure 11. Car Body Roll Versus Speed 1.25" Input**

### 4.1.3 D7 Configuration (Long Travel Suspension)

Figures 12 through 14 contain plots from the D7 truck spring configuration. Each figure includes side bearing clearances of 1/4- and 3/8-inch side bearing clearance. The figures are presented in order from 0.75, 1.00 and 1.25 track input amplitudes respectively.

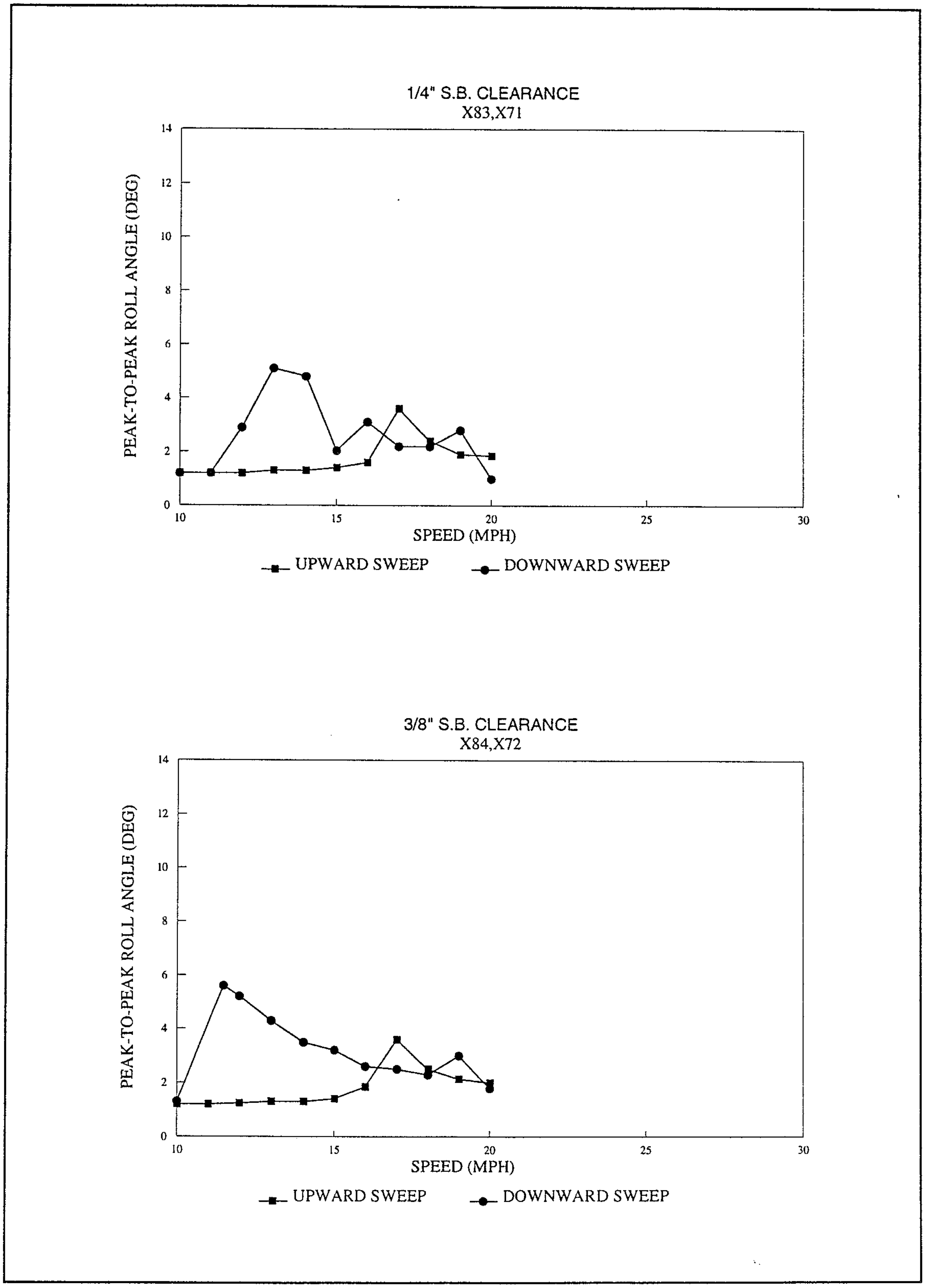



Figure 12. Car Body Roll Versus Speed 3/4" Input

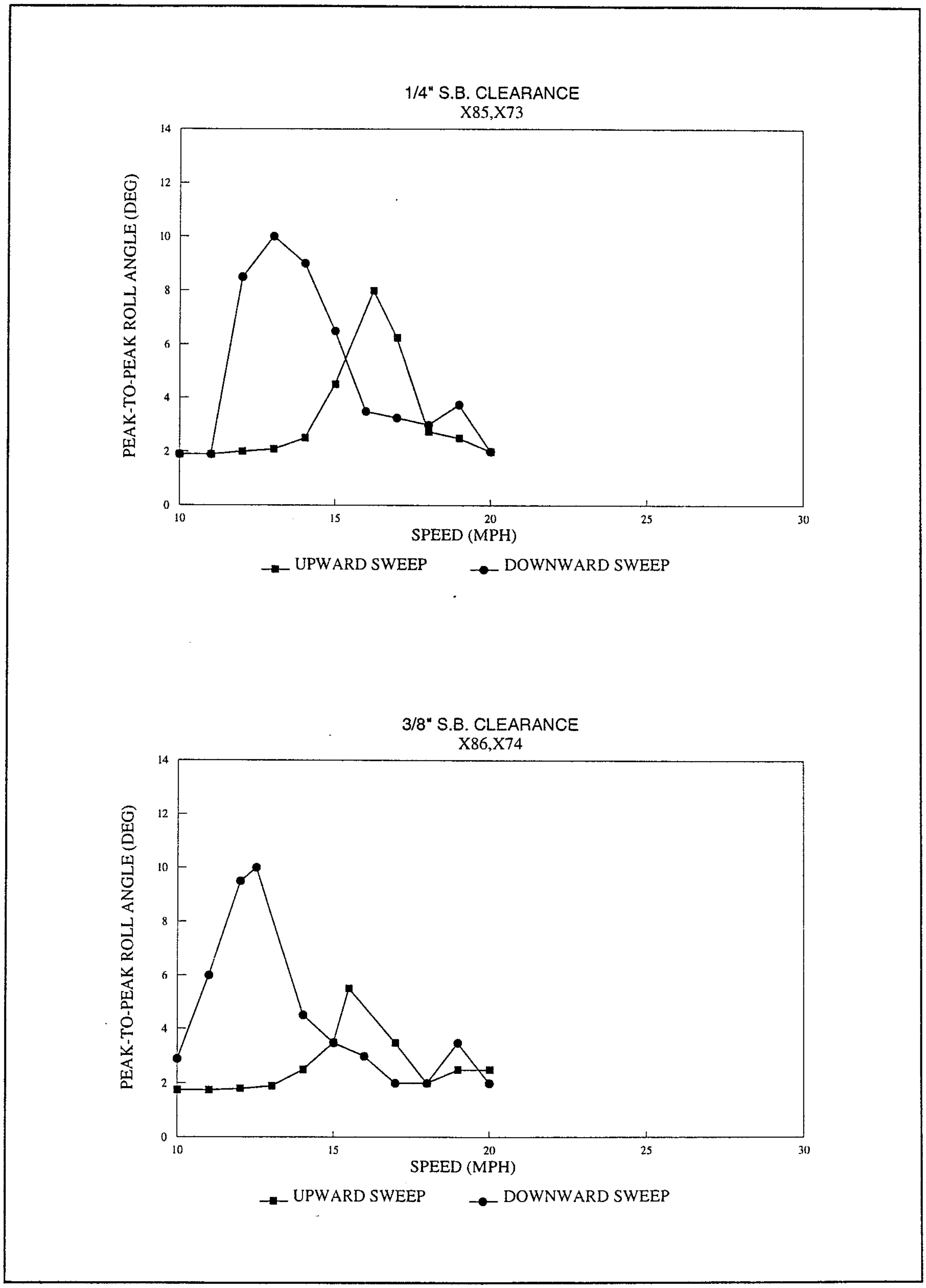



**Figure 13. Car Body Roll Versus Speed 1.0" Input**

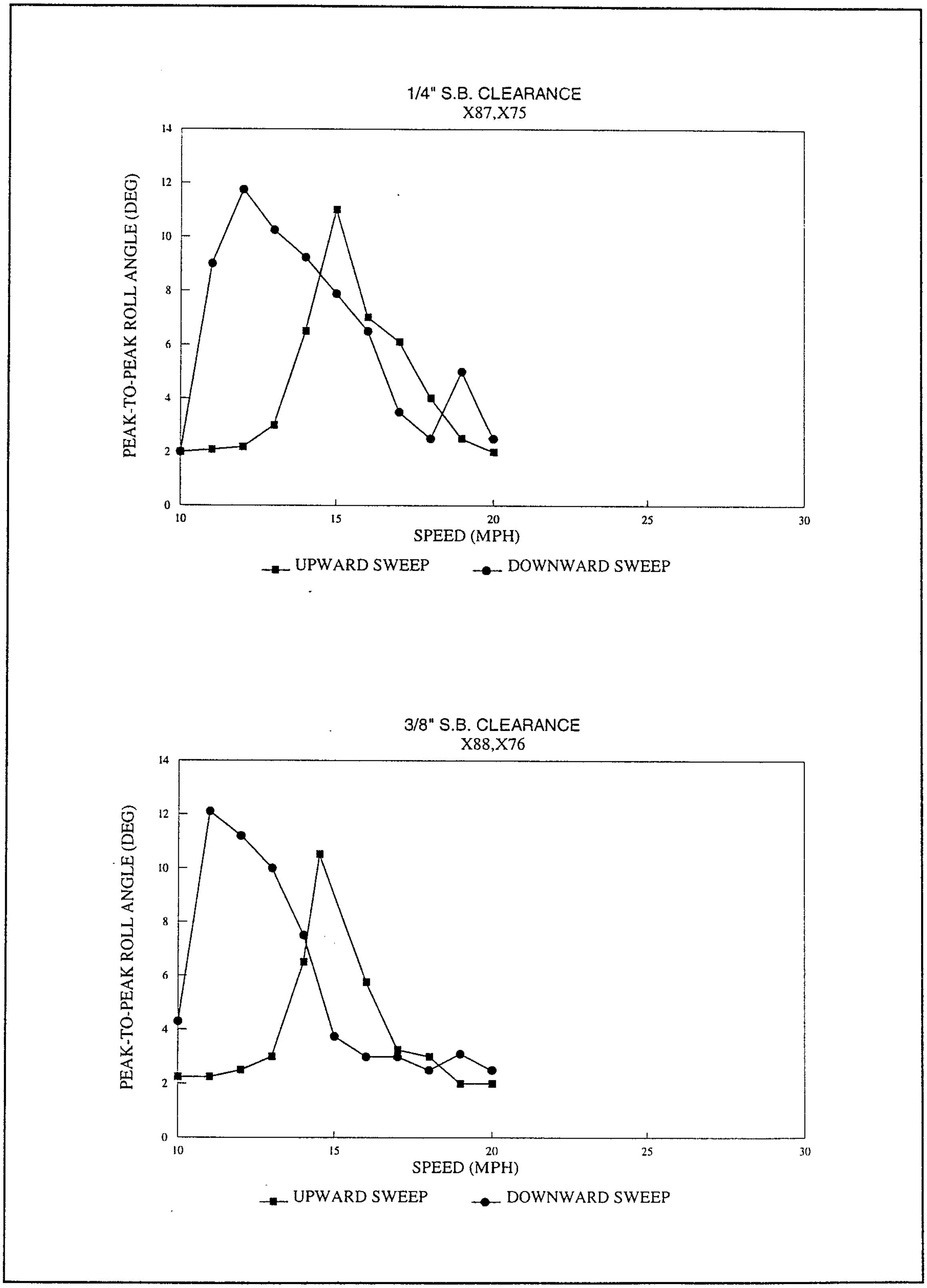



**Figure 14. Car Body Roll Versus Speed 1.25" Input**

## 4.2 VARIATION IN COEFFICIENT OF FRICTION

Figures 15 through 17 show history plots of car body roll versus speed for each variation of the wheel/rail coefficient of friction (0.1, 0.3, and 0.5.). Results from these plots show that no significant differences in car body roll were detected when the wheel/rail coefficient of friction was changed. This analysis, conducted at the beginning of the modeling runs, concluded that a wheel/rail coefficient of friction of **0.5** would be used for the remainder of the modeling runs.

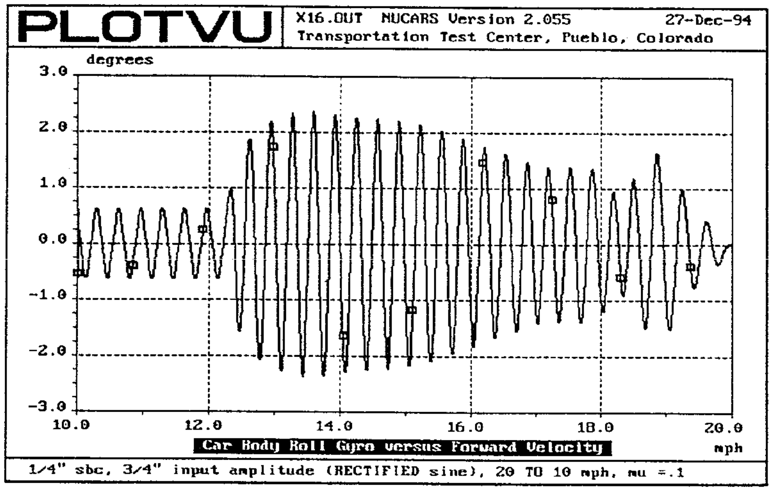



Figure 15. Time History -- Car Body Roll Versus Distance COF = .1

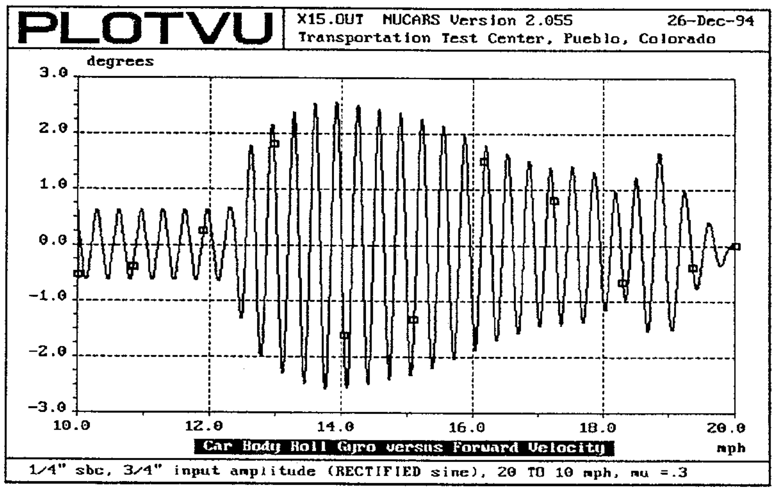



Figure 16. Time History -- Car Body Roll Versus Distance COF = .3

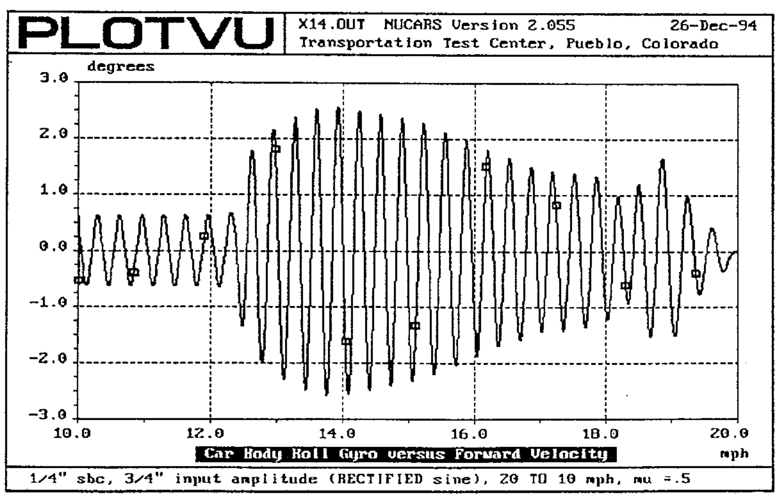



Figure 17. Time History -- Car Body Roll Versus Distance COF = .5

## 4.3 VARIATION OF TRACK GAGE

Figures 18 and 19 show x-y plots of peak-to-peak car body roll versus speed during the two test modes (sweep-up and sweep-down). Configurations of narrow (56 inch), standard (56.5 inch), and wide (57 inch) track gage are included in each plot. Both test modes consistently defined that the greatest peak-to-peak car body roll occurred when a narrow track gage of 56-inches was used. Since the change in the peak-to-peak roll angle was typically less than 1 degree, the remainder of the modeling runs were conducted using an standard track gage (56.5 inches).

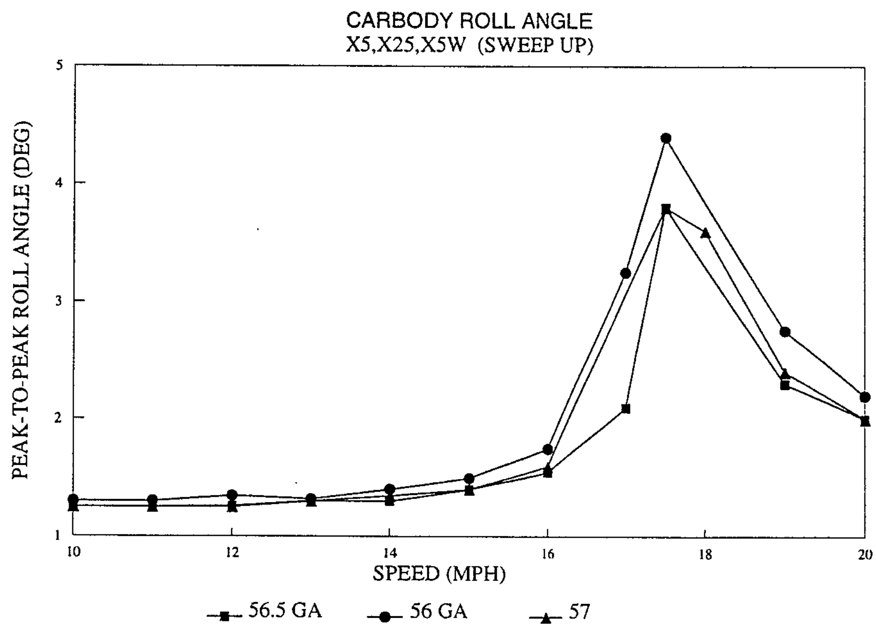



Figure 18. Variation of Track Gage -- Sweep Up

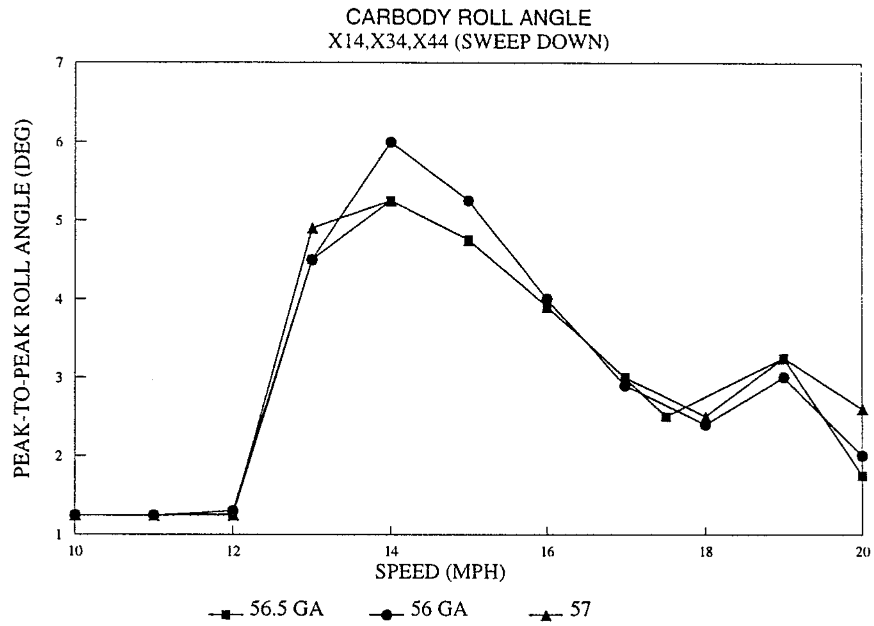



**Figure 19. Variation of Track Gage -- Sweep Down**

## 4.4 COMPARISON TO CHAPTER XI LIMITS

Figures 20 through 22 show plots of compiled wheel loads and peak-to-peak car body roll angles for each of the three input amplitudes, 0.75 inch, 1.0 inch and 1.25 inch. Each plot shows a comparison between the values obtained from selected simulations and the allowable AAR specification limits for car body roll and minimum vertical wheel load.

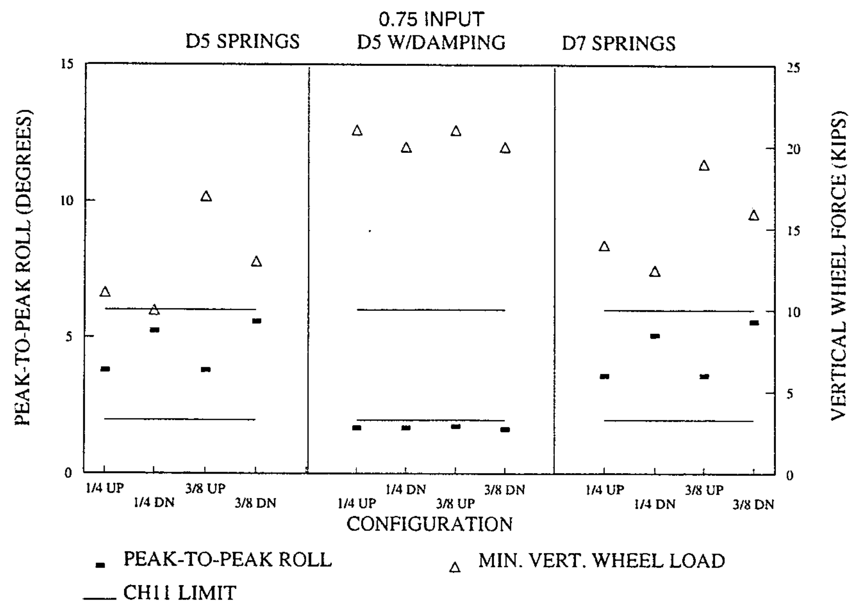



**Figure 20. Wheel Loads and Car Body Roll Plot -- 3/4" Input**

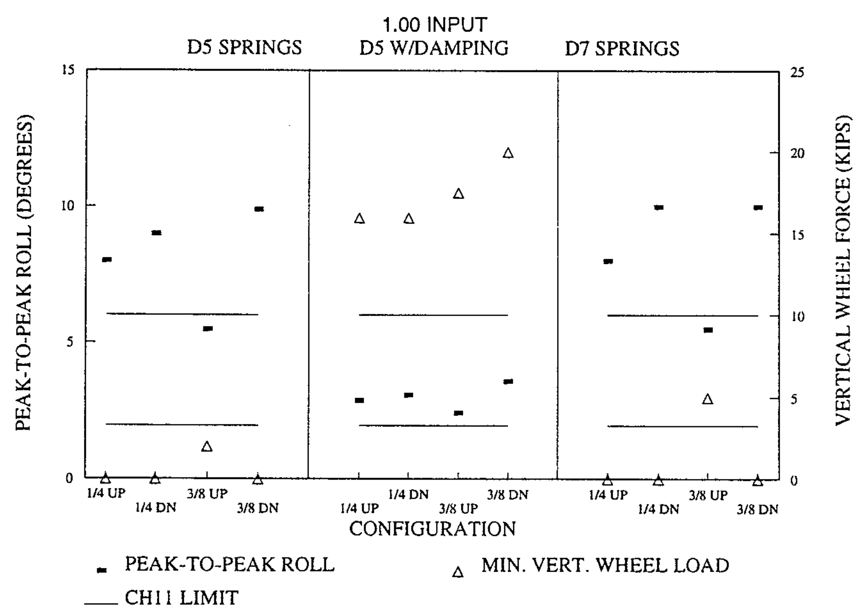



**Figure 21. Wheel Loads and Car Body Roll Plot -- 1.00" Input**

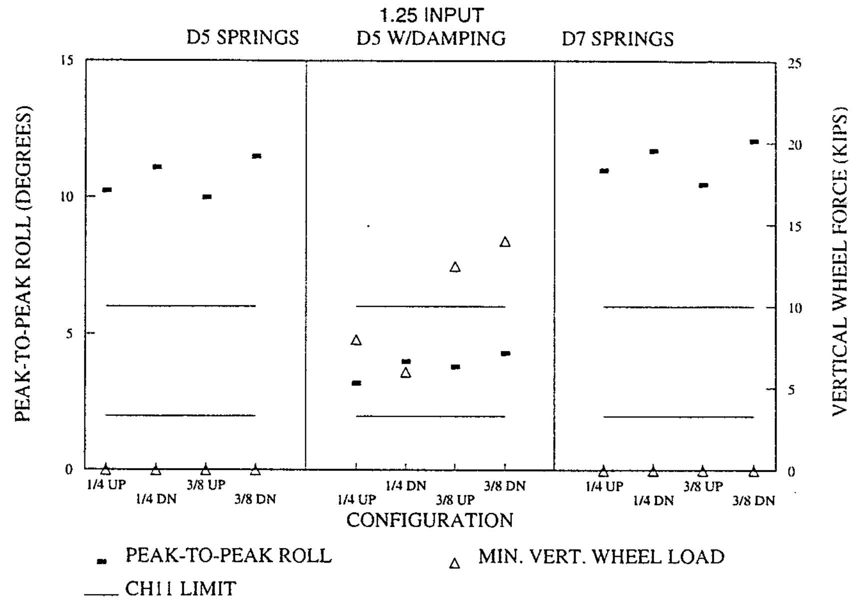



**Figure 22. Wheel Loads and Car Body Roll Plot -- 1.25" Input**

## 5.0 CONCLUSIONS

The following observations are based on the modeling configurations included only in this report. These observations may not, in some cases, directly apply to the previous test data where a much wider spectrum of testing was analyzed. Typical examples include prior test data from configurations of 1/8 inch, 1/2 inch and 3/4 inch side bearing clearance, compared to observations of 1/4 inch and 3/8 inch included in this report.

### 5.1 GENERAL OBSERVATIONS - CAR BODY ROLL

- Downward sweep runs always produce a lower frequency peak roll angle when compared to the upward sweep. Typically, it is of greater amplitude than the upward sweep.
- Vehicle roll angle increases with input amplitude; the largest, increased between 0.75- and 1.0-inch input amplitude.
- The variation of side bearing clearance modeled, 1/4 inch and 3/8 inch, had little influence on roll angle. Previous test results showed that larger side bearing clearance gap changes did affect car body roll angles.[1]
- Spring type (D5 vs D7) has a very small effect on peak-to-peak roll angle.

- Hydraulic damping significantly reduces the peak-to-peak roll angle.
- Wheel/rail coefficient of friction and track gage have a negligible impact on roll angle.
- 0.75-inch input produced no roll angle in excess of the AAR 6-degree limit.
- For the 1.00-inch input amplitude, D5 and D7 undamped configurations, 6 of 8 simulations produced peak-to-peak roll angles above the 6-degree specification limit.
- At 1.25-inch input amplitude, all configurations without additional damping exceeded the 6-degree limit. All cases produced peak-to-peak roll angles between 10 and 12 degrees.

### 5.2 GENERAL OBSERVATIONS -- WHEEL LOADS

- For 0.75-inch input amplitude, the minimum wheel load is never less than 30 percent of static wheel load regardless of spring/damper arrangement. For the D5 damped arrangement it is never less than 60 percent of the static load. It is anticipated that very worn trucks with lower damping capability as tested previously, could have produced wheel unloading of greater significance if modeled.
- With 1.0-inch input amplitude, half of the cases produced wheel lift although none of these were with hydraulic damping.
- At 1.25-inch input amplitude, all D5 and D7 configurations without hydraulic damping generated wheel lift.
- With 1.25-inch input amplitude, D5 configuration with hydraulic damping maintained at a minimum of 18 percent of static wheel load.

## 6.0 RECOMMENDATIONS

This study covered only a small range of possibilities in terms of harmonic roll response for high c.g. vehicles. The repeated jump response detected with the upward and downward sweeps illustrate the difficulty in identifying a safe, low operating speed.

The different side bearing clearances modeled, 1/4 inch and 3/8 inch, provide little or no additional margin of safety. Hydraulic damping at its optimal level can greatly reduce the risk of wheel lift and therefore derailments. The durability of hydraulic dampers may still be questionable in addition to these benefits given purchase, installation, and maintenance expense.

If the roll performance is a proven safety concern, then additional investigation of roll moderation techniques should be performed. This should also include a survey of the track geometry for amplitude, wavelength, and repeat distance.

A vehicle survey with regard to the population of vehicles equipped with vertical damping would also be of value. An estimate of high c.g. limits typically produced should be tabulated. Upon completion, a program consisting of testing and computer modeling should be developed to verify new railcar designs and track safety standards.

## References


1. Cackovic, Dave L., Kerry Hopkins, David Tyrell. "Side Bearing Clearance Testing Research." Federal Railroad Administration, FRA/ORD/94-11, May 1994.
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