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A Parametric Study of Spring Bottoming

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Summary

Using a NUCARS™* model of a typical freight car, Transportation Technology Center, Inc. studied the effect of friction damping condition on spring bottoming. Spring bottoming is more likely to occur with a worn truck (worn friction surfaces) than with a new truck, due to the change in loaded suspension height. Although the results presented here are for a specific car and loading, they demonstrate how closely heavy axle load cars, with conventional spring nests, are getting to spring bottoming boundaries.

Spring bottoming is undesirable because it causes high vertical forces to be transmitted to the carbody, truck, and track. It is normally prevented by designing the spring nests to have a generous dynamic capacity (a measure of displacement from the loaded car to the solid spring conditions). This study has shown that dynamic capacity tends to decrease as car weight increases. Thus, it is important to understand the effect that key parameters have on the likelihood of spring bottoming.

The degree of wear on the friction wedges, bolster and side-frame columns, and the coefficient of friction on the surfaces are the two parameters that were studied. A NUCARS model of a loaded 110-ton hopper car was used for the parametric study.

In response to a simple vertical bump in the track, spring bottoming is unlikely unless the speed is high (>60 mph) and the irregularity height is greater than the reserve travel in the spring nest. Reserve travel is the vertical displacement between the loaded and solid spring heights.

Reducing the coefficient of friction between the wedges and the bolster and side-frame columns causes spring bottoming to occur at lower speeds, unless the bump height is less than the reserve travel or the coefficient of friction is very low. Spring bottoming is unlikely to occur when the vehicle speed is below 40 mph unless the bump height is very large (>2.0 inch).

In response to a series of parallel dipped rail joints, relatively small amplitude joints will cause spring bottoming at a critical speed. At this speed, a natural frequency of the vehicle corresponds to a harmonic of the track input. With a high coefficient of friction (≥ 0.4) on the faces of the wedges, spring bottoming is avoided in both the new and worn conditions even at the critical speed for joint depths less than 0.75 inch.

Close to the critical speed, spring bottoming is very sensitive to the coefficient of friction between the friction wedges and the bolster and sideframe column faces. In the worn condition, spring bottoming is likely with a joint depth of 0.75 inch and a coefficient of friction less than 0.4.

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*NUCARS is a trademark of Transportation Technology Center, Inc.



INTRODUCTION

Spring bottoming occurs when the dynamic displacement of a carbody is large enough to cause the coils of the springs to go solid. When this happens the inertia of the carbody produces large vertical forces on the carbody, truck, and track. This is shown in Figure 1, which is the calculated response of a car to a series of low rail joints. The spikes above 55 kips correspond to spring bottoming events.

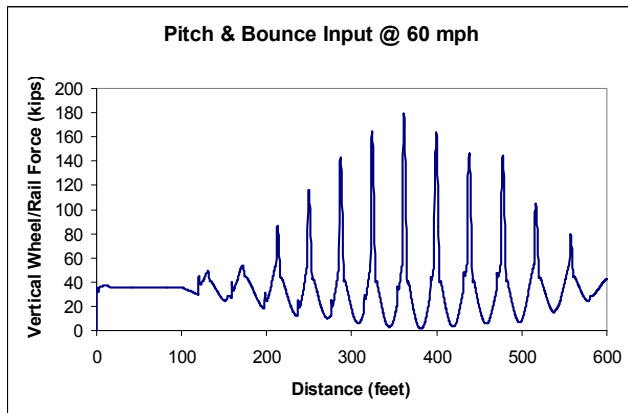


Figure 1. Spring Bottoming Forces

Spring nests are designed so that spring bottoming in service is a relatively rare occurrence. The approval of new designs of cars and trucks does not include a test for spring bottoming. Previous research has shown that spring bottoming is sensitive to speed and friction wedge conditions. This *Technology Digest* provides the results of a parametric study of these parameters. The results come from NUCARS simulations of a typical car. NUCARS has been demonstrated to give similar spring bottoming results to test data.²

Acceptance tests for rail cars are performed over sections of track with parallel and staggered low rail joints.³ This study investigated spring bottoming at parallel rail joints and at a simulated bump in the track.

DYNAMIC CAPACITY OF VARIOUS SPRING NESTS

The dynamic capacity of a spring nest is a measure of the travel from the loaded height to the solid height of the main coils. If D_L is the displacement from the free height (free height is typically 10 5/16 inch) to the loaded height, and D_S is the displacement from the free height to the solid height (solid height is typically 6 9/16 inch), then the dynamic capacity is defined as:

$$\frac{(D_S - D_L)}{D_S} \times 100 \%$$

The AAR *Manual of Standards and Recommended Practices* gives recommended spring nest configurations for various car weights.⁴ Truck manufacturers also give similar

recommendations. The dynamic capacity for several of these recommended configurations is plotted against car weight in Figure 2.

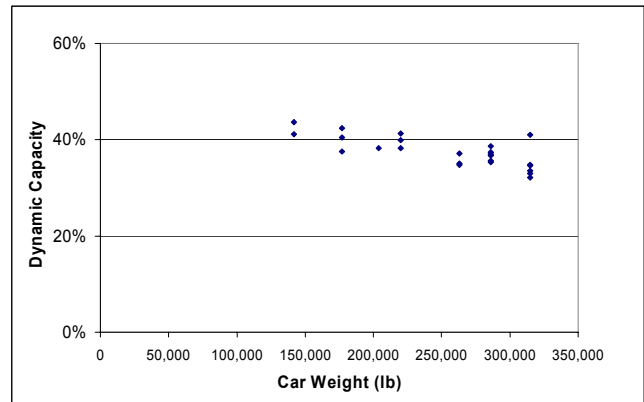


Figure 2. Variation of Dynamic Capacity with Car Weight

There is a trend for dynamic capacity to reduce as car weight is increased. This could mean that spring bottoming is more likely for heavier cars. When spring bottoming does occur, the heavier cars are likely to produce higher dynamic track forces. Hence, the need for an improved understanding of the factors that affect spring bottoming.

NUCARS MODEL

A NUCARS model of a loaded 110-ton hopper car (car weight 286,000 lb) was used for the parametric studies. The spring nest consisted of seven D5 outer coils, four D5 inner coils, and four D6A inner/inner coils at each spring nest. In addition, there were friction wedges providing variable damping with D7 inner and D7 outer control coils.

Two different damping conditions were modeled: new and worn. In the new condition, the friction wedge was modeled with a coefficient of friction on both rubbing surfaces of 0.4. In the worn condition, a 1-inch rise was given to the wedges. The worn wedges were modeled with coefficients of friction of 0.001, 0.1, 0.2, and 0.4.

The force/displacement characteristics for the spring nest in the new and worn conditions are shown in Figure 3 and summarized in Table 1. The car used in the parametric study is represented by a point towards the right of Figure 2.

Table 1. Spring Nest Characteristics

	Loaded Height (in.)	Reserve Travel (in.)	Dynamic Capacity (%)
New Wedges	7.950	1.3874	37.0
Worn Wedges	7.735	1.1725	31.3



Figure 3. Spring Nest Characteristics

MODELING INPUTS

Two different inputs to the NUCARS model were used in the parametric study. The first was a vertical bump consisting of a ramp-up followed by a ramp-down (Figure 4). The height of the bump was varied from 0.75 inch to 1.8 inches. The length of each ramp was kept constant at 7 feet. The flat portion on the top of the bump was 20 feet long. This input could represent the effect of increased track stiffness, such as at a grade crossing or a bridge. It was intended to investigate spring bottoming in response to a discrete input. The highest bump produced a mid-ordinate offset on a 62-foot chord, less than the FRA safety standard for Class 5 track.

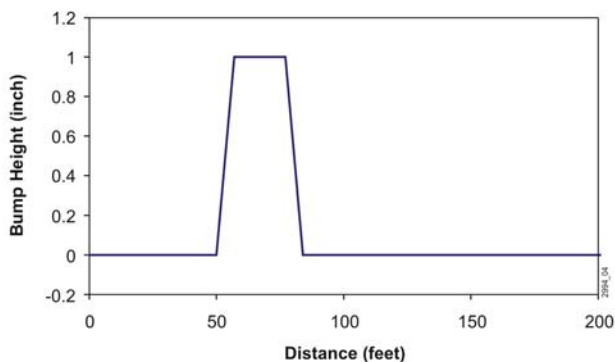


Figure 4. Bump Input

The second input was a series of parallel, dipped rail joints, as described in AAR’s Specification, M-1001, Ch XI.¹ The depth of the joints was varied from 0.2 inch to 3 inches. This input was intended to investigate spring bottoming in response to a periodic input.

RESULTS

Four sets of results were obtained from the parametric study. The first set is shown in Figure 5 and compares the spring bottoming boundaries for the new and worn wedges with the

bump input. If the combination of speed and bump height is below or to the left of the boundary, the springs are not expected to bottom. The boundary is the combination of speed and bump height that just causes spring bottoming.

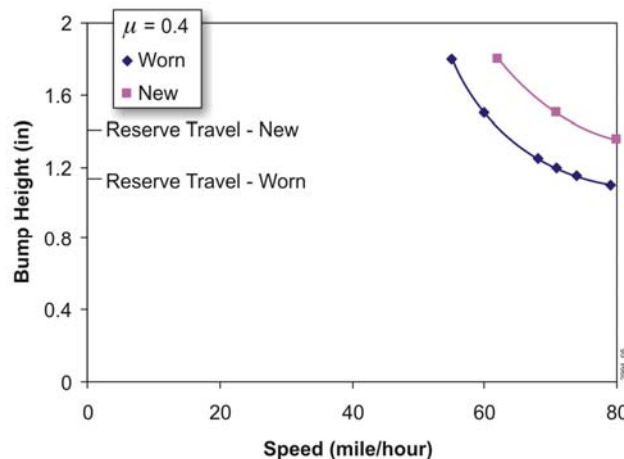


Figure 5. New and Worn Results with Bump Input

Spring bottoming only occurs with the new wedges at high speeds and with large bumps. It is more likely to occur with the worn wedges. If the bump height is significantly below the reserve travel of each spring nest, then spring bottoming is unlikely to occur no matter how fast the car is traveling.

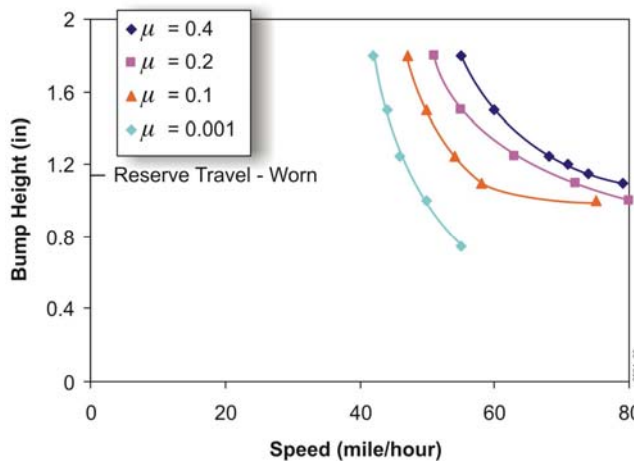


Figure 6. Worn Friction Results with Bump Input

Figure 6 shows the effect of varying the friction level on the worn wedges. Spring bottoming occurs at lower speeds as the coefficient of friction is reduced. This is due to a lightly damped response of the carbody to the bump input. Except for the very lightly damped case ($\mu = 0.001$), the spring bottoming boundary is independent of speed for bump heights less than the reserve travel.

Figure 7 compares the spring bottoming boundaries for the new and worn wedges with the parallel dipped rail joint input.

Spring bottoming is not expected to occur for combinations of speed and joint depth below the boundary.

Relatively small joints will cause spring bottoming at a critical speed. At the critical speed (approximately 53 mph), the natural frequency of the vehicle in its bounce mode (approximately 2 Hz) corresponds to the harmonic in the track input at 39 feet.

Figure 7 shows that, at any specified speed, spring bottoming occurs with a smaller joint depth for the worn wedges compared to the new wedges. In neither case is spring bottoming likely to occur when the joint depth is equal to the AAR Chapter XI value of 0.75 inch.

Figure 8 shows the results for worn wedges with a range of coefficients of friction with the parallel dipped rail joint input. The effect of reducing the coefficient of friction at any specified speed is to reduce the size of the joint required to cause spring bottoming. This is due to there being less damping in the vehicle modes of vibration when the friction is lower. With low coefficients of friction, spring bottoming is quite likely to occur close to the critical speed.

Away from the critical speed, the effect of coefficient of friction is less pronounced.

Below 40 mph, a relatively large joint depth is required to cause spring bottoming, unless the coefficient of friction is very low.

CONCLUSIONS

Spring bottoming is undesirable because it causes high vertical forces to be transmitted to the carbody, truck, and track. It is normally prevented by designing the spring nests to have a generous dynamic capacity (a measure of displacement from the loaded car to the solid spring conditions). This study has shown that dynamic capacity tends to decrease as car weight increases.

A parametric study of the effect of wedge condition on spring bottoming has been performed using NUCARS. Spring bottoming is more likely to occur with a worn truck (worn friction surfaces) than with a new truck running over a single, vertical bump. For the 110-ton car modeled, spring bottoming was only likely for high speeds (greater than 60 mph) and bump heights greater than the reserve capacity of the spring nests, unless the friction damping was very low.

For the parallel dipped rail joint input, a critical speed was found when the bounce of the carbody's natural frequency matched the input wavelength of 39 feet. Close to the critical speed, spring bottoming was found to be very sensitive to the coefficient of friction between the wedges and the bolster and side-frame columns. For the 110-ton car modeled, spring bottoming is likely to occur on low joints with depths equal to the Chapter XI depth of 0.75 inch, if the coefficient of friction is low (less than 0.4).

A test for spring-bottoming is recommended for inclusion in Chapter XI testing.¹ Further work is needed, however, to evaluate limits in other specifications,² and may involve correlation between vertical accelerations and spring travel.

REFERENCES

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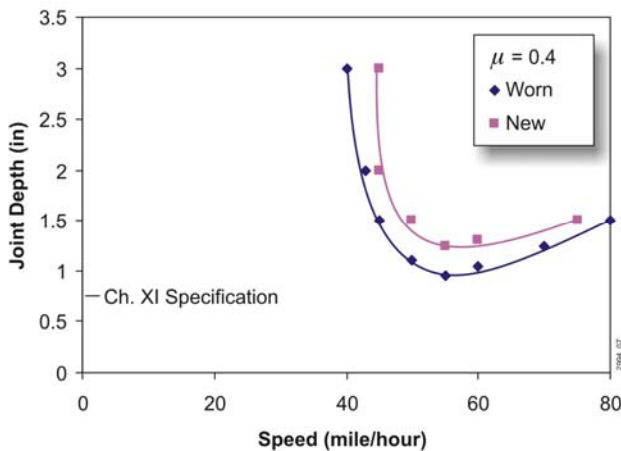


Figure 7. New and Worn Results with Pitch & Bounce Input

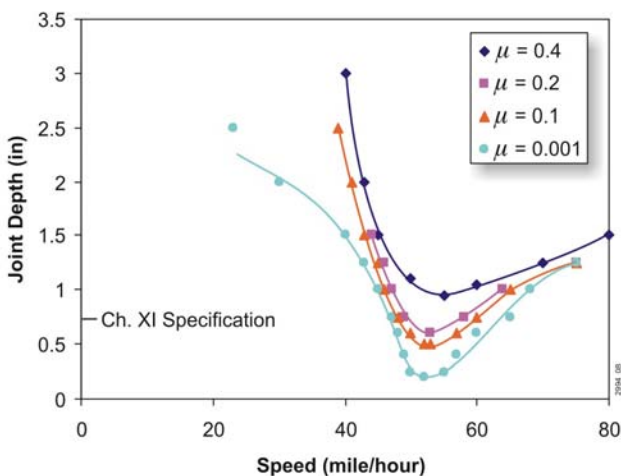


Figure 8. Worn Friction Results with Pitch & Bounce Input

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