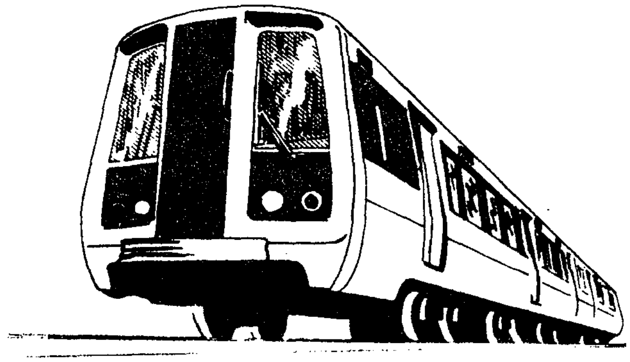
TECHNICAL NOTE



U.S. Department of Transportation
Urban Mass Transportation Administration

TRANSPORTATION TEST CENTER

AAR REPORT T-009

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DCTA TRANSIT CAR TESTING: STATIC TRUCK CHARACTERIZATION

by
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SUMMARY

This report discusses the Static Truck Characterization of the Dade County Transportation Administration (DCTA) Transit Car. This special engineering test was conducted in conjunction with General Vehicle Testing of the DCTA car at the Transportation Test Center near Pueblo, Colorado, from October 1982 to March 1983. Static Truck Characterization, described here, is an adjunct test series done to verify the manufacturer's specifications and to evaluate vehicle performance in stability, curving, and ride. The characterization data is used to develop mathematical models and provide a basis for study and comparison of performance parameters throughout the service life of the truck.

INTRODUCTION

The truck assembly of the Dade County transit car essentially consists of a steel truck bolster and a split H-frame connecting two steel sideframes with traction motor and gear box mounts. The truck and suspension subsystems include the components illustrated in Figure 1. The primary suspension is contained in the journal ring assembly bolted to the sideframe ends. The secondary suspension is provided by two air springs installed on the bolster under the carbody. The vertical and lateral damping of the air springs are provided by the hydraulic shock absorbers mounted between the truck bolster and the carbody. The following vehicle parameters, which have an important effect upon the lateral stability and curving performance of the transit vehicle, are measured during truck characterization tests:

1) Lateral spring rate of the primary suspension.

2) Primary suspension longitudinal spring rate.

3) Primary suspension vertical spring rate.

4) Truck rotational stiffness.

5) Axle alignment.

In addition to the above static tests, a simple resonance test was conducted on the test vehicle to establish values for the carbody on secondary suspension natural frequencies.

PRIMARY SUSPENSION LATERAL SPRING RATE TEST

To determine the suspension lateral spring rate, the third rail collector assembly was removed from the truck assembly and a special bracket was mounted to the current collector mounting plate. A hydraulic cylinder/load cell assembly was attached to the bracket and to a reacting structure by means of cables. A lateral force was applied to the truck sideframe assembly by a hydraulic pumping unit (Figure 2). The load was measured by the load cell and the lateral displacement of the sideframe relative to the stationary wheel (axle) was measured with the linear transducers and dial gages at each axle journal. A digital multimeter and a strip chart recorder recorded the applied force and the displacement of the sideframe with respect to the stationary wheel through suitable signal conditioning equipment. A maximum lateral load of 8000 lbs was applied at 500 lb increments.

Figure 3 presents the lateral spring rate of the primary bushings of the left front and left rear axle boxes, with force applied to the left sideframe bracket. During the entire loading range, no lateral deflection was noticed in the primary bushings of the right front and right rear axle boxes. This phenomenon is attributed to the flexibility of the split H-frame at its center with the side frames being independent of one another in the pitch direction. It is assumed that the total applied force was divided equally between the left front and rear bushings. The average primary lateral stiffness/journal was found to be 240,000 lbs/inch.

During the primary lateral stiffness test, dial gages were also inserted between the bolster and sideframes across the elastomeric side bearers to measure the deflection between the sideframe and the bolster. With the application of lateral load to the left sideframe bracket, no displacement was observed between the right sideframe and the bolster, whereby it was assumed that the entire lateral load was felt by the left side bearer elastomer pad only. Figure 4 presents the lateral spring rate of the side bearer elastomer. The average lateral stiffness between the bolster and sideframe was 375,000 lbs/inch.

PRIMARY SUSPENSION VERTICAL SPRING RATE TEST

With the test car parked over the rails of the Urban Rail Building (URB), two jacks were placed underneath the jacking pads of the carbody near the test truck. 20 kip load cells were placed between the jacks and the carbody. The jacks were so operated that both the load cells registered a minimum amount of compressive preload. The secondary suspension air springs were connected to the shop air supply. Both wheelsets of the truck were incrementally relieved of their vertical loading by reducing the air pressure in the secondary suspension air springs, and the relieved weight was registered by the two load cells between the jacks and the carbody. The load carried by the primary suspension was measured by four dial gages, one at each axle box. At this point, the air pressure in the secondary suspension air springs was increased incrementally and the vertical load on the load cells was relieved by transferring the weight back to the wheelsets resting on the rails. The loads registered by the load cells were monitored by a digital multimeter and a strip chart recorder, with suitable signal conditioning equipment. Figure 5 presents the primary vertical spring rate based on the average deflection of left front and rear primary bushings. Figure 6 presents the vertical spring rate characteristic based on the average deflection of right front and rear primary bushings. From the slopes of the two vertical spring rate characteristics, the primary vertical stiffness/journal was estimated to be 420,000 lbs/inch.

During the primary vertical spring rate test, dial gages were also installed across the side bearer elastomer pads between the bolster and the sideframes to measure their vertical stiffness. Figure 7 presents the vertical spring rate characteristic of the side bearer elastomer. From the slope of this characteristic, the vertical stiffness of the side bearer elastomer is estimated to be 400,000 lbs/inch.

PRIMARY SUSPENSION LONGITUDINAL SPRING RATE TEST

One wheelset of the test truck was supported on an air bearing table with the second wheelset resting on the floor. The brake shoes for the floating wheelset were removed and suitable load cells were assembled between the actuating lever and wheel tread so that when the corresponding brake cylinders were activated, longitudinal load was transmitted to each wheel via the load cell. The load cells directly measured the applied force. The brake cylinders were energized by the shop air with a pressure regulator to precisely vary its pressure in increments. The dial gages were positioned to measure the longitudinal displacement between the wheelset and the truck sideframe. LVDT's were also mounted across the rubber bushings to measure directly the longitudinal deflection of the primary suspension. Digital multimeter and strip chart recorder monitored the incremental longitudinal load and the corresponding deflection across the primary suspension. Figure 8 presents the longitudinal spring rate characteristic of the primary bushing. From the slope of the above characteristic, the longitudinal stiffness of the primary bushing is found to be 630,000 lbs/inch.

TRUCK ROTATIONAL STIFFNESS TEST

The break-away value of the yaw torque between the truck and the carbody, was established in this test.

Both the axles of the test truck were supported on a single air bearing table so that the truck was allowed to yaw freely with respect to the ground. Equal and opposite lateral forces were applied at the diagonally opposite corners of the air bearing table. The magnitudes of the applied forces were measured using load cells. The angular displacement of the air bearing table with respect to the ground was measured using a string potentiometer and dial indicators. With the truck floating freely on the air bearing table, the applied forces were increased slowly up to the point where gross rotation of the truck took place. Dial gages were also installed between each sideframe and the bolster to monitor the rotation of the sideframe with respect to the bolster, before and during gross rotation. Digital multimeters and strip chart recorders recorded the force levels and the rotational displacement. Figure 9 presents the rotational torque versus the displacement of the sideframe with respect to the bolster. The magnitude of the break-away torque from the levels of applied forces required to obtain gross rotation was found to be 10,500 lb ft.

AXLE ALIGNMENT TEST

The alignment of the axles of the test truck was measured using a Brunson optical transit tool. The wheels of the truck were positioned on air bearings with each axle being independently supported. Precision scales were positioned against the front faces of the wheels at four locations. The scales were sighted through the optical instrument which determined the lateral location of scale from an optical line-of-sight (Figure 10). From the four lateral distances between the outside faces of the wheels and the optical line-of-sight, the angular misalignment of the two axles with respect to one another was computed (Figure 11). The air bearing tables allowed the two axles to take up a relative position in which the primary suspension was unstrained. When the air bearing tables were deflated and lowered to the ground, it was assumed that the primary suspension remained unstrained.

The misalignments of the axles for the A-end truck are presented in Table 1 and Figure 12. As shown, the misalignment can be considered to have two components--a radial misalignment, wherein the axles have equal and opposite angles with respect to the truck centerline, and a lateral offset of one axle relative to the other.

VEHICLE RESONANCE TESTS

The natural frequency of the carbody on the secondary suspension was evaluated by hand excitation of the vehicle in the following modes: i) bounce, ii) pitch, iii) yaw, and iv) lower center roll. Hydraulic dampers installed between the bolster and carbody were removed and the vehicle was excited manually to determine the natural frequencies of the carbody on secondary suspension, with the air bag pressure maintained at 90 psi.

The points of excitation on the carbody were selected to excite the required mode of vibration. Figures 13 through 16 present the recommended points of application of manual excitation for the evaluation of rigid body modes in bounce, pitch, yaw, and lower center roll. Once the vehicle was set in motion in the required mode, the natural frequency was determined by measuring the time taken for given cycles of oscillation. The signals from the floor accelerometer were put on a strip chart recorder and the frequencies were calculated by counting cycles over a 30 second period.

The results of the vehicle resonance test are presented below:

RIGID BODY MODE	CRITICAL FREQUENCY (HZ)
BOUNCE	1.250
YAW	1.327
PITCH	1.467
LOWER BODY ROLL	0.676

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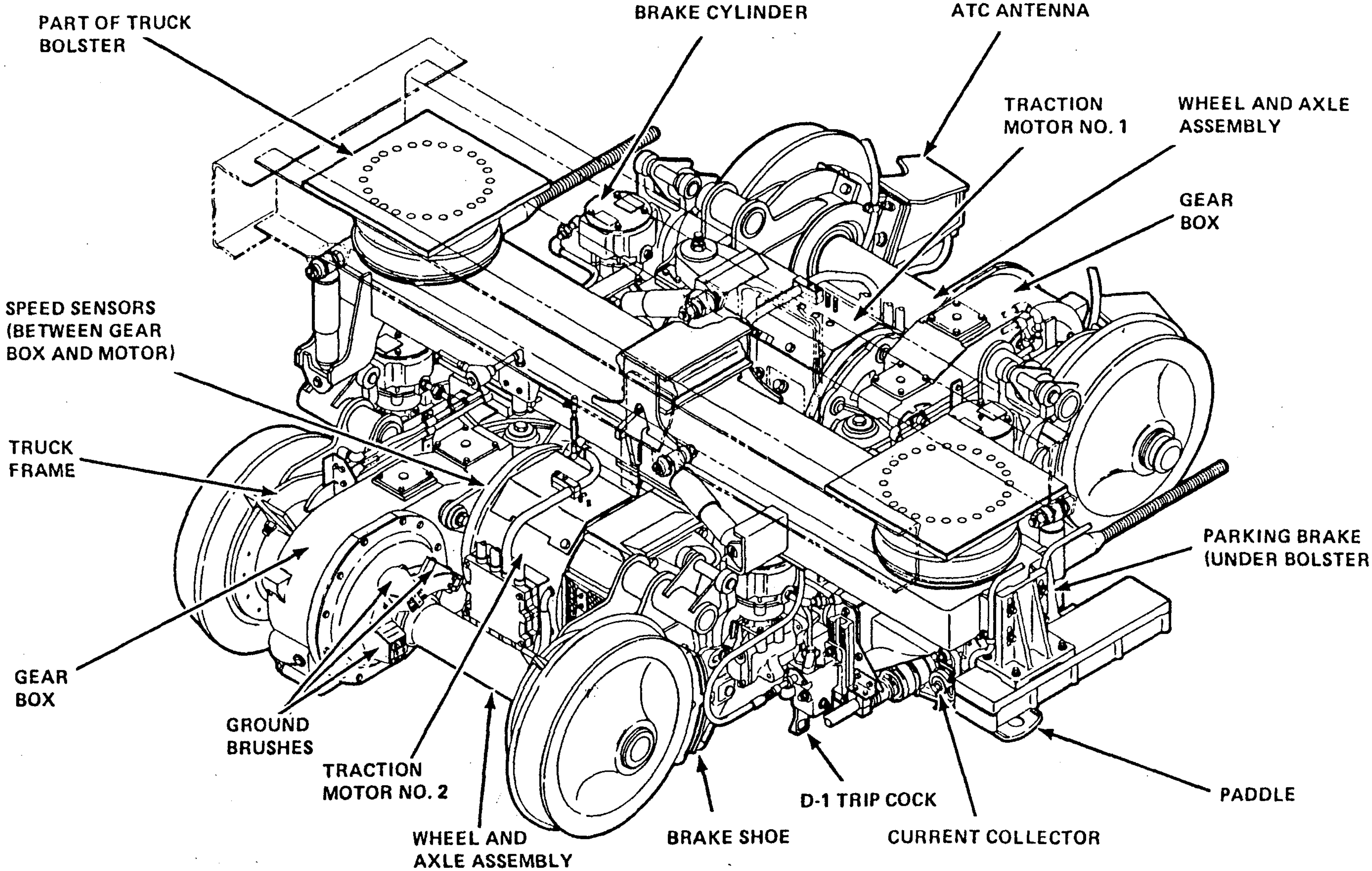

FIGURE 1. TRUCK ASSEMBLY ARRANGEMENT.

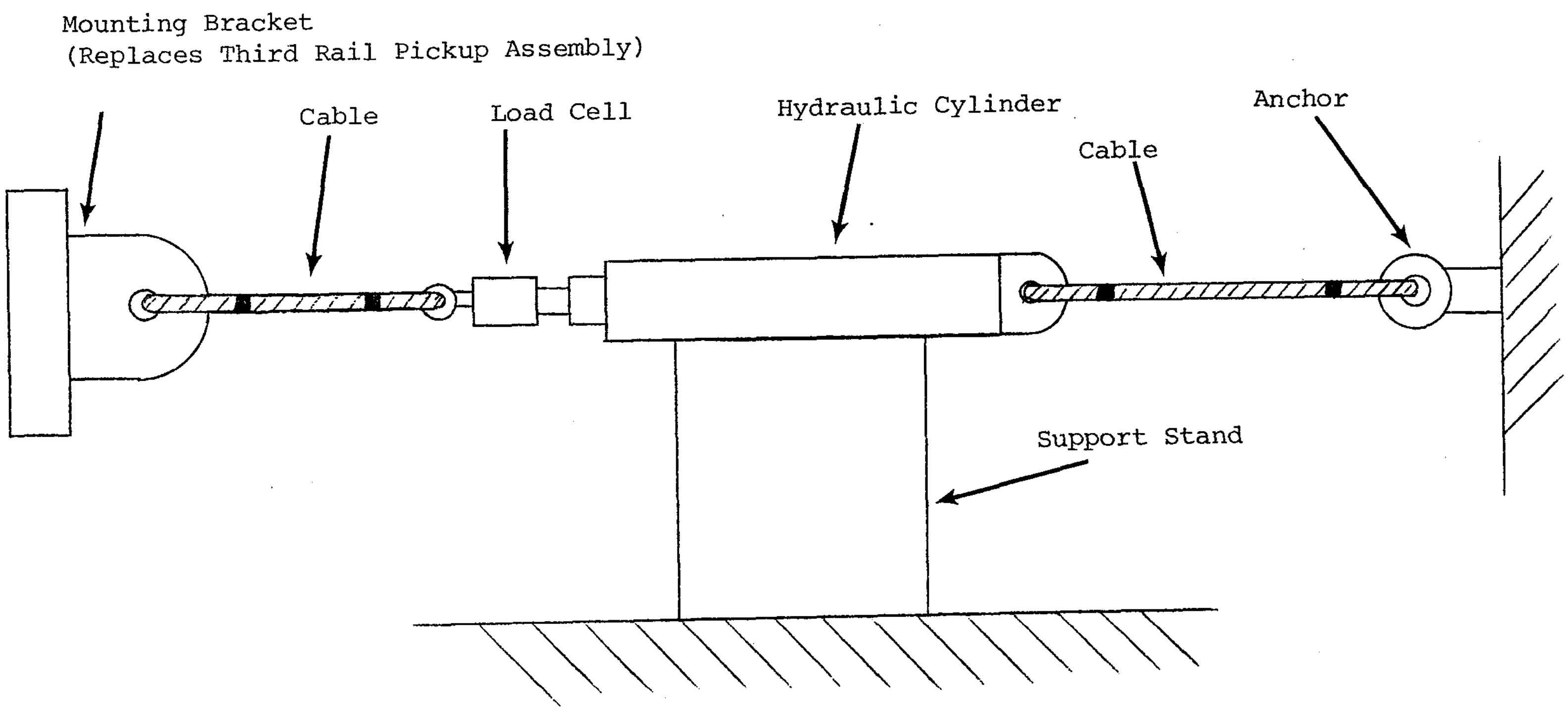

FIGURE 2. LATERAL LOAD ARRANGEMENT.

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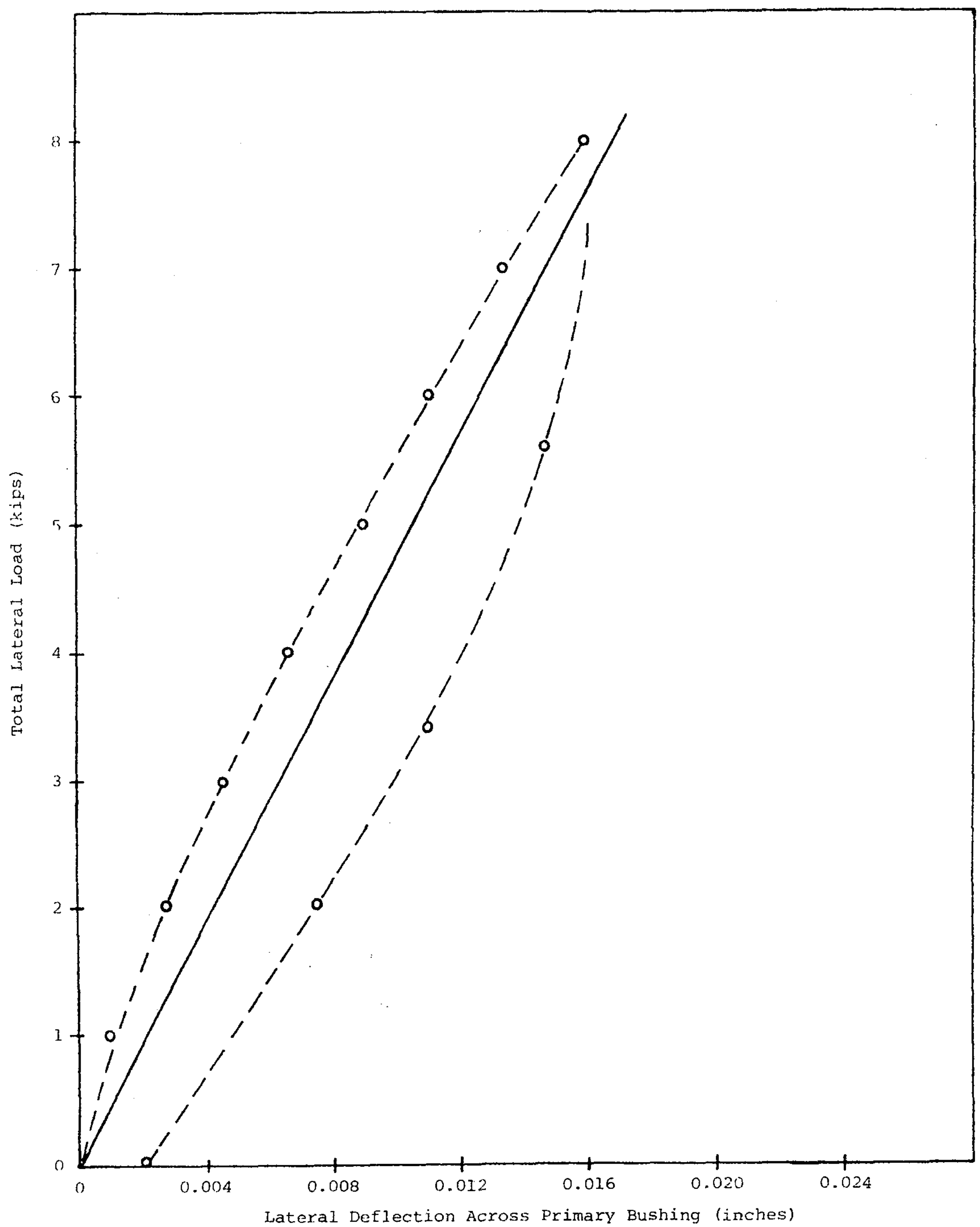


FIGURE 3. LATERAL STIFFNESS CHARACTERISTIC OF PRIMARY BUSHING.

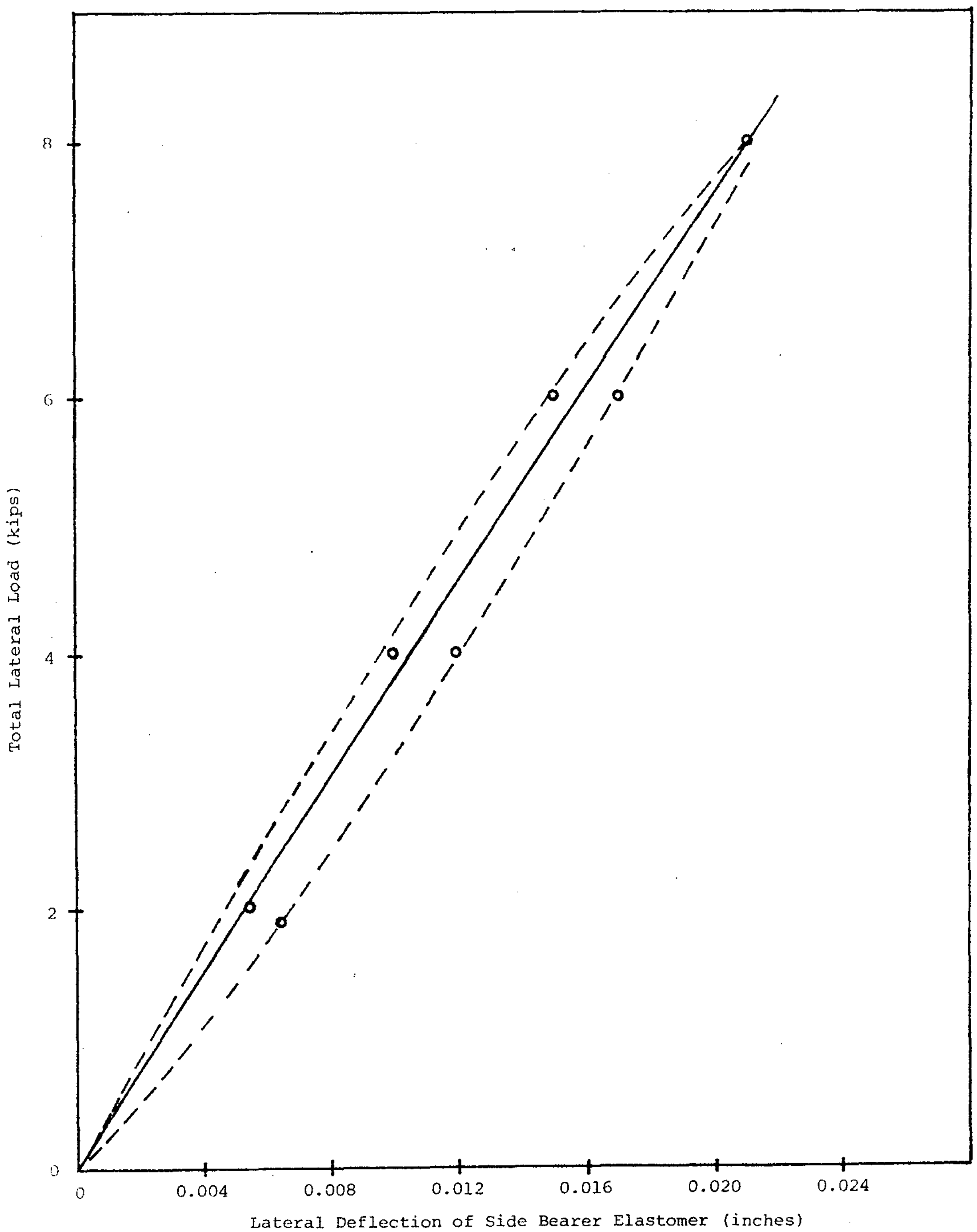


FIGURE 4. LATERAL SPRING RATE CHARACTERISTIC OF SIDE BEARER ELASTOMER.

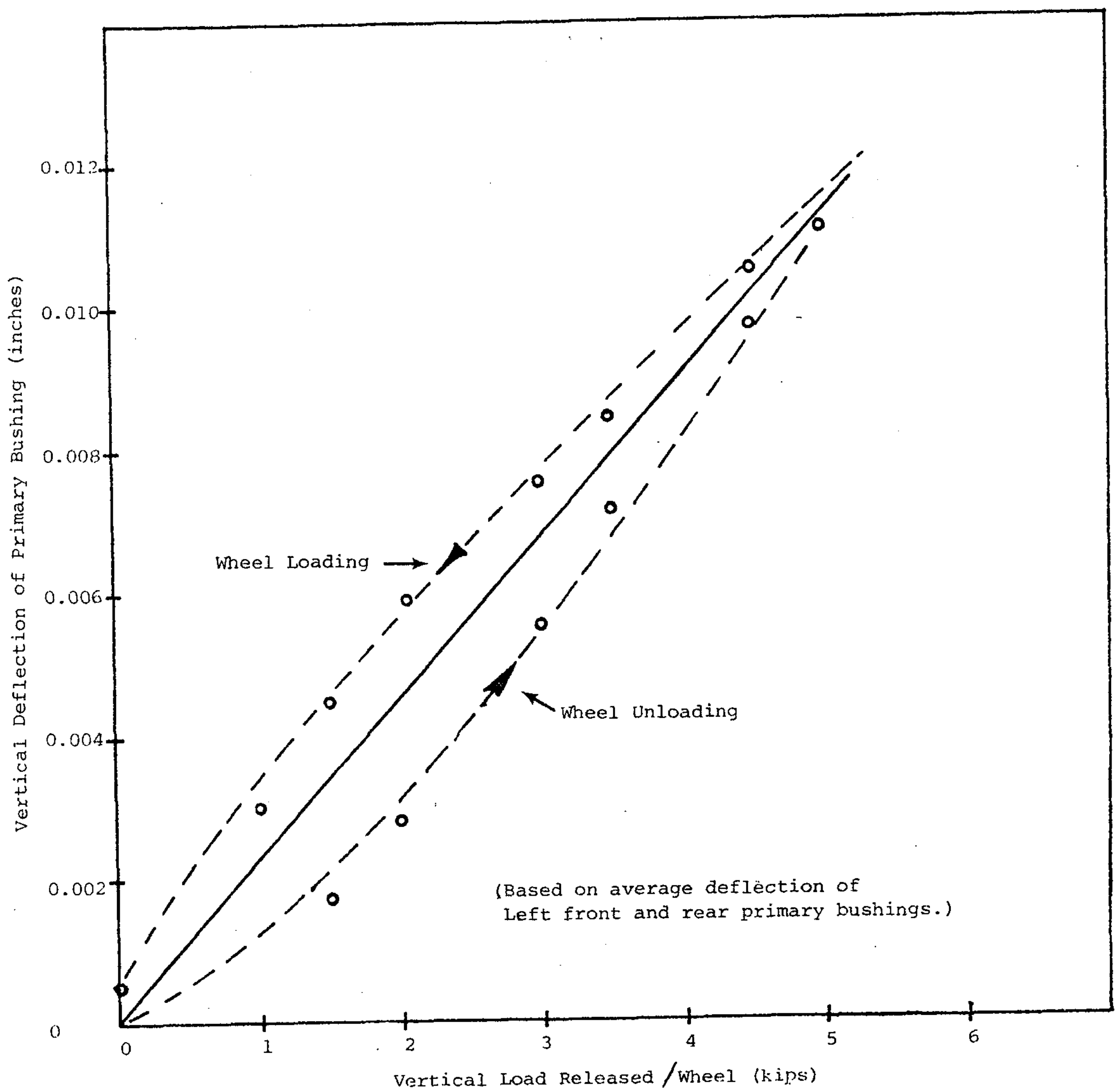


FIGURE 5. VERTICAL SPRING RATE CHARACTERISTIC OF PRIMARY BUSHING (STARTING FROM AW2 WHEEL LOADING).

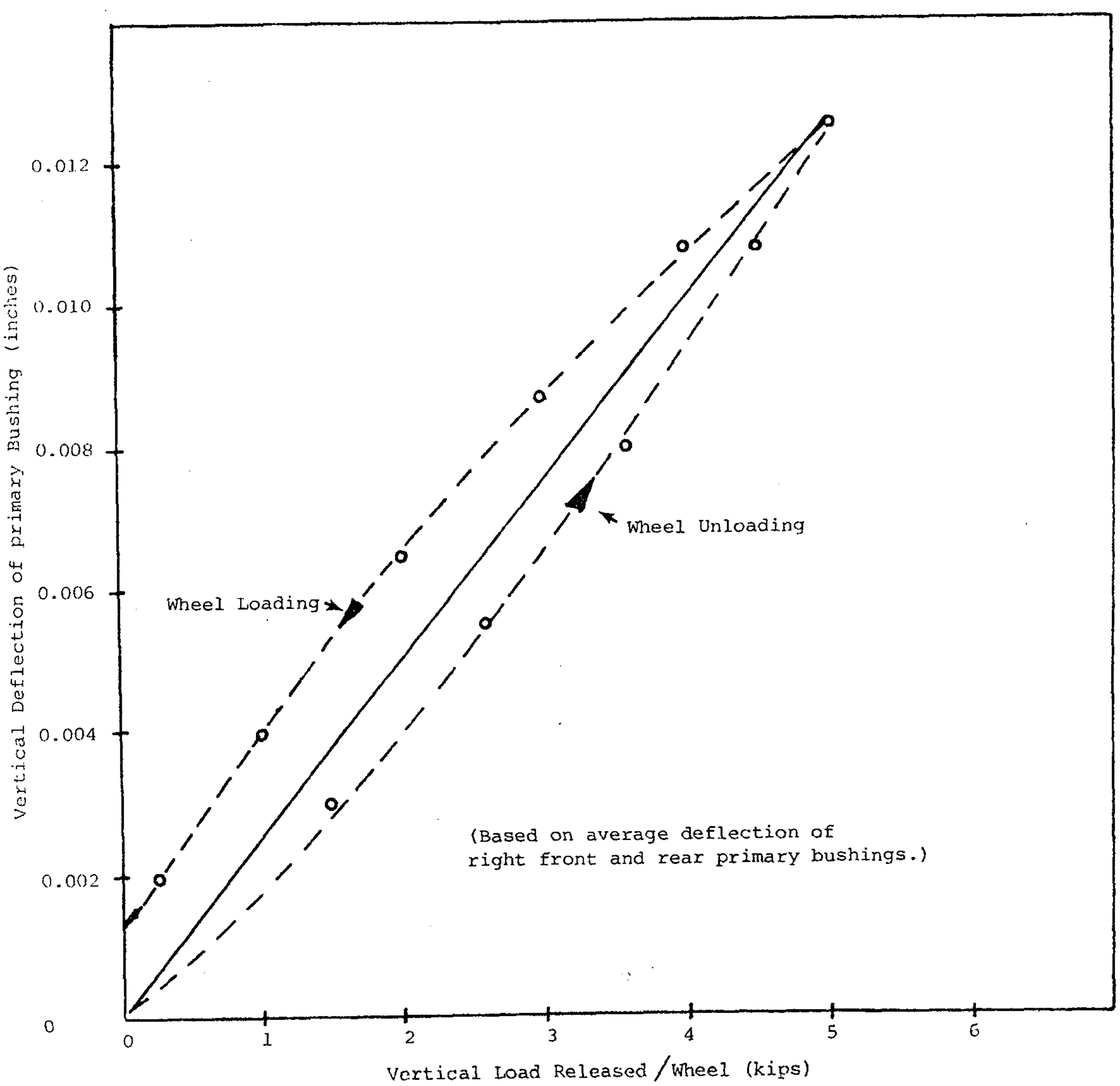


FIGURE 6. VERTICAL SPRING RATE CHARACTERISTIC OF PRIMARY BUSHING (STARTING FROM AW2 LOADING).

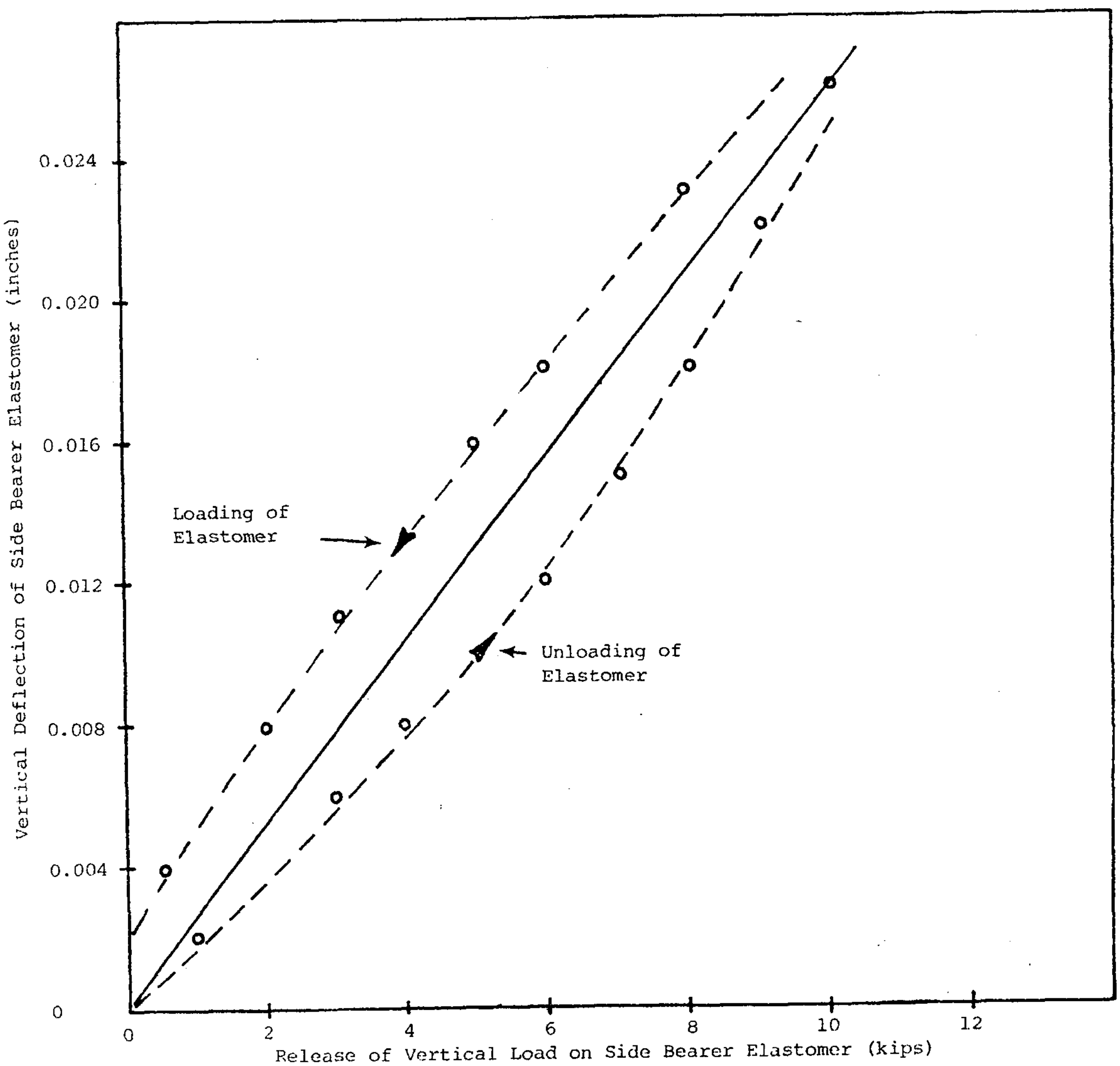


FIGURE 7. VERTICAL SPRING RATE CHARACTERISTIC OF SIDE BEARER ELASTOMER.

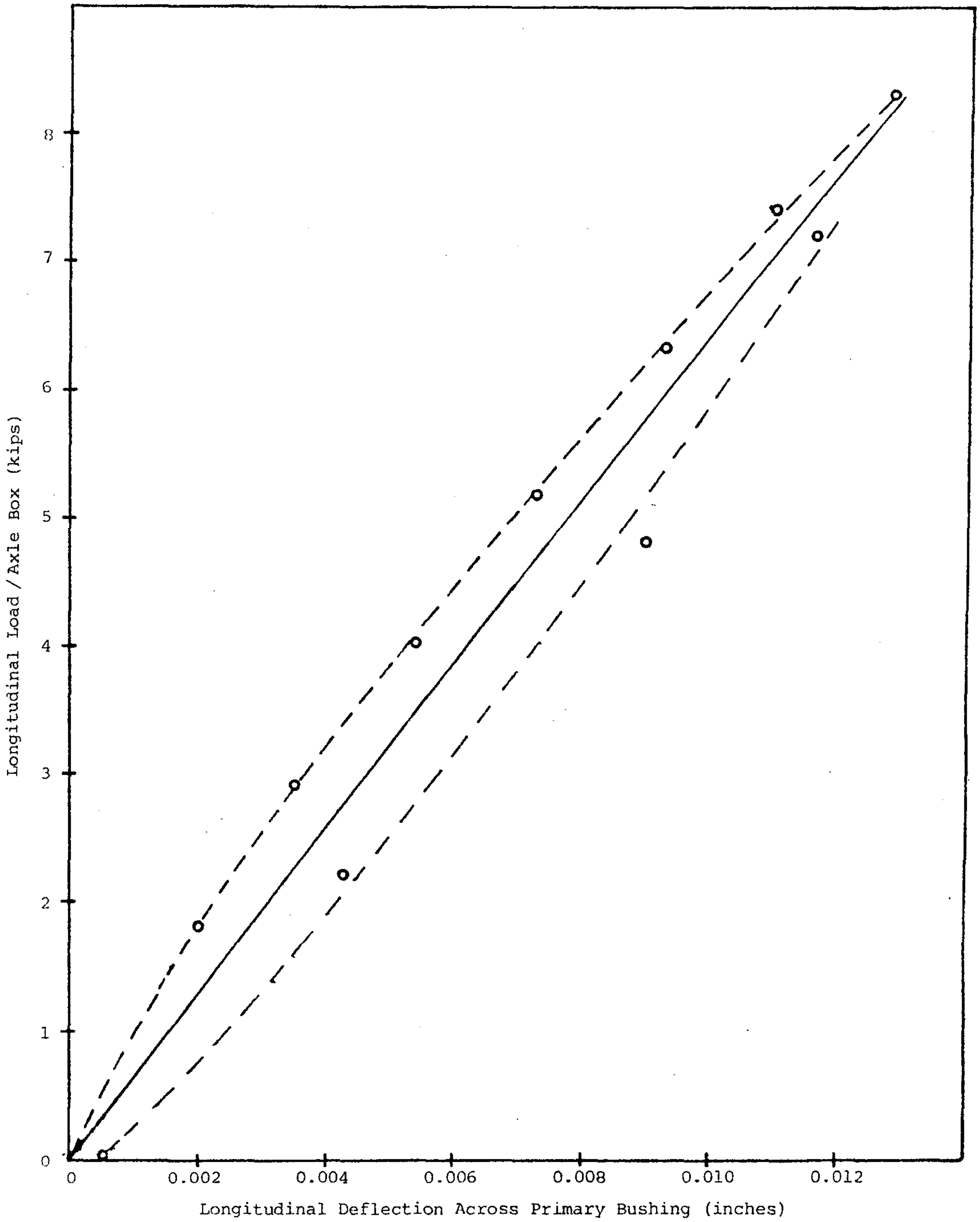


FIGURE 8. LONGITUDINAL SPRING RATE CHARACTERISTIC OF PRIMARY BUSHING.

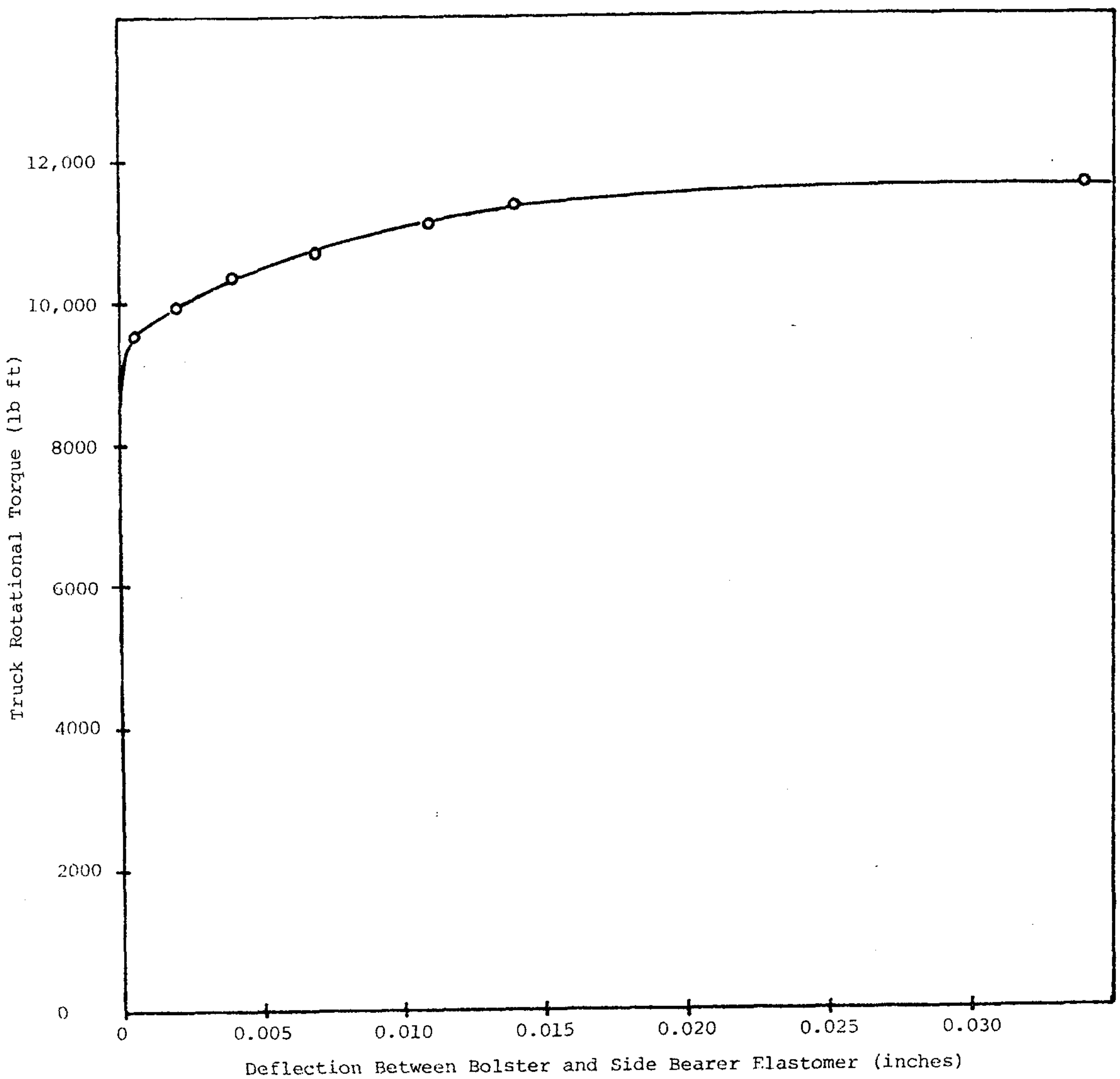


FIGURE 9. SECONDARY YAW TORQUE CHARACTERISTIC.

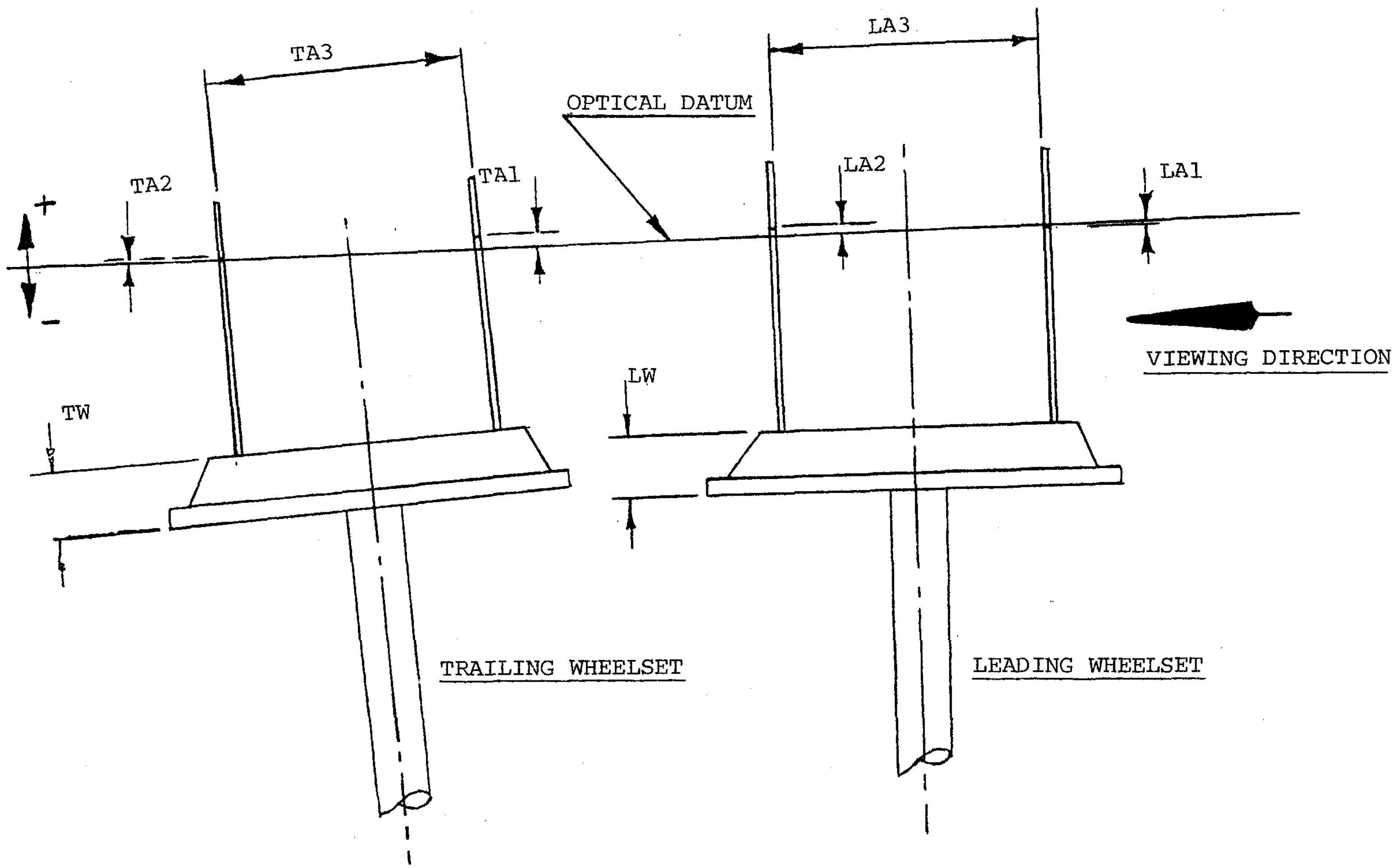
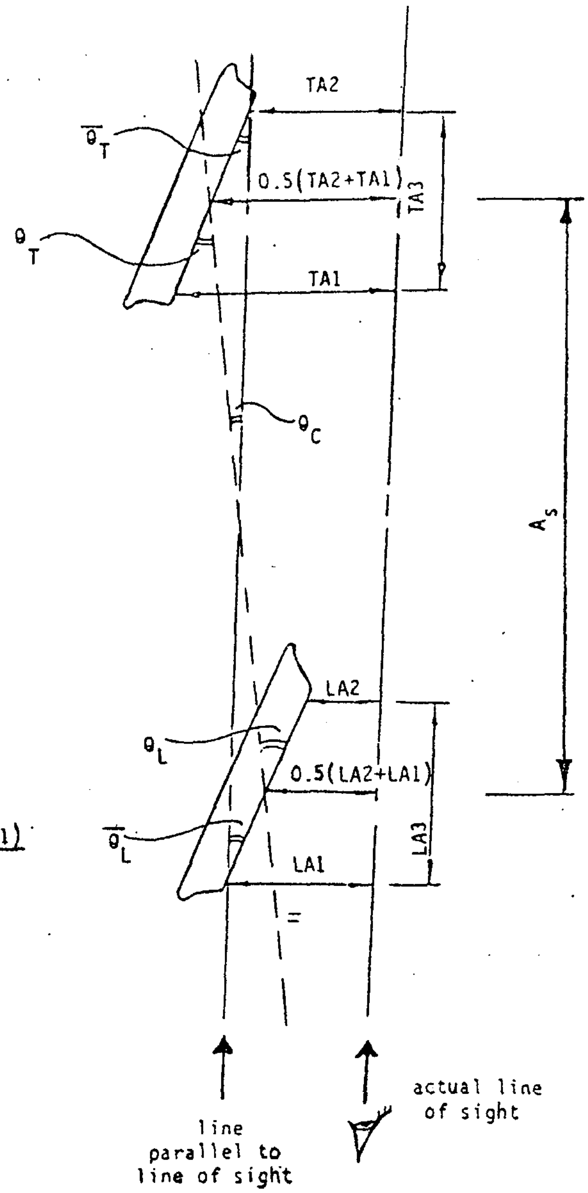


FIGURE 10. SCHEMATIC OF AXLE ALIGNMENT MEASUREMENT.

OBJECTIVE:

To Calculate θ_T and θ_L



$$\theta_T = \bar{\theta}_T - \theta_C$$

$$\theta_L = \bar{\theta}_L - \theta_C$$

Where:

$$\bar{\theta}_T = \frac{(TA1 - TA2)}{TA3}$$

$$\bar{\theta}_L = \frac{(LA1 - LA2)}{LA3}$$

$$\theta_C = \frac{0.5\,(LA2 + LA1) - 0.5\,(TA2 + TA1)}{A_s}$$

$$= \frac{LA2 + LA1 - TA2 - TA1}{2A_s}$$

FIGURE 11. AXLE ALIGNMENT CALCULATION.

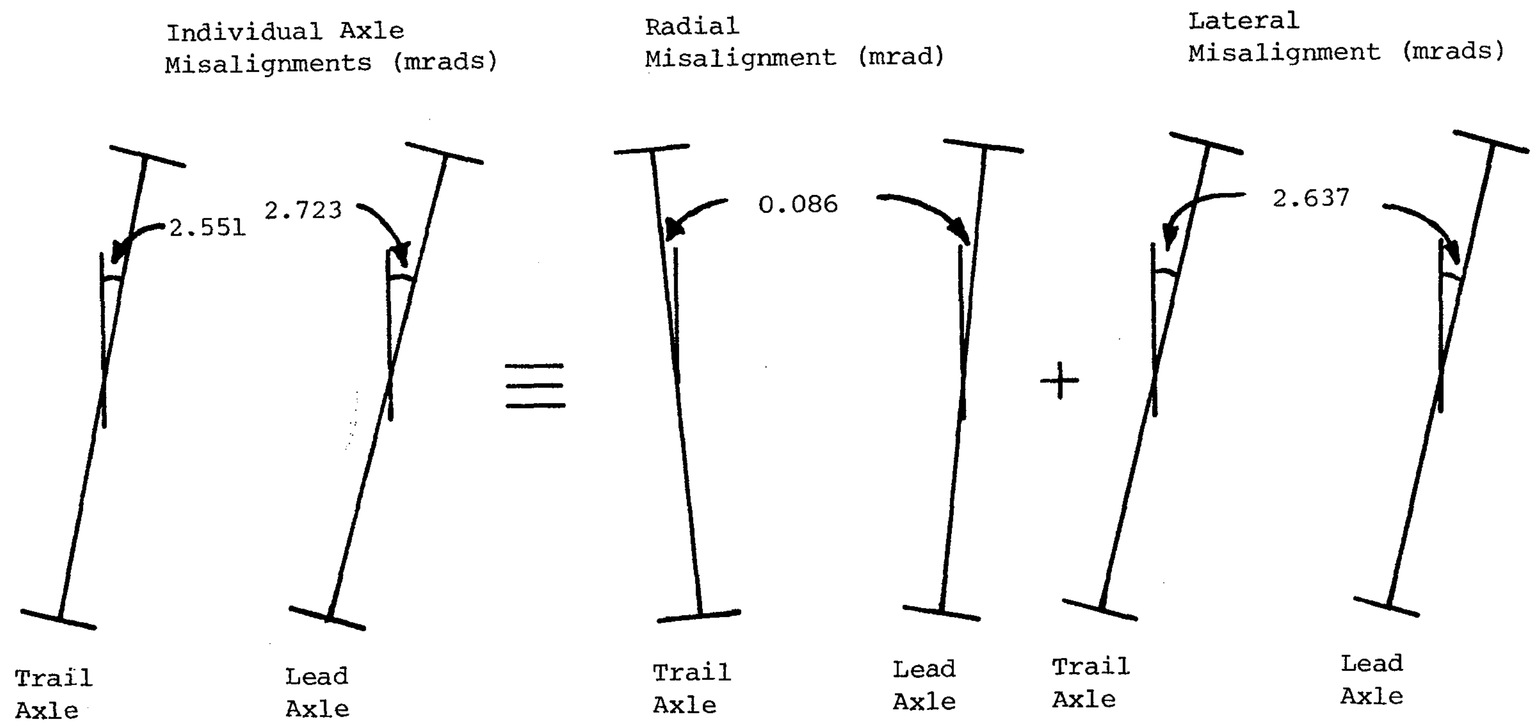


FIGURE 12. AXLE MISALIGNMENTS OF TEST TRUCK.

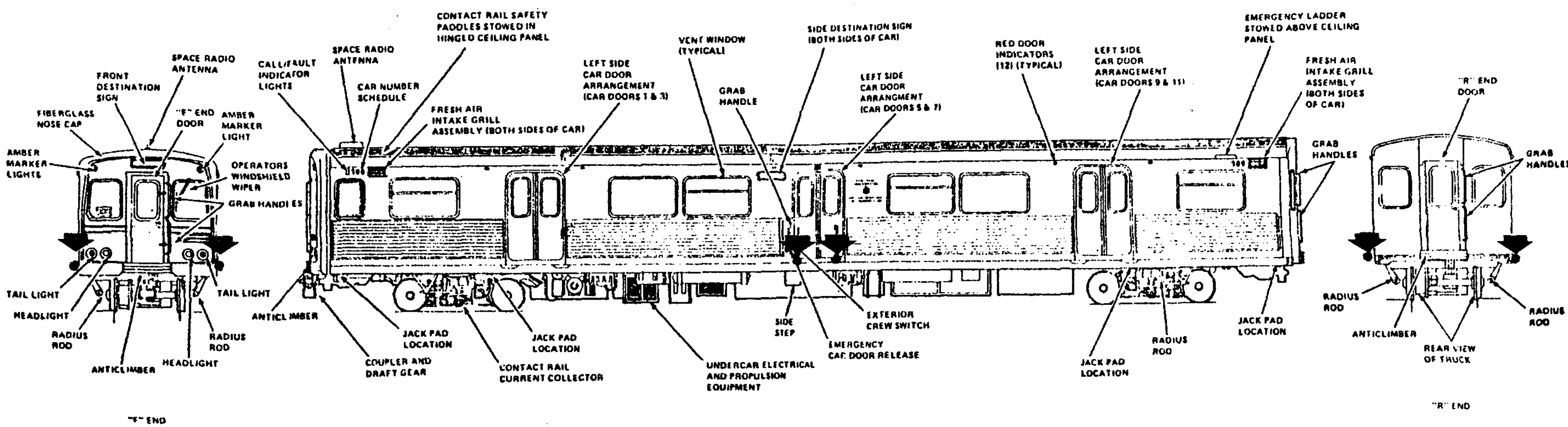


FIGURE 13. BOUNCE MODE EXCITATION.
Points of Force Application: At the Center of the Vehicle on Either Side, In Phase.

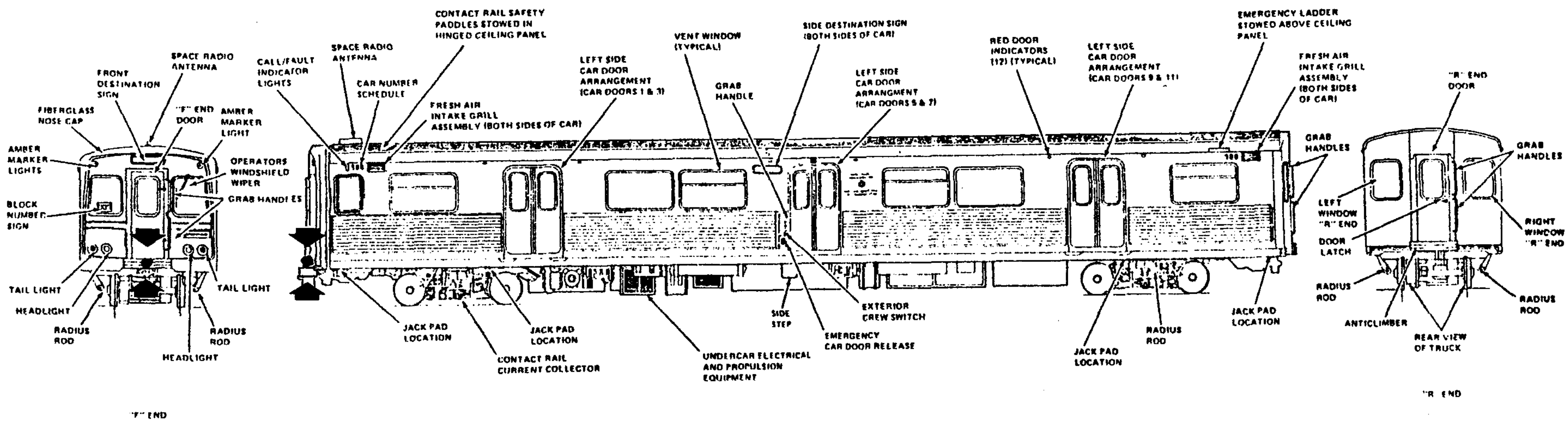


FIGURE 14. PITCH MODE EXCITATION.
Points of Force Application: At the Center of Front End of the Vehicle.

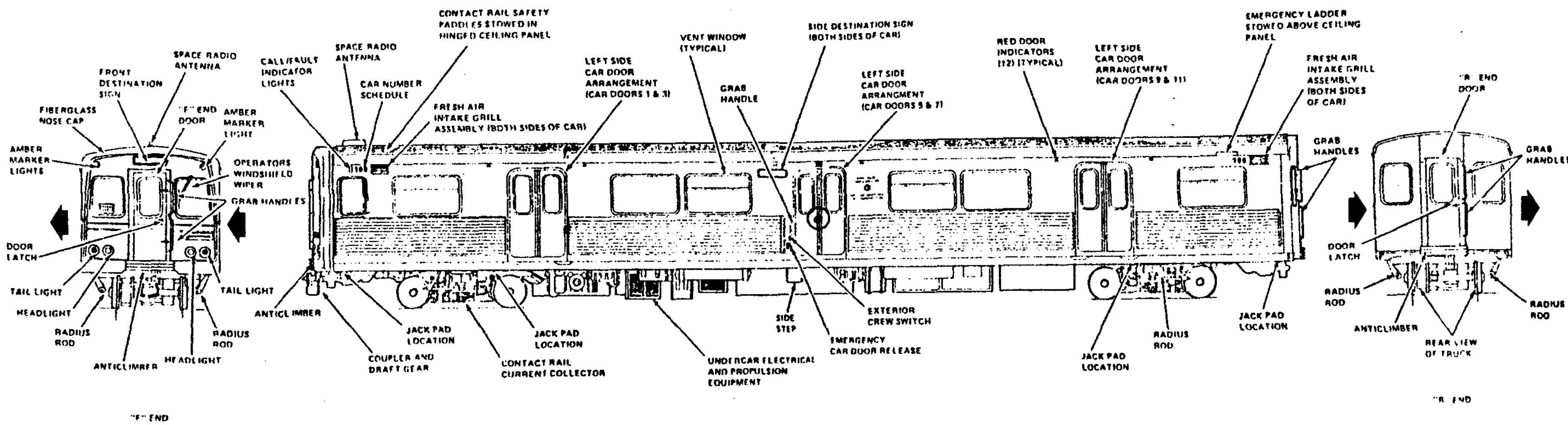


FIGURE 15. YAW MODE EXCITATION.

Points of Force Application: In the Lateral Direction at Diagonally Opposite Ends of the Vehicle in Line with the Truck Centers, Out of Phase 180 Degrees.

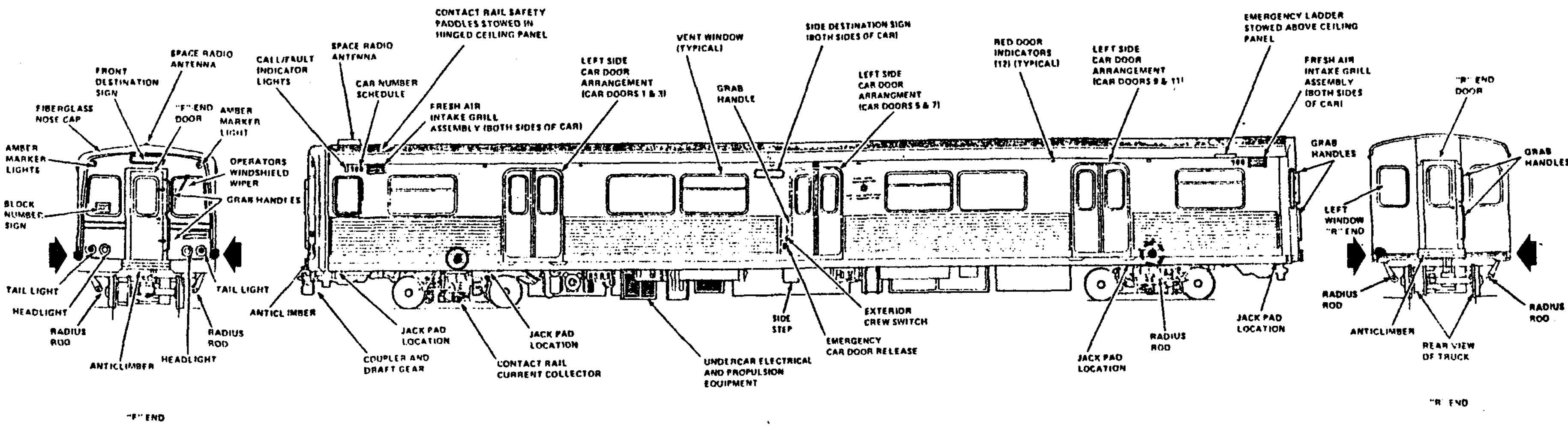


FIGURE 16. LOWER CENTER ROLL EXCITATION.
Points of Force Application: On Either Side, at the Center of Vehicle Midway Between Sill and Roof Levels, In Phase.

TABLE 1. AXLE ALIGNMENT CALCULATIONS.

(ALL ANGLES EXPRESSED IN MILLI RADIANS)

TEST NO.	$\overline{\theta}_T$	$\overline{\theta}_L$	θ_C	θ_T	θ_L	θ_T Average	θ_L Average
1	2.157	2.35	-0.483	2.64	2.833		
						2.551	2.723
2	2.100	2.25	-0.362	2.462	2.612		

INDIVIDUAL AXLE MISALIGNMENTS OF TRAIL & LEAD AXLES WITH RESPECT TO TRUCK CENTERLINE = 2.551 & 2.723 mrads

RADIAL MISALIGNMENT = 0.086 mrads

LATERAL MISALIGNMENT = 2.637 mrads