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EFFECTS OF LONGITUDINAL IMPACT FORCES ON FREIGHT CAR TRUCK BOLSTERS

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16. Abstract

The design of truck bolster center plate rims was investigated as a result of increased reports of their failure on 100-ton capacity freight cars. The damage occurs when cars are coupled at moderate to high speeds, since the rapid deceleration of the truck causes high loads between the truck and body bolsters. Test measurements were made on an unloaded 100-ton hopper car impacting a string of loaded cars. The forces between the truck and body bolsters on the moving car were determined at impact speeds from 2.9 to 9.2 mph. Tests were made with two different energy absorbing capacities of draft gear.

Loads at the truck-bolster/body-bolster interface averaged approximately 40,000 lbs for impact velocities up to 5 mph and reached 100,000 lbs at 7 mph. A peak load of 160,000 lbs was measured at 8.4 mph. Within the lower speed range there were no significant differences in load associated with the two draft gear, but at 6.7 mph the loads with the higher capacity gear were 25 percent less. Strain gages placed near the center rim indicated yielding on the first impact at 2.9 mph. Additional yielding continued as the impact velocity was increased.

A finite-element stress analysis showed that loads of the magnitude measured on the test would cause severe stresses in the center plate rim and that yielding of the material would be expected. Several potential modifications of the truck bolster center plate rim were analyzed which showed that significant improvements could be obtained by making the rim wider and by increasing the radius of the fillet at the inside of the rim.

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PREFACE

The work described in this report was conducted by IIT Research Institute (IITRI) under the authorization of Transportation Systems Center (TSC). The IITRI project manager was Dr. M. R. Johnson. The finite-element analysis performed as part of the program was conducted by Dr. R. E. Welch and Mr. G. Ojdrovich of IITRI.

The project involved analysis and interpretation of data obtained from tests on freight car truck bolsters. The test work was performed at the Miner Corporation, Research and Development Division, Chicago, under the direction of Mr. E. J. Hart. This organization also installed the transducers and recorded the data. The instrumented bolsters and the hopper car in which they were mounted during the impact tests were furnished by the Burlington Northern (BN) through coordination with Mr. R. E. Taylor, Assistant Vice President, Mechanical. Mr. G. Breeding of BN witnessed these tests.

Throughout the project frequent consultations were held with Mr. R. Evans, Project Director, Railway Progress Institute/ Association of American Railroads (RPI/AAR), Truck Research Safety and Test Project, and Mr. R. Cook of the AAR staff. Through the effort of these individuals, the AAR provided high-speed motion picture coverage of the impact tests.

The cognizant Federal Railroad Administration technical monitor during the initial stages of the program was Mr. D. Bray. During the latter phases of the program, this responsibility was transferred to Mr. R. Steele, TSC. This report covers one phase of a more extensive program for the evaluation of freight car truck and wheel fatigue, which is being conducted by IITRI under contract to TSC. A report describing this work is scheduled to be published in June 1974.

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1. INTRODUCTION

1.1 Background

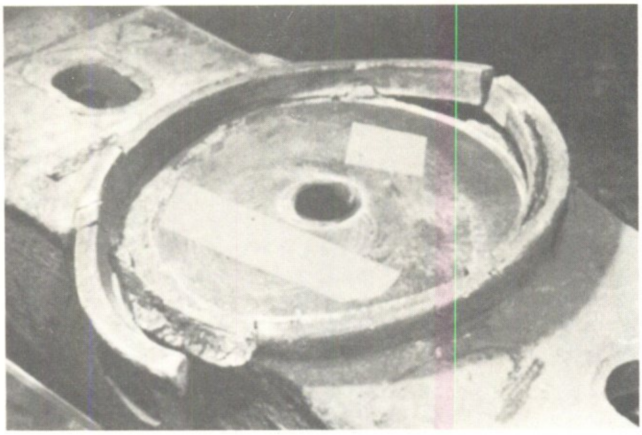
This program was initiated by the Federal Railroad Administration (FRA) as a result of concern over the safety and economic aspects of failures in freight car truck bolster center plate rims.

Some railroads have experienced an increase in the failure frequency of this component over the last several years.* Typical failures are illustrated in Figure 1. Rim failure on both sides of the bolster along a plane 45 degrees from horizontal is illustrated in Figure 1(a). A shear failure along a horizontal plane is shown in Figure 1(b). Rim failure occurring on only one side of the bolster is shown in Figure 1(c). In each case, the rim also failed in the circumferential direction.

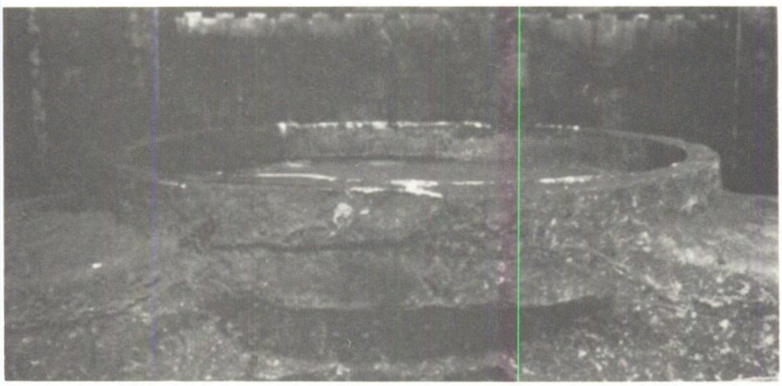
Discussions with individuals within the industry have revealed that truck bolster center plate failure is most frequent on 100-ton cars. The Burlington Northern (BN), for example, has experienced high failure rates on 100-ton hopper cars which are used in the unit train movement of low-sulphur coal. The failures have occurred on bolsters made by various manufacturers and have not been confined to any one bolster design.

It is recognized that many thousands of 100-ton cars will be ordered by the railroads over the next several years as unit train movements of coal and other bulk commodities increase. If failures of center plate rims necessitate the premature removal of truck bolsters from service, a substantial economic loss will be experienced. Safety is another consideration. Although the initial fracture of the rim rarely leads to immediate derailment, it marks the beginning of progressive destruction of the rim and shearing of the center pin, which eventually will result in separation of the truck from the car.

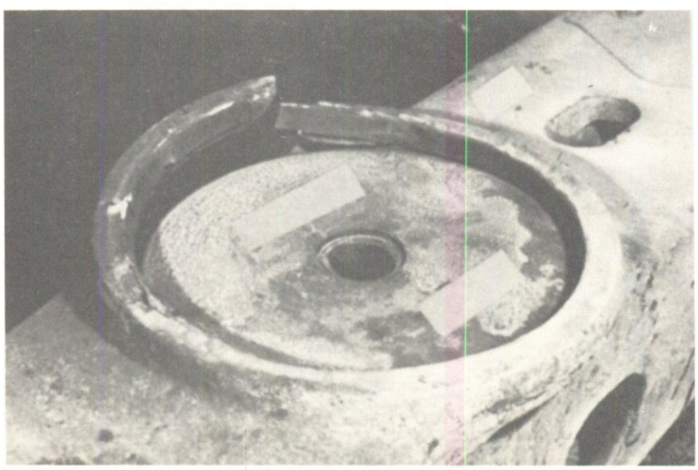
*See Proceedings of the 1972 Railroad Engineering Conference, Dresser Transportation Equipment Division, Dresser Industries Inc., pp 36 and 39.



(a) Failure on Opposite Sides of the Bolster along a Plane 45 deg from Horizontal



(b) Failure of the Rim along a Horizontal Plane



(c) Failure on One Side of the Bolster

Figure 1. EXAMPLES OF TRUCK BOLSTER CENTER PLATE RIM FAILURES (photographs courtesy AAR Technical Center)

An examination of the various aspects of truck bolster center plate rim loadings led to the conclusion that the impact of a car into other cars, such as occurs during switching operations, represents one of the most severe loading situations that is encountered in the normal operating environment. The structural details of the truck-bolster/body-bolster connection are illustrated in Figure 2. The clearances are such that the side of the body bolster contacts the rim of the truck bolster before the clearances at the center pin are taken up. Thus, most of the force decelerating the truck under car impact is transferred between the side of the body bolster and the rim of the truck bolster. Some of the load can be transmitted by friction on the horizontal bearing surface of the center plate, but since this is a lubricated surface, only a small fraction of the load will be transmitted in this manner.

Force levels would be expected to be greater during the coupling of empty cars than when coupling loaded cars because the light car condition would result in a more rapid deceleration of the car. The BN indicated that in their unit train movement there was one point in the switching operations after the car had been unloaded where coupling at moderate speeds has been encountered.

1.2 Objectives

The program was limited to two objectives. The first objective was the experimental determination of interfacial loads between the truck bolsters and body bolsters on a car during its impact into standing cars. The effects of variations in car impact velocity and draft-gear energy-absorbing capacity on these loads were also to be examined. The second objective was a review of truck bolster center plate design. This review was to include an evaluation of the truck bolster center plate rim to carry the longitudinal impact loads and to provide recommendations for improvements in the design details of truck bolster construction, which would minimize the possibility of failure.

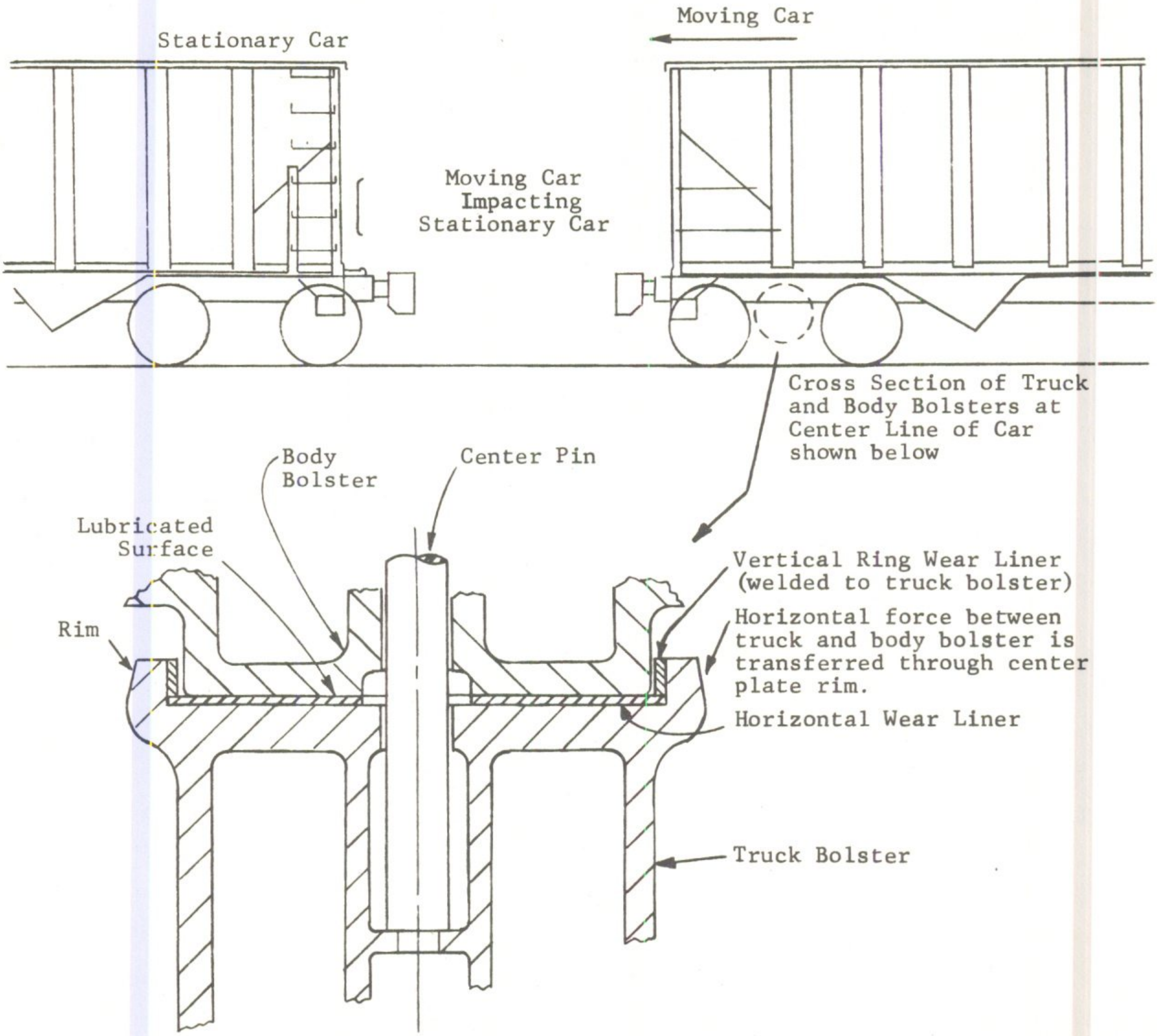


Figure 2. TRANSFER OF HORIZONTAL FORCE BETWEEN TRUCK AND BODY BOLSTERS THROUGH CENTER PLATE RIM UNDER CAR IMPACT CONDITIONS

2. TEST PROCEDURES

2.1 Test Plan

A test plan was formulated for measuring longitudinal* bolster forces under car impact conditions. A 100-ton capacity BN hopper car was made available for use as the hammer (striking) car. The test plan called for impact of this car in the unloaded condition and three standing cars each loaded to a 220,000 lb rail load. The hammer car was equipped with standard roller bearing trucks (6.5 by 12 inch journals). New truck bolsters were used conforming to AAR Specification M-202. The light weight of the car was 61,700 lbs. The body center plate was of standard design incorporating a 1:24 taper on the side surface, although this surface was worn to a near vertical orientation at the time of the tests.

Two different draft gear, representing different energy absorbing capacities, were used on the hammer car so that the effect of this variable on longitudinal bolster force could be evaluated. A Cardwell Westinghouse Mark 50 gear, which was standard equipment on the BN car, was used to determine effects with a higher capacity gear (AAR Specification M-901E). The measured capacity of this gear was 46,100 ft-lbs with a travel of 3.47 inches. A Miner A-22-XL gear was used to determine effects with a lower capacity draft gear (AAR Specification M-901). The measured capacity of this gear was 22,000 ft-lbs with a travel of 2.54 inches.**

The test plan called for beginning at an impact velocity of approximately 3 mph and increasing the speed in approximately 0.75 mph increments until damage occurred preventing further testing or until 1.25 million lbs was recorded as the maximum coupler force. Center plate surfaces were lubricated with grease before the tests.

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*The "longitudinal" direction is with respect to the car orientation. This direction is also referred to as "transverse" with respect to the bolster.

**A Miner A-22-XL gear was used in the struck car for all tests.

2.2 Instrumentation

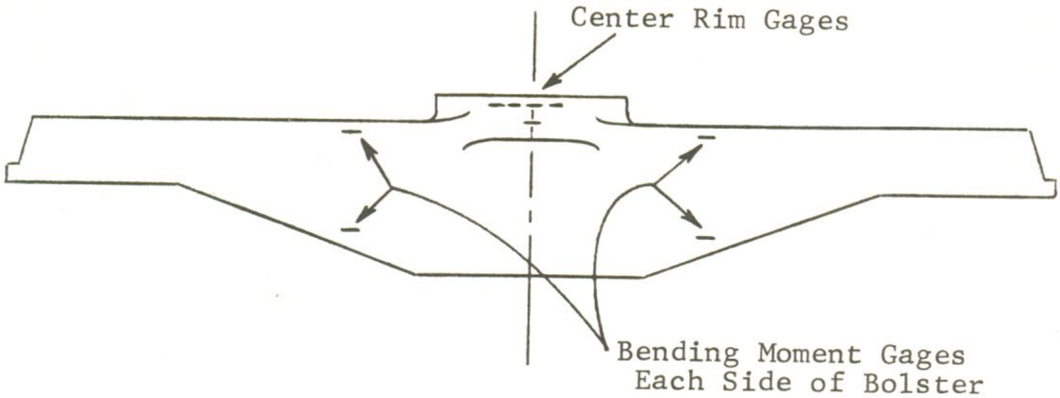
The primary instrumentation was strain gages applied to the bolsters of both the leading and trailing trucks of the hammer car. As shown in Figure 3, seven strain gages were placed in the vicinity of the center plate rim on the trailing side of the bolster. Four gages were mounted on each side of the bolsters for the measurements of the transverse bending moment, which would be caused by the longitudinal load. These gages were wired as two independent bridges on each bolster, one consisting of the top four gages and the other including the bottom four gages. They were calibrated before and after the tests by applying loads to the sides of the bolsters while they were supported at the friction shoe pockets.

The arrangement of the strain gages was designed to provide two independent measures of the total longitudinal load between the truck and body bolsters. The bending moment gages on the sides of the bolster were one means of measuring this load. The second means was the analysis of the distribution of strains measured by the gages on the trailing side of the center plate rim. A data reduction scheme was developed through the use of a finite-element analysis whereby both the magnitude of the load and its distribution over the inside face of the rim could be determined from the strains shown by these gages.

Other test parameters were monitored by additional instrumentation. A slide wire transducer was used to measure the draft gear travel. The coupler force was measured on the struck car. Motion picture cameras operating at 64 frames/second monitored the behavior of both trucks on the hammer car during each test. In addition, the AAR provided Fastex camera coverage (1,000 frames/second) of one of the trucks.

2.3 Test Operations

The tests were conducted at the Research and Development Division of Miner Enterprises, Inc., located in Chicago, Illinois. General views of the test track are shown in Figures 4 and 5. The three struck cars had their hand brakes set and a track skate was placed under one wheel of each car.



Detail of Center Plate Rim Gage Placement

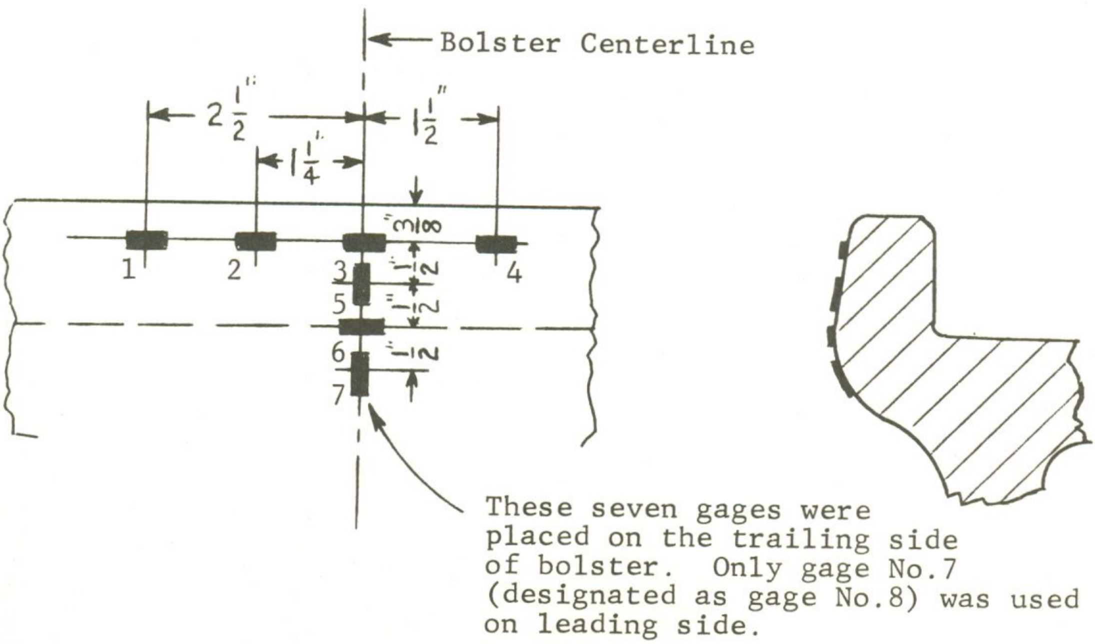


Figure 3. STRAIN GAGE POSITIONS ON SIDE OF BOLSTER

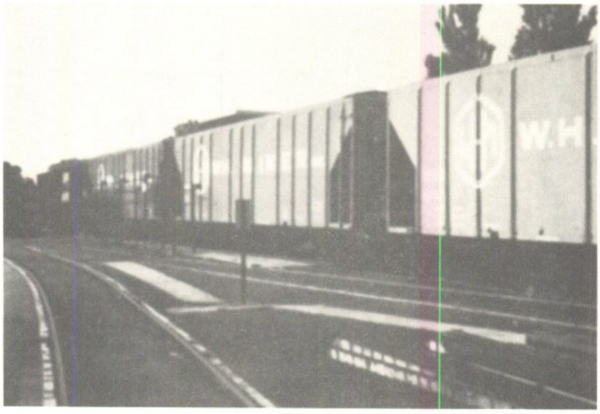


Figure 4. MINER TEST TRACK SHOWING BURLINGTON NORTHERN HAMMER CAR AND THREE STANDING CARS

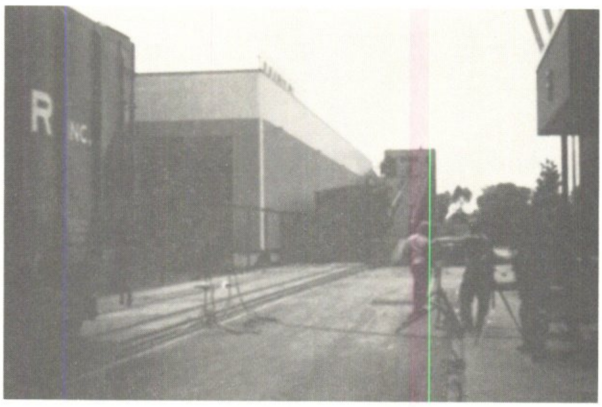


Figure 5. BURLINGTON NORTHERN HOPPER CAR BEING RAISED ON INCLINED TRACK

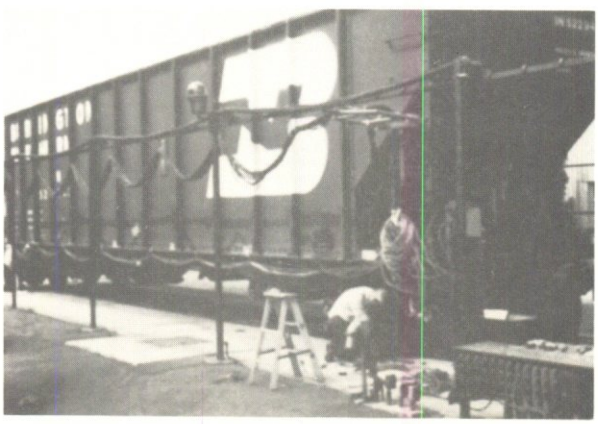


Figure 6. BURLINGTON NORTHERN HOPPER CAR SHOWING CONNECTION OF TRANSDUCER CABLES TO OVERHEAD CABLES SUSPENDED FROM TROLLEYS

The BN car was raised on the inclined track and released to provide for its motion into the standing cars. Prior to the placement of the standing cars, its speed was determined at various release positions on the inclined track so that the impact velocity could be controlled for each test.

Figure 6 shows a closer view of the hammer car, and illustrates the cable system for transmitting data from the impact car to the recorder. The cables from the transducers were brought to the end of the car and connected to cables suspended from trolleys. The trolleys run on an elevated rail along the test track.

Data were recorded on two oscillographs housed in an instrumentation room adjacent to the test track. The velocity was determined immediately before the point of impact by an electrically operated timing switch.

The tests were conducted on July 19, 1973. Table 1 summarizes the 17 tests. The first seven tests were made using the higher capacity draft gear. The tests began at 2.9 mph and were increased incrementally to a speed of 6.7 mph. Following the seventh test, this draft gear was removed and replaced by a lower capacity gear. Tests were then resumed at 3.1 mph, and progressed until a speed of 9.2 was reached on Test No. 17. The tests were suspended at that point because the coupler load limit was being approached and because both trucks slid out from under the body bolsters at that speed.

2.4 Test Results

2.4.1 Center Plate Rim Strain Gages

One of the unexpected results was the early measurement of permanent strains by the center plate rim gages. The increments of permanent strain indicated by these gages for each test are tabulated in the appendix. On the first test, at 2.9 mph, four of the gages on the lead truck bolster yielded. Additional yielding of the rim gages was indicated as the speed increased. After Test No. 5, it was decided to drop back in velocity and see whether or not the yielding would still be indicated. The supposition was that the rim was being work-hardened and that subjecting it to a lower velocity impact would not give an indication of yielding. Test No. 6 was run at a reduced velocity of 5.1 mph and the supposition was proved correct since there was no indication of additional permanent strain.

Substantial permanent strains were indicated on the next test, at 6.7 mph. The tests with the higher capacity draft gear were then suspended. It was feared that the strain gages might be damaged by a higher speed impact, and this would prevent obtaining data with the other draft gear.

TABLE 1. SUMMARY OF TEST CONDITIONS

Draft Gear in Hammer Car	Test Number	Impact Velocity (mph)	Coupler Force (lbs)
High Capacity Draft Gear (AAR Specification M-901E; measured capacity 46,100 ft-lbs)	1	2.90	220,000
	2	3.63	197,000
	3	4.25	233,000
	4	5.20	269,000
	5	5.99	294,000
	6	5.08	299,000
	7	6.65	284,000
Lower Capacity Draft Gear (AAR Specification M-901; measured capacity 22,000 ft-lbs)	8	3.14	180,000
	9	3.02	175,000
	10	3.58	238,000
	11	4.23	185,000
	12	5.08	260,000
	13	5.83	370,000
	14	6.82	690,000
	15	7.65	904,000
	16	8.41	1,020,000
	17	9.22	1,170,000

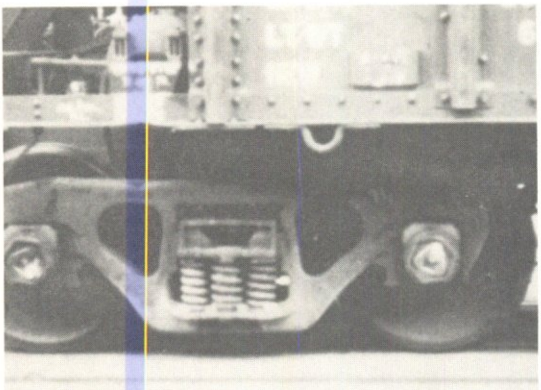
Impact tests were begun with the lower capacity draft gear at 3.1 mph. Permanent strains were indicated by four of the rim gages. (As explained later, a higher than average bolster load was measured on this test.) The test was repeated at 3.0 mph and no additional permanent strains were indicated. The tests continued with incremental increases in velocity. Significant permanent strains were not indicated until a speed of 6.8 mph was reached. This was the approximate impact velocity at which the tests with the higher capacity gear were suspended. It is likely that work-hardening of the rim during the earlier tests raised the yield strength of the steel so that yielding did not occur on the lower velocity tests with the lower capacity gear.

On Test Nos. 14 through 17 there were substantial permanent strains indicated with each impact. Several gages indicated incremental strains of 2,000 microinches/inch or more. Recognizing that the gages were strained beyond their normal operating range, there is still the indication of substantial permanent strains. Furthermore, it would be expected that strains on the inside of the rim near the fillet would be several times the indicated magnitude.

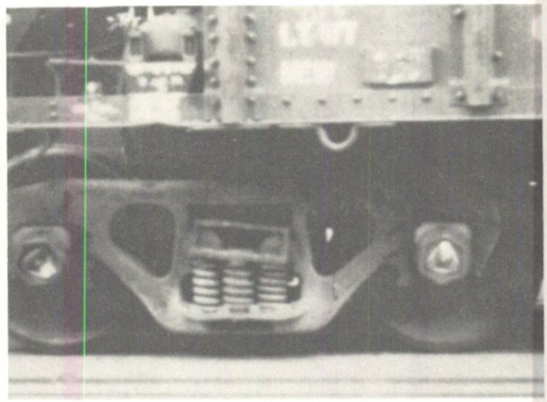
2.4.2 Bolster Rotation

It was anticipated that rotation of the bolster (with reference to an axis transverse to the car) would be observed at higher impact velocities. This rotation would be due to the force couple which results from the difference in elevations of the horizontal load on the truck bolster at the rim and its reaction at the friction shoe pockets. A consequence of this rotation is that it raises the line-of-action of the load on the center plate rim, which is undesirable from the standpoint of the rim resistance to this load.

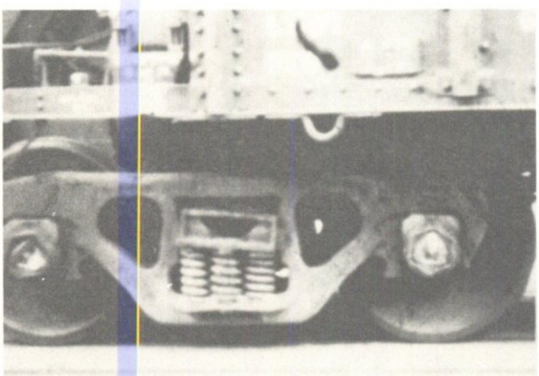
Bolster rotation was observed on Test Nos. 15, 16 and 17. Four frames from a motion picture sequence of the lead truck during the 7.7 mph impact on Test No. 15 are shown in Figure 7. The first frame, Figure 7(a), shows the bolster just before the impact; 0.04 second later, Figure 7(b), the bolster reaches its maximum tilted position and then returns to a less distorted position 0.08 second after the impact, Figure 7(c). This tilted position was held for an additional 0.11 second before the bolster returned to its normal horizontal position, Figure 7(d).



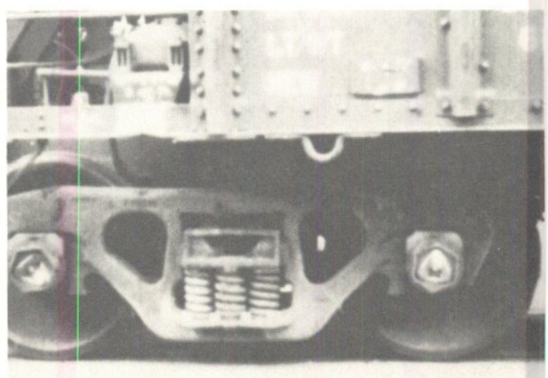
(a) At Beginning of Impact



(b) 0.04 second After Impact



(c) 0.08 second After Impact



(d) 0.19 second After Impact

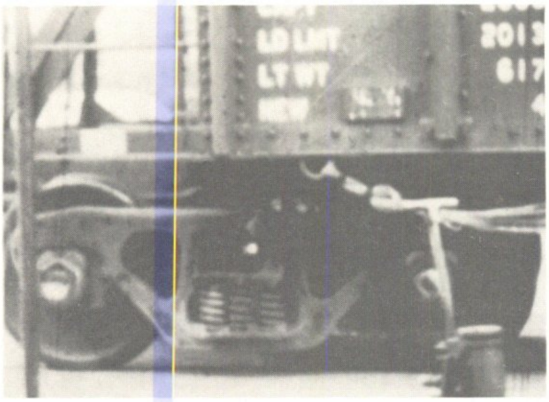
Figure 7. BOLSTER ROTATION ON LEAD TRUCK DURING TEST NO. 15 (7.7 mph impact speed, hammer car motion was to the left)

On Test No. 17, both bolsters were displaced from their normal position. Figure 8 shows a sequence of positions taken by the bolster on the trailing truck. Figure 8(a) shows the bolster just prior to impact. The second frame, Figure 8(b), shows the maximum tilted position, which was attained 0.03 second later. The impact caused a noticeable pitching motion of the car which raised the trailing end. This resulted in an extension of the support springs as shown in the third frame, Figure 8(c), where the maximum extended position is reached 0.12 second after the impact. Following this, the bolster returned to its normal height, but remained in the tilted position as shown in the final frame, Figure 8(d).

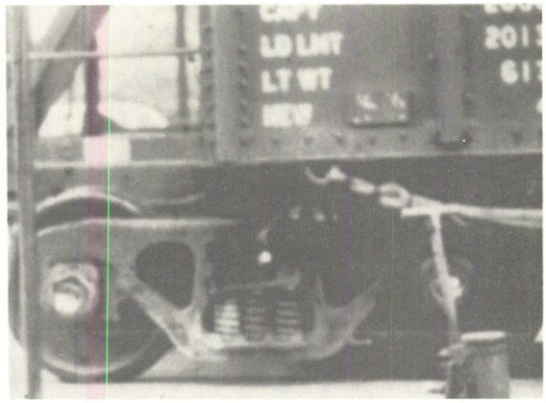
Figures 9, 10, and 11 show bolster orientation after the final test. Figure 9 shows the lead truck and Figure 10 shows the trailing truck where the rotation was greater. In both cases, the truck bolsters slid approximately 1 inch out from underneath the body bolsters and were being held in the tilted position by contact between the top of the rim and the lower surface of the body bolster. This is shown in Figure 11, a photograph taken from the end of the car showing the position of the trailing truck bolster. The two truck center pins were badly distorted, as shown in Figure 12.

2.4.3 Coupler Force and Draft Gear Travel

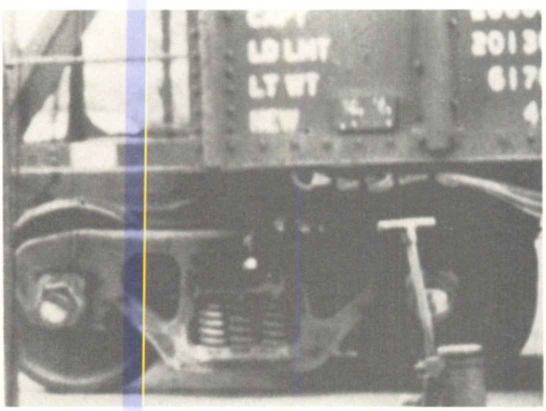
Figure 13 shows coupler force and draft gear travel as a function of impact velocity for both test series. The higher capacity draft gear used in the first seven tests did not go solid although it was close to the solid position (3.29 inches compared to a solid position of 3.47 inches) on the 6.7 mph impact. As a result, the coupler force measurements with this gear were maintained at an approximately constant level. The measured draft gear travel showed that the lower capacity gear used in Test Nos. 8 to 17 reached the nominal 2.54 inch solid position on Test No. 12 at 5.1 mph. As a result there was a rapid increase of coupler force for the higher speed impacts.



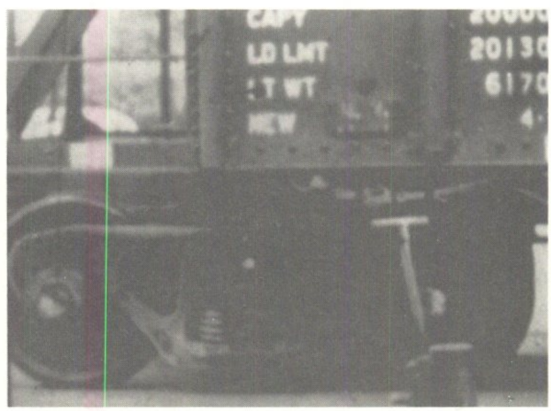
(a) At Beginning of Impact



(b) 0.03 second After Impact



(c) 0.12 second After Impact



(d) Final Position After Impact

Figure 8. BOLSTER ROTATION ON TRAILING TRUCK DURING TEST NO. 17 (9.2 mph impact speed, hammer car motion was to the right)

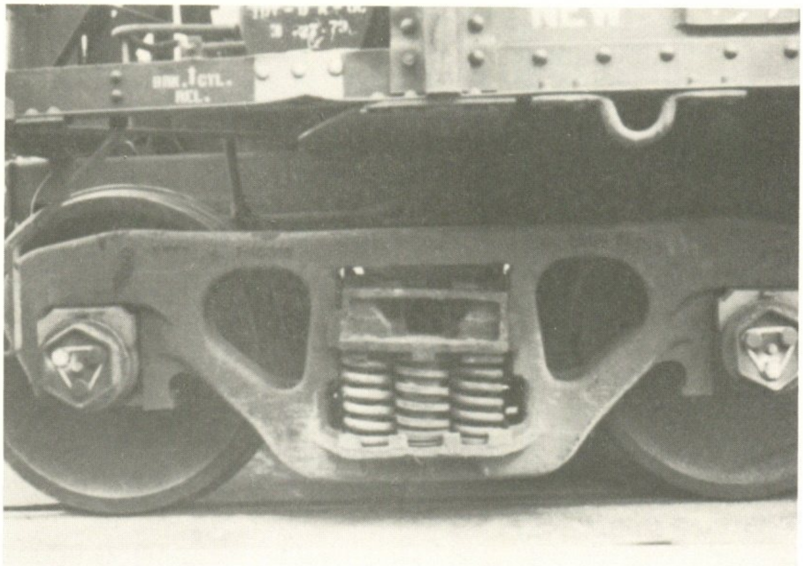


Figure 9. LEAD TRUCK FOLLOWING FINAL TEST

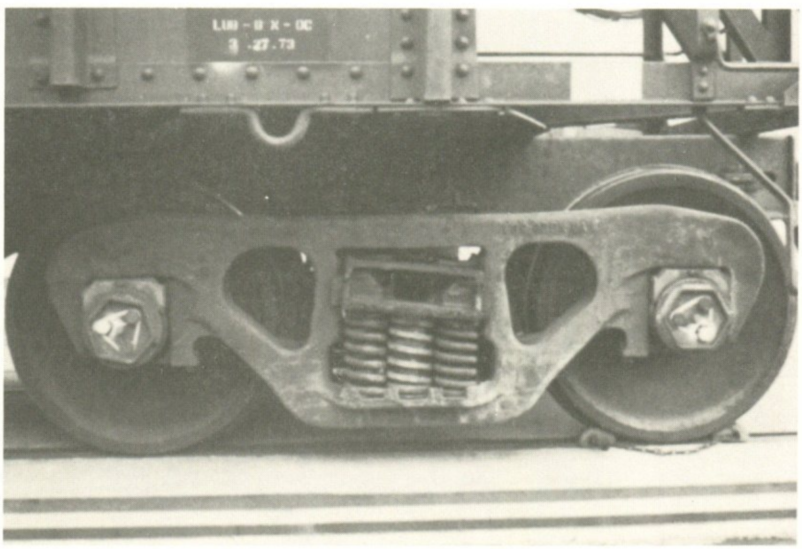


Figure 10. TRAILING TRUCK FOLLOWING FINAL TEST

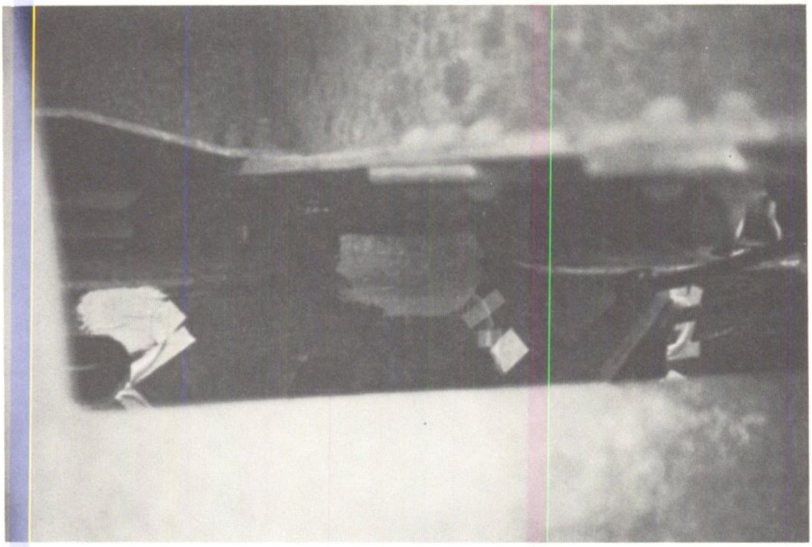


Figure 11. END VIEW OF TRAILING TRUCK FOLLOWING FINAL TEST SHOWING POSITION OF CENTER PLATE

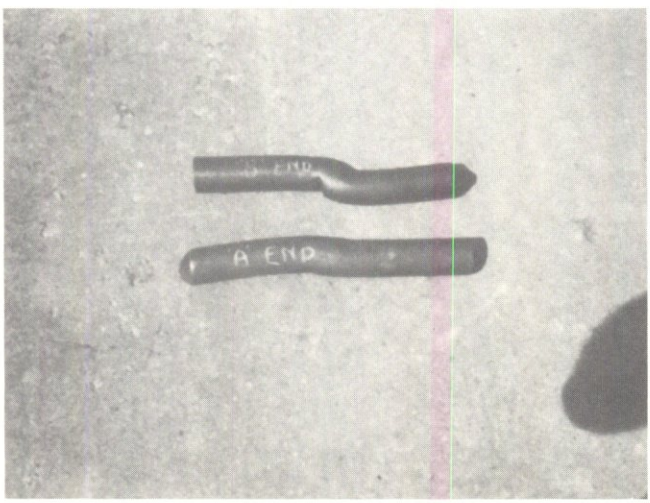


Figure 12. TRUCK CENTER PINS FOLLOWING TESTS (B end lead truck)

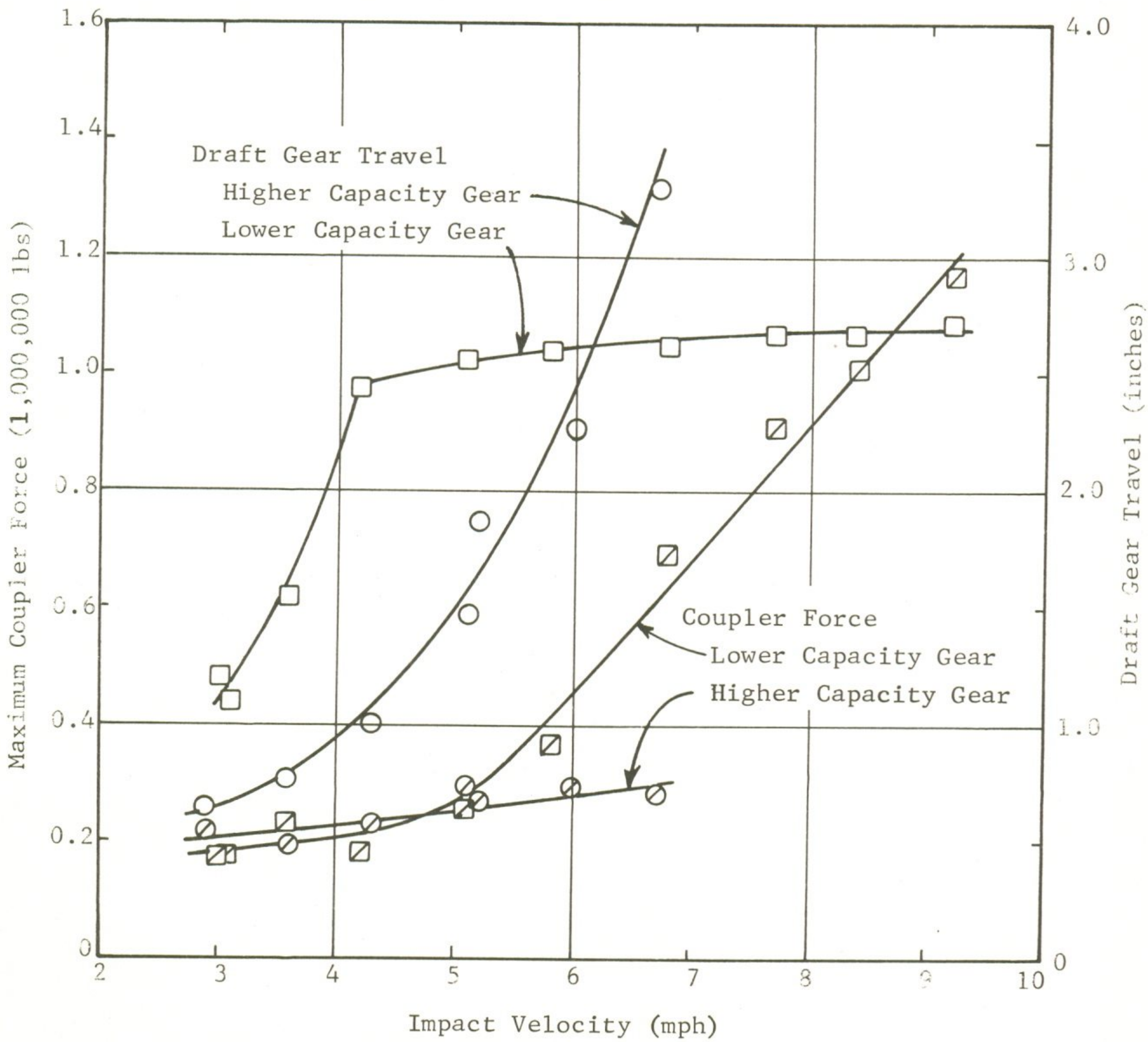


Figure 13. MAXIMUM COUPLER FORCE AND DRAFT GEAR TRAVEL AS A FUNCTION OF IMPACT VELOCITY

2.4.4 Truck Bolster Forces

Figures 14(a) and 14(b) summarize the maximum horizontal load obtained from reduction of the transverse bending moment strain gage data. These plots show that the load between the truck and body bolsters increases with increasing car impact velocity and that the rate of increase is greater above 6 mph. It also shows that, in every case except one, a higher load is indicated for the lead truck than for the trailing truck. Comparing Figures 14(a) and 14(b) with Figure 13 shows that the general trend of the bolster load data as a function of impact velocity can be correlated with the coupler force data.

The force data also show that there is no significant difference in the measured bolster load with the two types of draft gear at impact speeds up to 6 mph. However, comparison of the 6.7 mph impact of Test No. 7 with the 6.8 mph impact of Test No. 14 shows a 25 percent increase in load level for the second test. This difference may be due to the use of the lower-capacity draft gear.

The maximum horizontal force data are presented in Table 2. This table also presents the duration of the load as determined from the transient strain records. Bolster forces increase with increasing velocity although there is considerable scatter in the data. The scatter of the force data can often be correlated with the time over which the deceleration takes place. Note, for example, in comparing data between Test Nos. 8 and 9, where the impact velocities were approximately the same, that a higher bolster load is associated with a shorter deceleration period.

The data indicate that the load duration time varied from 0.04 to 0.10 second. The data also show that there is no distinct trend in load duration with impact velocity, truck position on the car, or type of draft gear used.

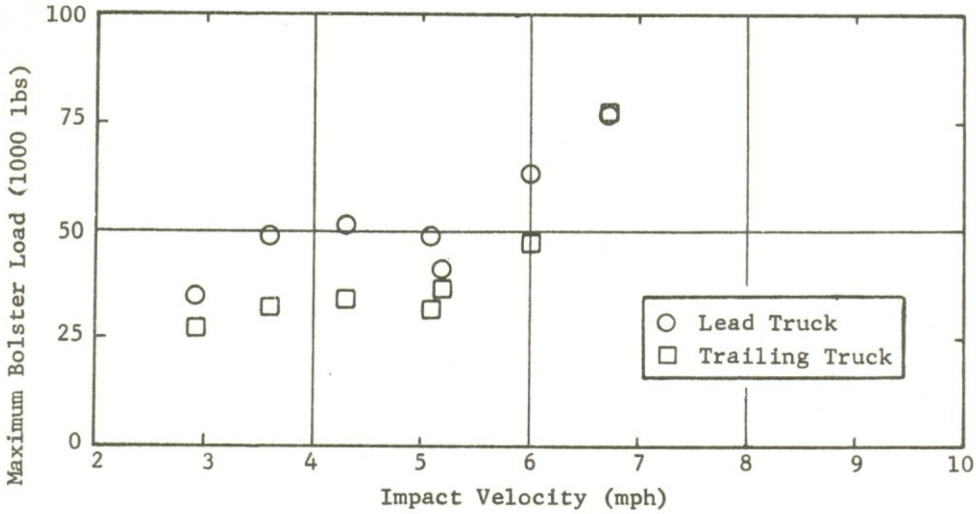


Figure 14a. MAXIMUM HORIZONTAL BOLSTER LOAD AS A FUNCTION OF CAR IMPACT VELOCITY, TESTS WITH HIGHER CAPACITY DRAFT GEAR

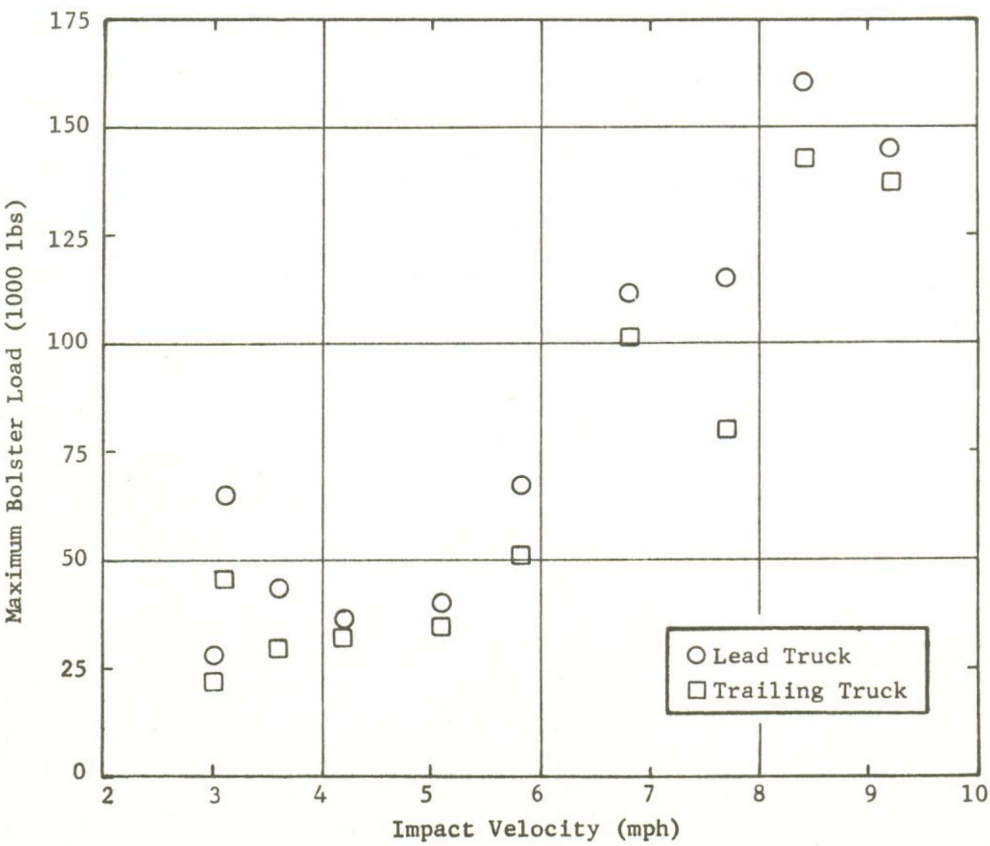


Figure 14b. MAXIMUM HORIZONTAL BOLSTER LOAD AS A FUNCTION OF CAR IMPACT VELOCITY, TESTS WITH LOWER CAPACITY DRAFT GEAR

TABLE 2. MAXIMUM HORIZONTAL BOLSTER LOAD AND DURATION OF LOAD APPLICATION (TRUCK DECELERATION PERIOD)

Draft Gear	Test No.	Impact Speed (mph)	Leading Truck		Trailing Truck	
			Maximum Bolster Load (lbs)	Load Duration (Deceleration Period) (sec)	Maximum Bolster Load (lbs)	Load Duration (Deceleration Period) (sec)
Higher Capacity Gear (AAR Spec. M-901E)	1	2.9	35,000	0.085	27,000	0.055
	2	3.6	48,000	0.080	32,000	0.070
	3	4.3	50,700	0.085	34,000	0.085
	4	5.2	41,500	0.100	37,000	0.100
	5	6.0	62,900	0.070	47,000	0.070
	6	5.1	49,000	0.090	32,000	0.090
	7	6.7	77,500	0.090	78,000	0.080
Lower Capacity Gear (AAR Spec. M-901)	8	3.1	65,000	0.040	45,500	0.040
	9	3.0	28,500	0.075	22,000	0.090
	10	3.6	43,100	0.070	30,000	0.080
	11	4.2	36,200	0.090	32,600	0.080
	12	5.1	40,300	0.100	34,500	0.100
	13	5.8	67,500	0.100	51,000	0.100
	14	6.8	112,000	0.070	101,000	0.055
	15	7.7	115,000	0.070	80,000	0.040
	16	8.4	160,000	0.070	142,000	0.100
	17	9.2	145,000	0.080	138,000	0.070

The yielding of the center rim strain gages on the first test (Test No. 8, 3.1 mph) using the lower capacity draft gear was described previously. The data in Table 2 show that the horizontal bolster load measured during this test was significantly higher than that measured on the other tests conducted in this velocity range, which probably accounts for the high strain levels. Note also that the deceleration period for this impact was the shortest measured on any of the tests.

An examination of the longitudinal bolster force-time records shows some variety in the load patterns. Most of the records show a high degree of correlation with the coupler force-time record as illustrated in Figure 15. The two distinct force pulses within the deceleration period are typical of the data, although there is some variation in the rise time and duration from one record to another.

The character of the force-time record is probably affected by the orientation of the truck components at the time of impact. There are numerous points of clearance between truck components, such as between the journal bearing adapters and the side frame pedestals, the bolster and the side frame columns, and in the truck center plate itself. The relative orientation of the components within these clearance limits existing at the time of impact would affect the character of the force-time record. This would also account for the scatter in the load data.

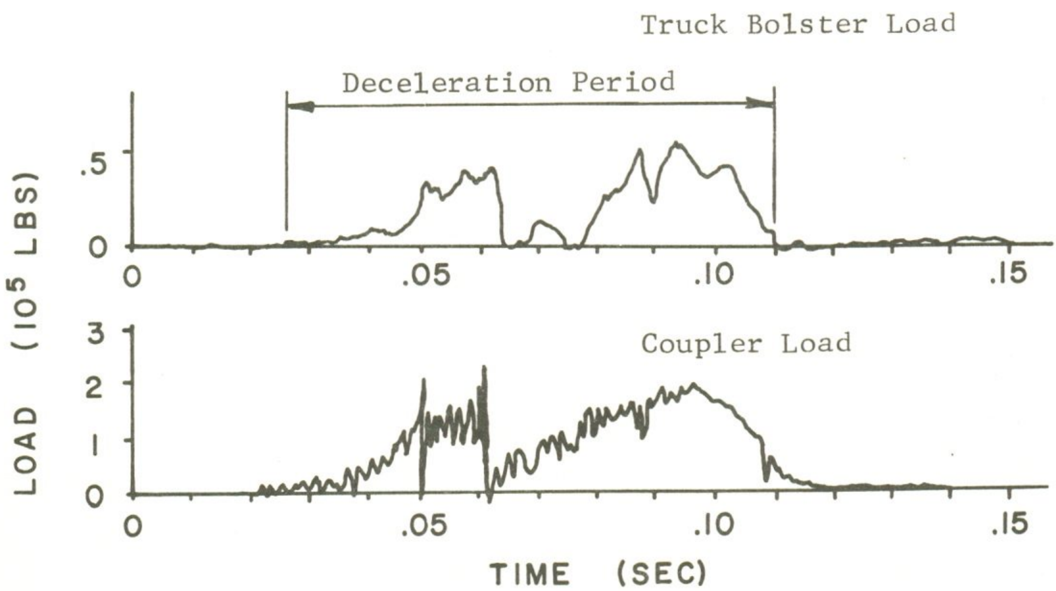


Figure 15. LONGITUDINAL TRUCK BOLSTER (LEAD TRUCK) AND COUPLER FORCE-TIME RECORDS, TEST NO. 3, 4.3 MPH

3. ANALYSIS

3.1 Reduction of Data from Center Rim Gages

The test plan called for the use of strain data from the gages around the center plate rim to provide an independent evaluation of the load between the body and truck bolsters. These data were also to be used to define elevation of the line-of-action of the load on the rim and its angular distribution. In general, it was not possible to fulfill these objectives because of the high strain levels recorded. The finite-element analysis on which the data reduction scheme was based depended on the linear behavior of the material throughout the region of the center plate rim. The analysis was applied to data from two of the lower speed tests (Test Nos. 9 and 10), where none of the strain gages indicated yielding, and the results showed that the most probable distribution of the load includes an area within the top 0.5 inch of the rim spread out over an angle from 15 to 30 degrees.

3.2 Evaluation of Center Plate Rim Design

The permanent strains recorded by the center plate rim gages are indicative of the marginal performance of the rim in carrying the horizontal impact loads. In order to evaluate this feature of truck bolster design, an elastic finite-element analysis was used to determine stresses in the vicinity of the rim caused by a horizontal load. The analysis was based on an axisymmetric representation of the center plate region of the bolster subjected to load distributions which vary in the circumferential direction.* A typical grid of finite elements and the definition of coordinates are shown in Figure 16. Provision was also made for variation in the vertical load distribution on the rim so that the area on the rim over which the load is distributed is a parameter that may be selected in the analysis.

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*The analysis is of the type described in Zienkiewicz, O. C., "The Finite Element Method in Engineering Science", Chapter 13, McGraw-Hill Publishing Co., 1971.

The horizontal and vertical wear liners were not considered in the analysis so that the structural representation is limited to the truck bolster casting itself. (Wear liners were installed in the normal manner on the trucks of the hammer car during the impact tests since this is the standard configuration.)

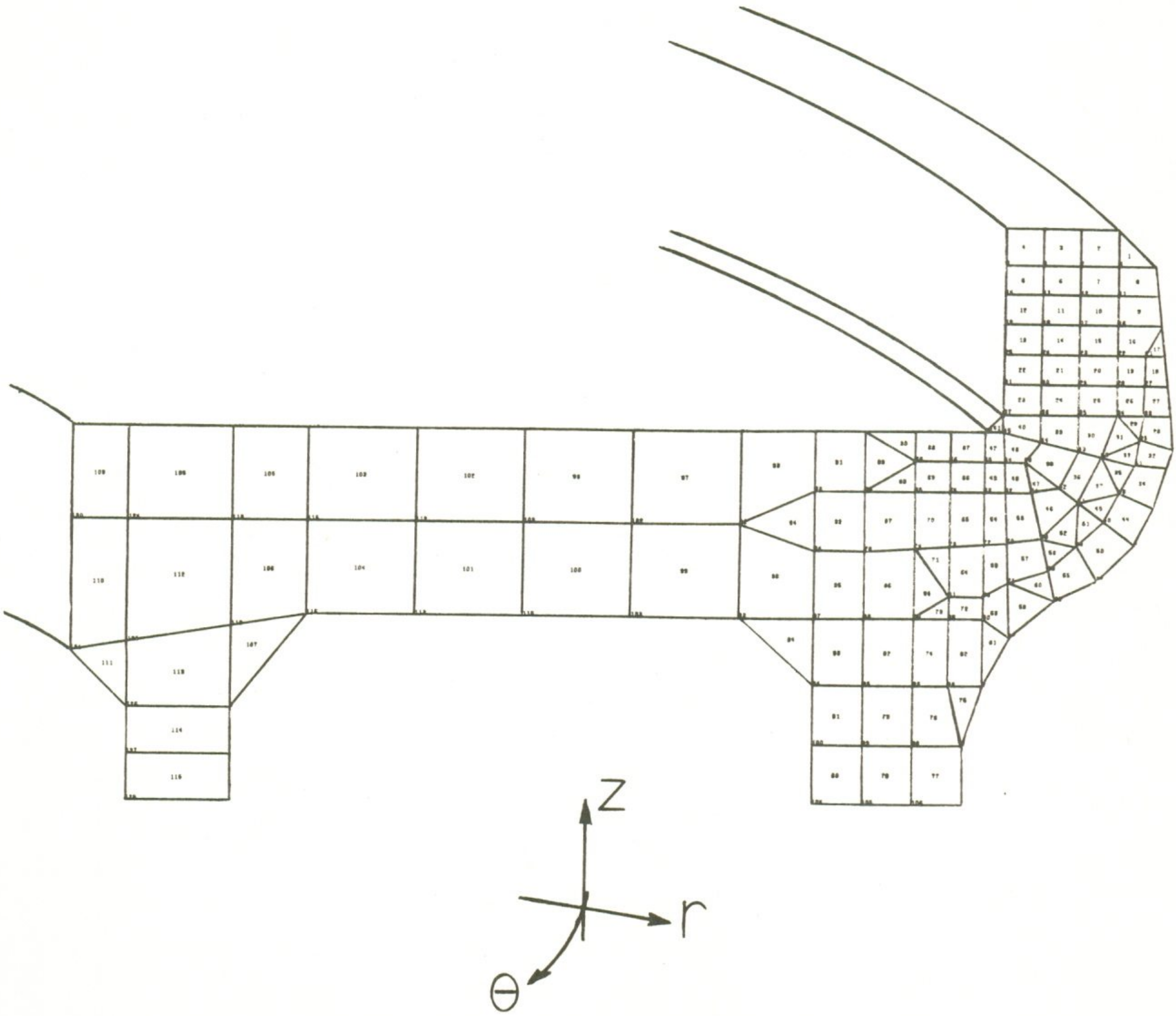

Figure 16. FINITE ELEMENT GRID USED IN THE ANALYSIS OF STANDARD TRUCK BOLSTER CENTER PLATE DESIGN

Figure 17 shows the results from an analysis of a standard truck bolster configuration. A total horizontal load of 100,000 lbs was assumed acting on the center plate rim. It was also assumed that the load was uniformly distributed over the full height and over an arc of 20 degrees on the rim. The octahedral shear stresses and the maximum tensile principal stresses are shown at five locations on a cross section at the center. The stress values are for elements adjacent to the surface. The octahedral shear stress is a measure of the tendency of the material to yield, and the maximum tensile principal stress is an indicator of the tendency of the material to fracture. (No principal stress is shown if all principal stresses are compressive.) At the central cross section of the bolster illustrated in the figure, the maximum principal stress would either be in the transverse plane shown in the figure (designated "rz") or perpendicular to this plane (circumferential direction, designated "θ"). This is indicated by the notation in parentheses following each tabulated value of principal stress.

The truck bolsters used in the tests were made of grade B steel. This material has a minimum yield strength of 38,000 psi and a minimum tensile strength of 70,000 psi. Yielding of the steel would be expected when the octahedral shear stress exceeds approximately 18,000 psi. It is apparent from the values on the figure that yielding would be expected within the center plate rim from the 100,000 lb load. Since the analysis is linear, the stress predictions would be proportional to the total load. The most severe stresses are located in the fillet between the horizontal center plate surface and the inside of the rim. At this position, the octahedral shear stress is over four times the expected yield strength. Yielding at this location would cause a redistribution of stresses which would increase the stresses in other portions of the rim. The observation of permanent strains on the rim gages, even at the lower range of impact loads measured (e.g., 30,000 lbs) is not unexpected on the basis of this analysis.

The analysis is somewhat conservative as the load is assumed to be distributed over a rather large area. If the angular distribution were made narrower or if the load were restricted to the upper

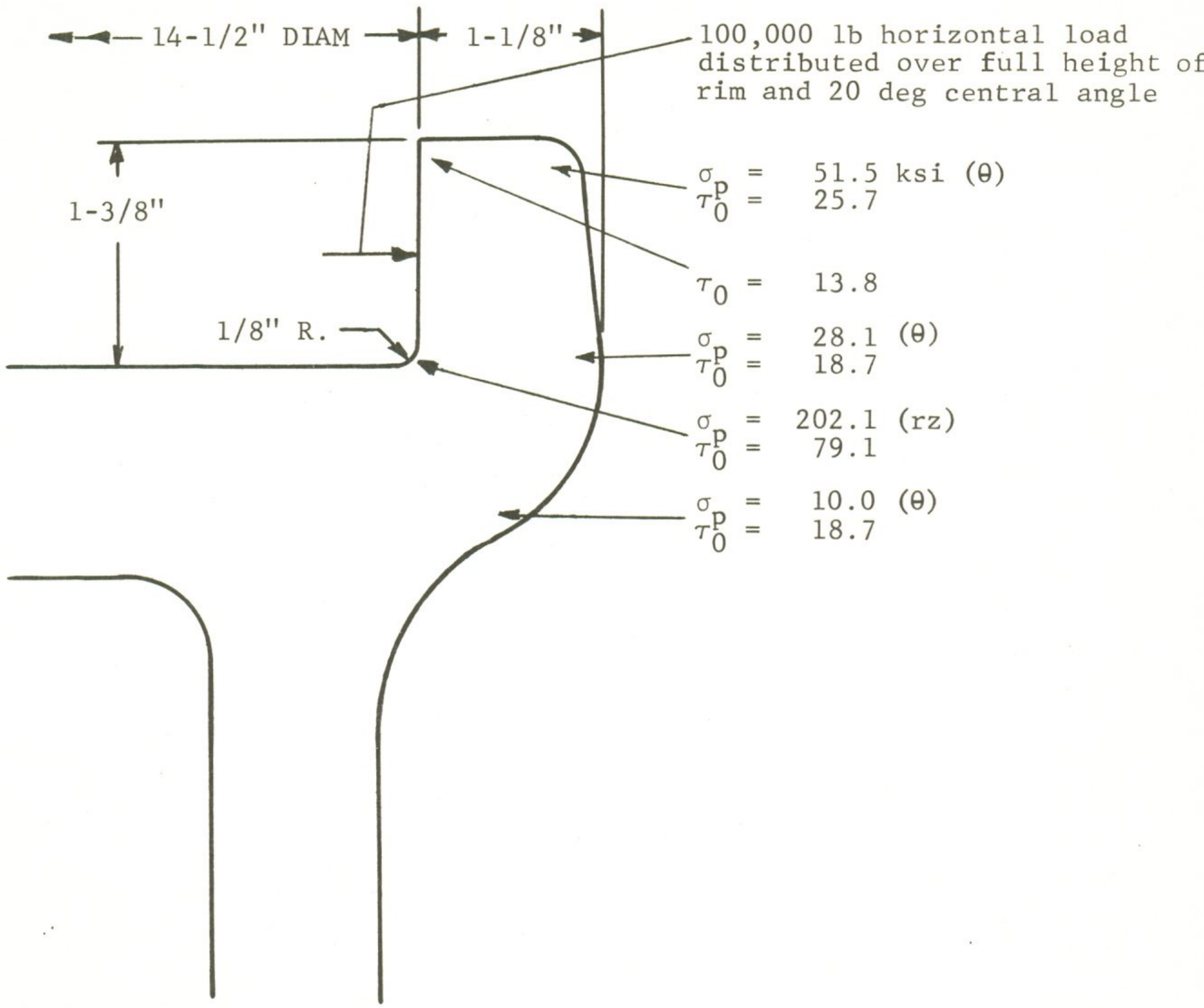

Figure 17. MAXIMUM PRINCIPAL (σ_p) AND OCTAHEDRAL SHEAR (τ_0) STRESSES AT CENTER OF TRUCK BOLSTER FOR STANDARD BOLSTER CENTER PLATE DESIGN

portion of the rim, the stresses would be increased. For example, if the load were applied to only the upper half of the rim, and over the 20 degree arc, the octahedral and maximum principal stresses at the inside fillet would be increased 12 and 17 percent respectively. On the other hand, if the load, applied over the full height of the rim, were spread out over an angle of 30 degrees instead of 20 degrees, the octahedral and maximum principal stresses at the inside fillet would be reduced 28 and 26 percent respectively.

The finite-element analysis was repeated for three modified center plate rim designs. This was done to evaluate the effectiveness of various types of design modifications, which were considered one at a time, on stresses within the rim. These modifications and the resulting stress levels for the same magnitude and distribution of load as shown in Figure 17 are shown in Figures 18, 19, and 20.

Figure 18 shows the stresses on a design where the nominal 14 inch center plate diameter has been increased to 16 inches. Maximum octahedral shear stress at the critical fillet location is reduced 4 percent to a value of 75.9 ksi. Other changes in the rim stresses are of a minor nature.

Figure 19 shows the changes from a 0.25 inch increase in the width of the rim. This results in a 21 percent reduction in the octahedral shear stress at the inside fillet location and a 22 percent reduction in the maximum principal stress. The octahedral shear stresses on the outside of the rim are also reduced approximately 20 percent.

Figure 20 shows the results from an increase in the radius of the fillet between the horizontal center plate surface and the inside rim from 0.125 inch to 0.5 inch. This results in a 28 percent reduction in the octahedral shear stress near the fillet. Stresses in other regions of the rim are not significantly altered by this modification. On the basis of these results, it would appear that a desirable design modification would involve increases in the radius of the inside fillet and width of the rim.

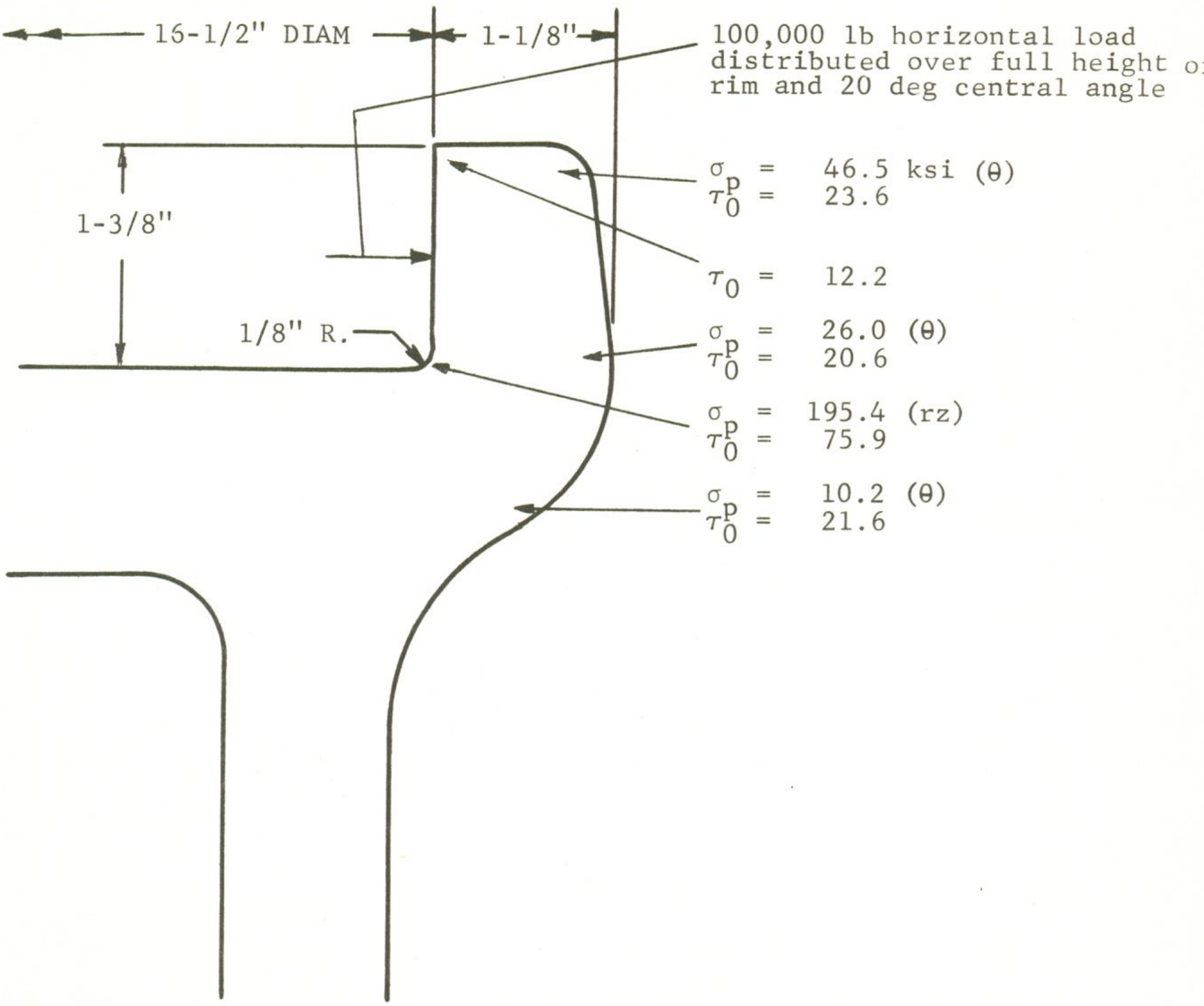

Figure 18. MAXIMUM PRINCIPAL (σ_p) AND OCTAHEDRAL SHEAR (τ_0) STRESSES AT CENTER OF TRUCK BOLSTER FOR 16 INCH CENTER PLATE DIAMETER

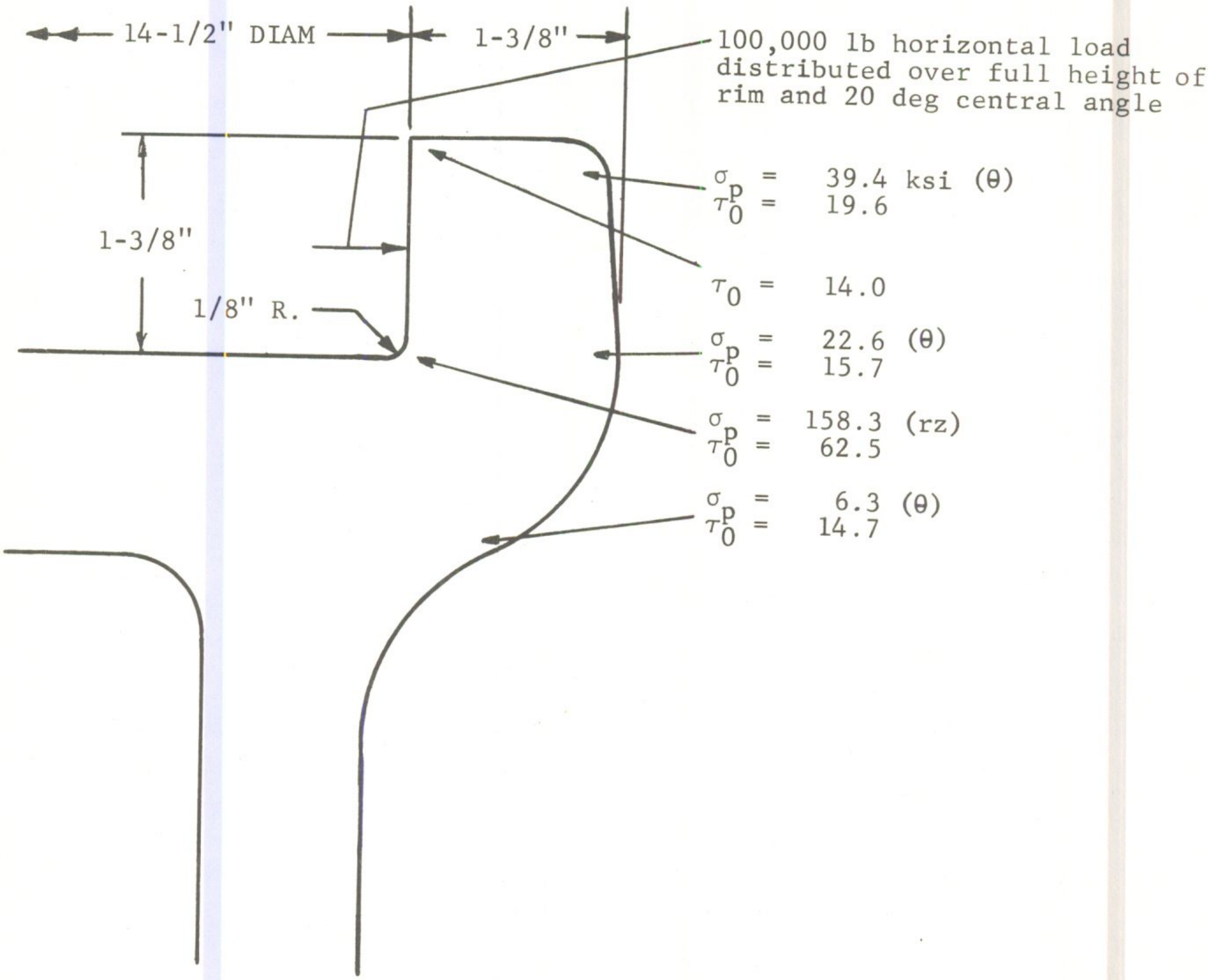

Figure 19. MAXIMUM PRINCIPAL (σ_p) AND OCTAHEDRAL SHEAR (τ_0) STRESSES AT CENTER OF TRUCK BOLSTER FOR 1/4 INCH WIDER CENTER PLATE RIM

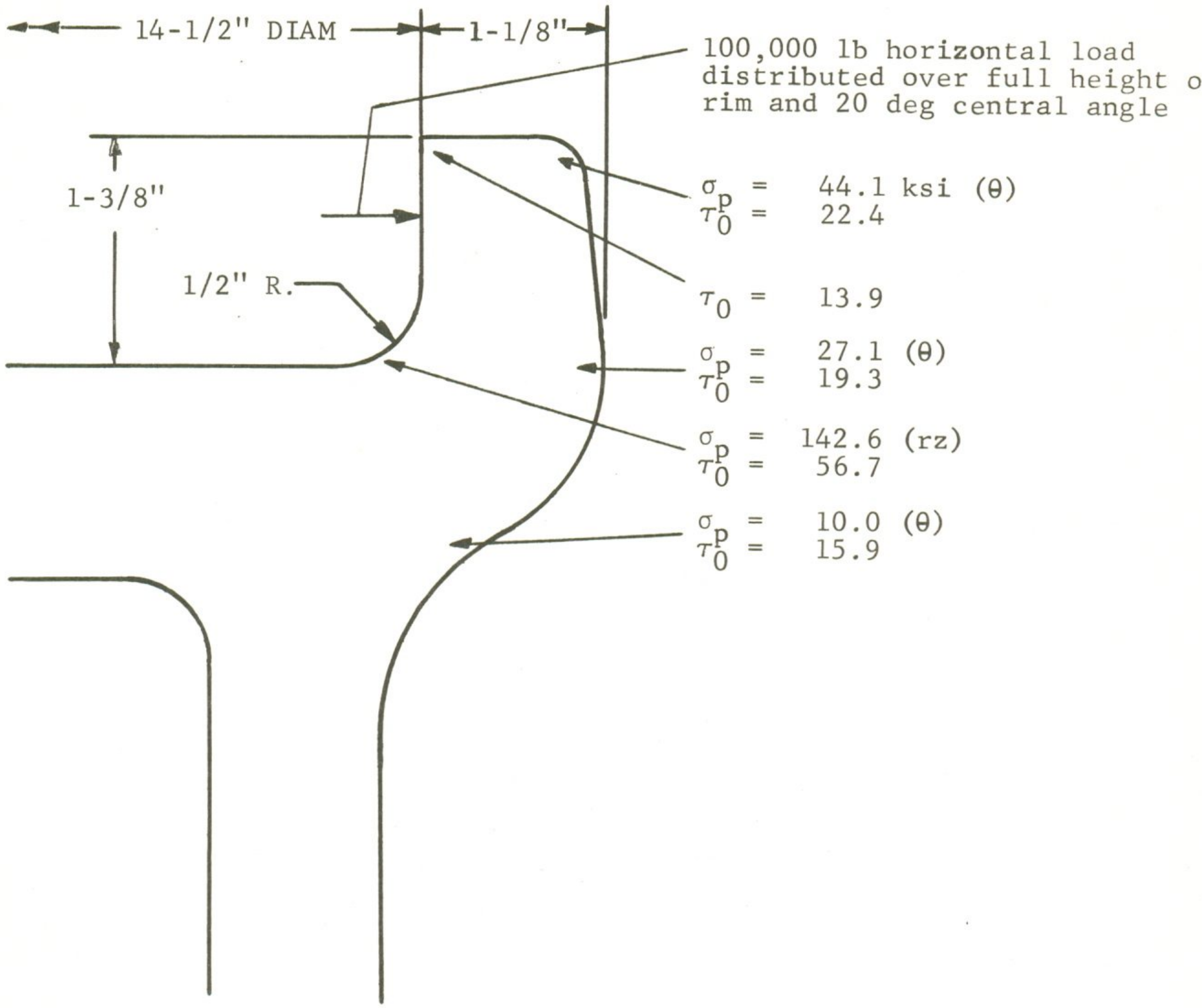

Figure 20. MAXIMUM PRINCIPAL (σ_p) AND OCTAHEDRAL SHEAR (τ_0) STRESSES AT CENTER OF TRUCK BOLSTER FOR 1/2 INCH RADIUS BETWEEN CENTER PLATE AND RIM

4. CONCLUSIONS

Tests were successfully completed determining the longitudinal load between truck and body bolsters on a car impacting standing cars. This load results from the rapid deceleration of the truck. Substantial loads were measured averaging a maximum of 40,000 lbs for velocities up to 5 mph, and reaching approximately 100,000 lbs at 7 mph. The magnitude of these loads is influenced by the rather severe impact conditions utilized in the tests, namely, an unloaded hammer car, loaded struck cars with the handbrakes set and track skates behind several of the wheels, and the use of a lower-capacity draft gear on the higher-speed tests. Nevertheless, the large load measurements and the yielding indicated by the center rim strain gages, even under low velocity impacts, shows that the standard truck bolster center plate rim design has only a marginal adequacy to carry impact loads. The analysis of the rim showed that stresses within the rim due to longitudinal loads can be substantially reduced by increasing the thickness of the rim and the radius of the fillet at the base of the inside surface of the rim.

The thickness and height of the rim used on truck bolsters for 100-ton cars is the same that has been used for many years on 55-ton capacity cars (although the diameter of the center plate is 2 inches larger). The heavier trucks used on 100-ton cars would result in higher inertial loads than for the lighter trucks used on lower capacity cars. While the center plate rim design may have been adequate for the lower capacity cars, the safety margin of this design detail is significantly reduced for the higher capacity cars. This correlates with the higher failure rates experienced on 100-ton cars.

The observation that a significant number of truck bolster center plate rim failures occur after they have received several hundred major impacts from couplings in the light car condition can be correlated with the test results. The progressive yielding of the material in the rim, which becomes especially prominent for longitudinal bolster loads above approximately 75,000 lbs, indicates that this region is susceptible to low-cycle fatigue failure phenomena. A rough rule of thumb sometimes used is that $\pm$ 1 percent

strain (2 percent range) causes failure in about 1,000 cycles.*
Larger strains would, of course, result in earlier failure, and lower strains would give longer life. The residual strain gage data suggests that strains exceeding 1 percent may be expected under moderate to high velocity impact conditions. Thus, one would anticipate a high probability of failure after several hundred impacts.

One of the characteristics of truck bolster performance at higher velocity impacts is its tendency to rotate. This was observed on the tests at impact velocities of 7.7 mph and above. Bolster rotation results from the difference in elevation of the line-of-action of the force between the body and truck bolsters and the reaction to this load from the side frame columns through the friction shoe pockets. An undesirable effect of bolster rotation is the raising of the line-of-action of the load on the rim, which increases the stresses within the rim. Another undesirable effect is the possibility of disengagement of the body center plate from the rim resulting in the transfer of the load to the center pin, which is a weaker member for carrying this load.

The following specific recommendations, which will lead to a reduced probability of truck bolster center plate rim damage, can be presented as a result of the work performed on this project:

- Redesign the truck bolster center plate to increase the thickness of the rim and the radius of the fillet between the horizontal center plate surface and the inside surface of the rim.
- Reduce the difference in elevation between the center plate rim and the friction shoe pockets on the truck bolster bringing these two longitudinal load paths closer together so that there is less of a tendency for the bolster to rotate under car impact conditions.
- Utilize long-travel high-capacity draft gear and other cushioning devices so that the period of truck deceleration under car impact conditions is spread out over as long a period of time as possible.

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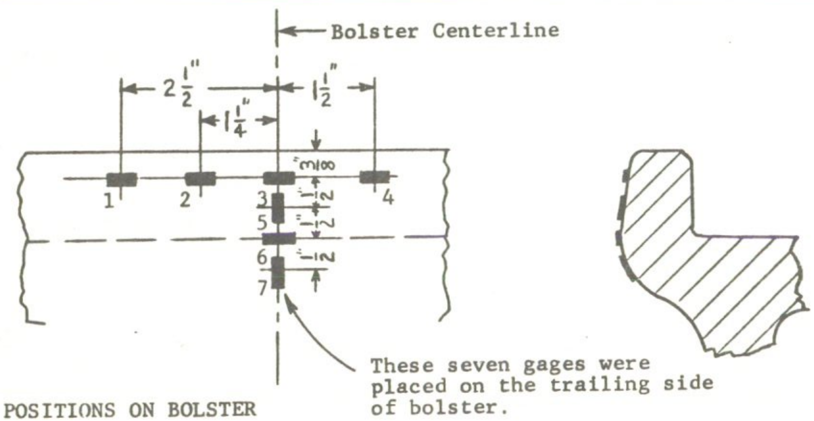
*Morrow, J.; Johnson, T. A.; "Correlation Between Cyclic Strain Range and Low-Cycle Fatigue of Metals," Materials Research and Standards 5(1), p 30 (January 1965)

APPENDIX A: PERMANENT STRAIN INCREMENTS INDICATED BY CENTER PLATE RIM STRAIN GAGES

Draft Gear	Test No.	Impact Speed (mph)	Leading Truck: Maximum Bolster Load (lbs)	Leading Truck: Indicated Permanent Strain (microinches/inch), Gage No. 1	2	3	4	5	6	7	Trailing Truck: Maximum Bolster Load (lbs)	Trailing Truck: Indicated Permanent Strain (microinches/inch), Gage No. 1	2	3	4	5	6	7
Higher Capacity	1	2.9	35,000	-	300	710	-	-	100	100	27,000	-	-	-	-	-	-	-
	2	3.6	48,000	-	230	900	-	300	160	240	32,000	-	-	80	-	-	-	-
	3	4.3	50,700	-	60	160	50	110	120	90	34,000	-	-	-	-	-	-	-
	4	5.2	41,500	-	-	-	-	-	-	-	37,000	-	-	-	-	-	-	-
	5	6.0	62,900	-	500	850	280	660	480	440	47,000	-	50	110	340	140	-	40
	6	5.1	49,000	-	-	-	-	-	-	-	32,000	-	-	-	-	-	-	-
	7	6.7	77,500	-	140	500	420	260	190	280	78,000	-	190	150	200	100	-	40
Lower Capacity	8	3.1	65,000	-	70	250	240	160	-	170	45,500	-	-	-	-	-	-	-
	9	3.0	28,500	-	-	-	-	-	-	-	22,000	-	-	-	-	-	-	-
	10	3.6	43,100	-	-	-	-	-	-	-	30,000	-	-	-	-	-	-	-
	11	4.2	36,200	-	-	-	-	-	-	-	32,600	-	-	-	-	-	-	-
	12	5.1	40,300	-	-	-	-	95	-	120	34,500	-	-	-	-	-	-	-
	13	5.8	67,500	-	70	-	-	-	-	-	51,000	-	-	-	-	-	-	-
	14	6.8	112,000	700	2000	1600	1000	1800	700	1700	101,000	1100	2100	870	1300	730	1200	520
	15	7.7	115,000	290	500	630	120	470	130	240	80,000	790	470	210	300	300	270	*
	16	8.4	160,000	910	780	1000	950	1200	250	930	142,000	690	750	1100	1300	190	460	2600
	17	9.2	145,000	580	290	300	330	730	120	510	138,000	670	430	540	420	430	430	570

*Undetermined



STRAIN GAGE POSITIONS ON BOLSTER

APPENDIX B: REPORT OF INVENTIONS

A diligent review of the work performed under this contract has revealed no new innovation, discovery, improvement, or invention.