Investigation of Multiple Tank Car Rollover Derailments Related to Double Shelf Couplers and its Solutions

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Transport Dangerous Goods Directorate
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Investigation of Multiple Tank Car Rollover Derailments Related to Double Shelf Couplers and its Solutions

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This report reflects the views of the performing organisation, and not necessarily those of the Transport Dangerous Goods Directorate of Transport Canada.

Since some of the accepted measures in the industry are imperial, metric measures are not always used in this report.

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16. Abstract

When transporting dangerous goods, the importance of safety cannot be understated. One particular aspect of safety in relation to the transportation of dangerous goods by rail using tank cars is the prevention of spills of dangerous goods. The introduction of double shelf couplers has reduced incidents of tank car puncture during derailments. However, another notable effect has been an increase in the number of cars involved in each derailment involving double shelf couplers. Often a few derailing tank cars are able to cause additional tank cars to derail and roll over as well.

After reviewing derailment reports and photographs, two mechanisms have been identified that likely work in combination to produce multiple tank car derailments. Based on these mechanisms, several solutions are also suggested that may prevent or at least minimize the extent of multiple tank car rollover derailments. These include rotary couplers, increased shelf heights and the use of locking centre pins. An economic analysis of two of the proposed solutions is presented. In addition, recommendations for future work are made.

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EXECUTIVE SUMMARY

When transporting dangerous goods, the importance of safety cannot be understated. One particular aspect of safety in relation to the transportation of dangerous goods by rail using tank cars is the prevention of spills of dangerous commodities. Any loss of dangerous goods cargo can endanger the health and safety of the public, railroad employees, property and have devastating effects on the environment.

The safety of tank cars underwent significant improvement after several accidents in the 1960's and 1970's involving tank cars carrying dangerous goods. These accidents prompted several design changes to improve safety. One of these design changes was the introduction of head shields to make the ends of the tank less prone to puncture. The second change was the introduction of double shelf couplers. Double shelf couplers equipped with bottom and top shelves are designed to prevent decoupling during derailment events. The use of double shelf couplers effectively prevents adjoining couplers from disengaging vertically from each other. If the couplers are able to disengage vertically there is a risk of coupler override which can lead to the puncture of the tank shell and the release of dangerous goods.

The introduction of double shelf couplers and head shields has reduced incidents of tank car puncture during derailments. However, the designed feature of double shelf couplers leads them to remain coupled during a derailment with the notable effect being an increase in the size of derailment events involving double shelf couplers. Often a few derailing tank cars are able to cause additional tank cars to derail and roll over as well. The progressive rollover derailment can be compared to dominos. Empty tank cars are particularly susceptible to this type of occurrence.

Centre for Surface Transportation Technology (CSTT) was contracted by Transport Canada (TC) to investigate multiple tank car derailments with the goal of identifying potential modifications to minimize the extent of derailments involving double shelf couplers. TC's interest in this issue is motivated in part by a number of incidents in Canada that involved unit trains.

The purpose of this study is:

- To gather relevant information regarding tank car derailments and shelf couplers.
- To understand the mechanisms involved in the propagation of multiple tank car rollovers.
- To suggest and examine potential remedies to this problem, including the economic impact of proposed solutions.

Data from various sources, including derailment reports were examined in this study. In order to provide solutions to reduce the risk of multiple tank car rollovers the report aims to understand the mechanisms involved in multiple tank car rollover derailments.

After reviewing derailment reports and photographs, two mechanisms have been identified that likely work in combination. The first mechanism investigated is rollover due to a moment transferred between couplers. The second mechanism is dominated by the transfer of vertical motion from a car that is derailing to the adjacent car that is still on the rails.

Rollover resistance calculations were performed for two cases: roller side bearings and long travel constant contact side bearings. The results for each case are presented. It is evident that the empty tank car body has a very large self-stabilizing moment once the body weight is carried on the side bearings. The peak rollover moment resistance values were 1,378,000 in-lbs and 1,324,000 in-lbs for an empty tank car with roller side bearings and long travel constant contact side bearings, respectively.

In addition to the large external moment required to roll an empty tank car body off its trucks, further evidence exists to suggest that an additional mechanism must also be involved to create the resulting roll over condition. The mechanism that causes overturning of the car immediately adjacent to the initiating car is thought to have a significant vertical component of motion imposed upon it. This idea is supported by several key pieces of evidence from derailment photos.

Based on these mechanisms, several solutions are suggested that may prevent or at least minimize the extent of multiple tank car rollover derailments. These include rotary couplers, increased shelf heights and the use of locking centre pins.

Rotary couplers are commonly used in unit coal train service. These enable the lading to be easily unloaded by rotating the railcar without uncoupling adjacent cars. The installation of a rotary coupler at each coupling position would likely eliminate the ability of a derailing car to transmit a torque to the adjacent car through the coupling. However, a rotary coupler would still permit the transfer of horizontal and vertical force components.

The installation of a rotary coupler at each coupling would likely eliminate the ability of a derailing car to transmit a moment to the adjacent car through the coupling, although a rotary coupler would still permit the transfer of horizontal and vertical forces.

In order to reduce the propensity for a derailing tank car to lift the near end of an adjacent tank car and lead to a multiple tank car rollover derailment, the height of the top and bottom shelves could be increased. This would allow greater relative vertical motion between two adjacent car bodies before a vertical force is transferred.

A third proposed option to mitigate multiple tank car rollovers is locking the centre pin to the truck and car body bolsters. The weight of the trucks increases the stabilizing moment on the car body, thus making a rollover derailment more difficult. In order to roll off the tracks with a locked centre pin, the tank car must roll about the wheel/rail contact point position. This ultimately increases the total rollover moment resistance of an empty tank car by 44 – 74 %.

Finally, the report presents economic assessments of the implementation of possible remedies. The economics of using rotary couplers with double shelves and using double shelves with increased heights has been evaluated. An economic assessment of the worst case cost for the implementation of double shelf rotary couplers and non-rotary increased height double shelf couplers was estimated to be $3,983 and $5,220 per car, respectively. These costs would be reduced substantially if the replacements were made when the car was in the shop for other reasons.

The next stage in this work is suggested to involve the evaluation of these proposed solutions through full scale physical testing.

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LIST OF ACRONYMS/ABBREVIATIONS

AAR	Association of American Railroads
ASF	American Steel Foundries
CISTI	Canada Institute for Scientific and Technical Information
CPR	Canadian Pacific Railway
CSTT	Centre for Surface Transportation Technology
FRA	Federal Railroad Administration
GATX	GATX Corporation
in.	Inch (or inches)
in-lbs	Inch-pounds
lbs	Pounds
mph	Miles per hour
NTSB	National Transportation Safety Board
NRC	National Research Council Canada
TC	Transport Canada
TDG	Transport Dangerous Goods Directorate
TSB	Transportation Safety Board of Canada

1 INTRODUCTION

When transporting dangerous goods, the importance of safety cannot be understated. One particular aspect of safety in relation to the transportation of dangerous goods by rail using tank cars is the prevention of the spill of dangerous goods. Any loss of dangerous goods cargo can risk the health and safety of the public, railroad employees, property and have devastating effects on the environment.

The safety of tank cars underwent significant improvement after several accidents in the 1960's and 1970's involving tank cars carrying dangerous goods. These accidents prompted several design changes to improve safety. One of these design changes was the introduction of head shields to make the ends of the tank less prone to puncture. The second change was the introduction of double shelf couplers. Double shelf couplers equipped with bottom and top shelves are designed to prevent decoupling during derailment events. The use of double shelf couplers effectively prevents adjoining couplers from disengaging vertically from each other, minimizing the risk of coupler override that can lead to the puncture of the tank shell and a release of dangerous goods.

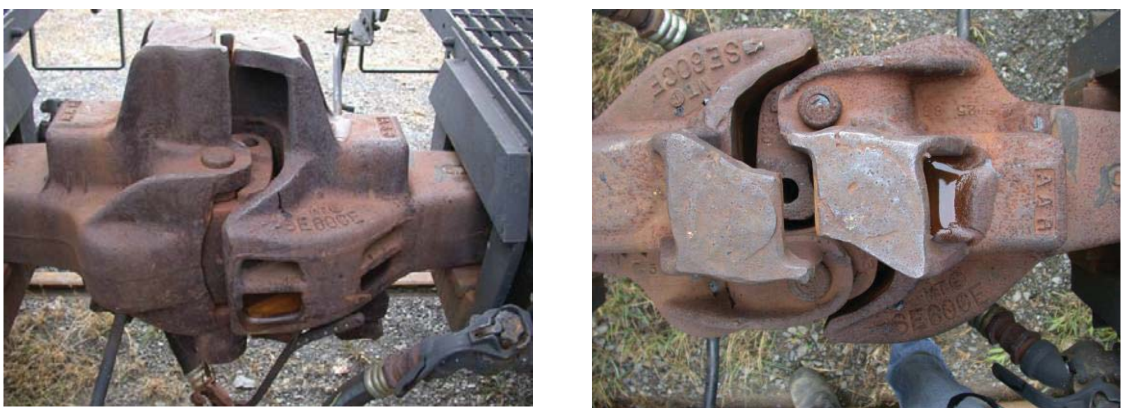

Figure 1: Double shelf couplers in service.

Double shelf couplers have proven very effective in preventing tank car punctures [1]. This is achieved by constraining knuckles with shelves to prevent couplers from disengaging vertically. The successful prevention of tank car puncture, however, has come at a cost. The introduction of double shelf couplers has resulted in a greater number of tank cars derailing when an accident occurs. Rollover and derailment of any single tank car in a train that contains a string of tank cars, for example unit trains, will frequently result in the rollover of many adjacent tank cars [2]-[7]. This creates a much larger derailment event known as a multiple tank car rollover derailment. TSB notes that empty tank cars are particularly susceptible to this event and that observation is confirmed by analysis in this report [2].

Since the introduction of double shelf couplers on tank cars, there have been many documented cases [2]-[7] where the derailment of a small number of tank cars has led to the rollover of many more coupled tank cars. During multiple tank car rollover derailments, an initial derailment of one or a few cars can lead to the progressive derailment of many tank cars. The progressive rollover derailment can be compared to dominos. An example of this is shown in Figure 2.

Figure 2: Rollover derailment of tank cars, Lévis, Québec, July 22, 2002 [8].

Six (6) TSB derailment reports have described incidents involving tank car unit trains where the initial derailment of a small number of cars subsequently derailed a large number of tank cars.

In TSB derailment Investigation report R05H0013 [2], the derailment of a unit tank car train was investigated. The derailment occurred in July 2005 near Prescott, Ontario. All 51 of its empty tank cars derailed. The reported cause of the initial derailment was due to lateral track deviation caused by track buckling. A very low rail neutral temperature promoted the track buckling. The report states that the double shelf couplers did what they were designed to do and no tanks were punctured. However, due to the engagement of the double shelf couplers, large torsional forces may be transferred through the couplers and when one car overturns; adjacent cars can be overturned. This can result in larger derailment events. The report describes five other incidents in the previous 10 years involving unit train tank cars with double shelf couplers where the initial derailment of a few cars lead to the overturning of a larger number of tank cars. These reports include:

R95D0016 – Gouin, Québec, January 21, 1995 [3] – a unit train had 28 of 44 loaded tank cars derail. Derailment was caused by gauge loss.

R99Q0019 – Bégin, Québec, April 13, 1999 [4] – a unit train had 10 of 68 loaded tank cars derail due to high cross-level variation in the track.

R02Q0041 – Lévis, Québec, July 22, 2002 [5] – a string of cars in a switching yard was set in motion by high winds which resulted in the derailment of 34 empty tank cars on one track. A struck tank car caused the remaining 33 cars to overturn and derail.

R04Q0026 – Saint-Charles, Québec, 2004 [6] – a unit train had 10 of 68 loaded tank cars derail due to lateral track deviation.

R04Q0040 – Saint-Henri-de-Lévis, Québec, August 17, 2004 [7] – a unit train had 18 of 68 loaded tank cars derail due to collapse of the track caused by subgrade failure.

A direct quote from Report R02Q0041 supports the belief that the double shelf couplers are responsible for increasing the total number of overturned cars in each derailment: "…the severity of the derailment was increased by the use of shelf couplers as the rollover of one car consequently provoked a rollover of the similarly equipped, adjacent cars" [5].

Figure 3 and Figure 4 are photographs of a tank car derailment that occurred in Clara City, Minnesota in October 2007. In this case, a train of empty tank cars was stationary on one track of a double-track main line. A moving train on the adjacent track derailed, and in the process derailed some cars of the empty tank car train. The tank cars that were initially struck on the stationary train then brought down much of the rest of the train. In Figure 3, the location of the initial derailment of the tank car train is seen near the top of the photograph. Figure 4 is a photograph of the derailed train taken from the opposite direction. This shows the extent of the tank cars which overturned following the initial derailment. It is also evident from the photographs that many of the trucks remained on the tracks and were essentially undisturbed. This is a very typical multiple rollover derailment related to the double shelf coupler. The field photos of the derailment have provided some important evidence in the present investigation on the mechanism of this kind of derailment.

Figure 3: Multiple tank car rollover derailment, Clara City, Minnesota, October 2007 [8].

Figure 4: Multiple tank car rollover derailment, Clara City, Minnesota, October 2007 [8].

CSTT was contracted by TC to investigate multiple tank car derailments with the goal of identifying potential modifications to minimize the extent of derailments involving double shelf couplers. TC's interest in this issue is motivated in part by a number of incidents in Canada that involved unit trains.

The purpose of this study is:

- To gather relevant information regarding tank car derailments and shelf couplers.
- To understand the mechanisms involved in the propagation of multiple tank car rollovers.
- To suggest and examine potential remedies to this problem, including the economic impact of proposed solutions.

In order to investigate the issue of multiple tank car rollover derailments and the impact of double shelf couplers, many sources were consulted. These include:

- Literature searches and review of information pertaining to double shelf couplers, tank cars, unit trains, and rollover derailments (FRA, NTSB, etc.)
- Data obtained from the FRA website containing statistics on multiple car rollover derailments of the past decade.
- Discussions with industry representatives regarding these types of derailments and double shelf couplers, including GATX, CPR, and ASF.
- Field trip to examine tank cars and double shelf couplers.

This report summarizes findings based on the detailed review and statistical analysis of the collected data and information. Two proposed mechanisms behind multiple tank car derailments are then discussed and several solutions are suggested for preventing or at least minimizing the extent of multiple tank car rollover derailments. Finally, the report presents economic assessments of the implementation of possible remedies.

2 DERAILMENT STATISTICS

2.1 Derailments

The FRA maintains a web page that allows one to perform a simple query on a database of rail accidents reported to the Office of Safety. This database includes a field for reporting the primary cause of the accidents, and one category for this field is “Equipment – Coupler and Draft System”. There are subcategories as well, listed below.

E30C- Knuckle broken or defective

E31C- Coupler mismatch, high/low

E32C- Coupler draw head broken or defective

E33C- Coupler retainer pin/cross key missing

E34C- Draft gear/mechanism broke/defective

E35C- Coupler carrier broken or defective

E36C- Coupler shank broken/defective

E37C- Failure of articulated connectors

E39C- Other coupler/draft system defects-car

None of these subcategories specifically addresses the type of coupler implicated as the cause of the accident. There are also no fields or categories to indicate derailments involving multiple car rollovers.

An advanced query can be performed, as well. In the advanced query, narratives are available for view. These are essentially a description of the accidents, in the words of the persons who witnessed, reported, or investigated them. There are fifteen narrative fields available for this purpose in the database. It is only in the narratives that one can find anecdotal evidence to implicate double shelf couplers as a cause of multiple car rollovers. There are a small number of direct statements to this effect, but most require some interpretation. The narratives do not indicate whether an accident involved a unit train or not, and there are no other fields to capture this,

All the narratives from the accident reports were reviewed and summarised in Table 1. In the twelve years of available data, there have been 308 accidents (out of 44,749 reported for all causes) where multiple (two or more) cars had rolled over, or that cars with shelf couplers had derailed. Of these, nineteen cases were directly stated as being the result of the use of double shelf couplers. Seventy-two cases were indirectly linked to the use of double shelf couplers. Data from Table 1 are shown graphically in Figure 5 and Figure 6.

Table 1: FRA statistics for multiple car derailments or accidents involving shelf couplers.

			Double Shelf Couplers		Rotary Couplers		Couplers	Cars		
Year	Main/Sdng	Yard/Ukwn	Directly	Indirectly	Directly	Indirectly	Unknown	Ovrtrned	Incidents	Records
1995	11	20	1	3	0	7	20	177	31	3226
1996	8	17	0	2	0	1	22	61	25	3183
1997	10	12	5	1	0	0	16	37	22	3069
1998	14	16	2	1	0	0	27	160	30	3295
1999	7	13	1	2	0	0	17	63	20	3550
2000	4	11	0	4	0	1	10	67	15	3867
2001	6	18	3	21	0	0	0	25	24	3984
2002	2	13	4	11	0	0	0	23	15	3593
2003	10	18	0	7	0	0	21	74	28	3975
2004	7	24	1	6	0	1	23	102	31	4497
2005	10	24	0	7	0	0	27	88	34	4467
2006	15	18	2	7	0	4	20	131	33	4043
Totals	104	204	19	72	0	14	203	1008	308	44749

FRA Statistics for Multiple Car Derailments or Accidents Involving Shelf Couplers

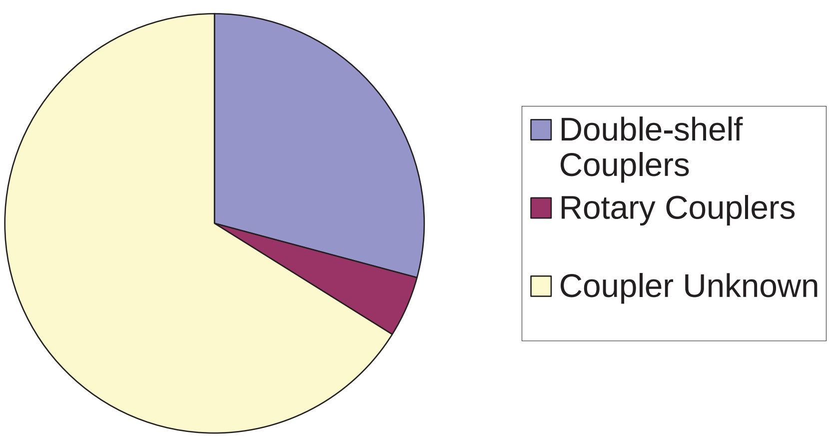

Figure 5: Double shelf couplers involved in a large number of multiple car derailments.

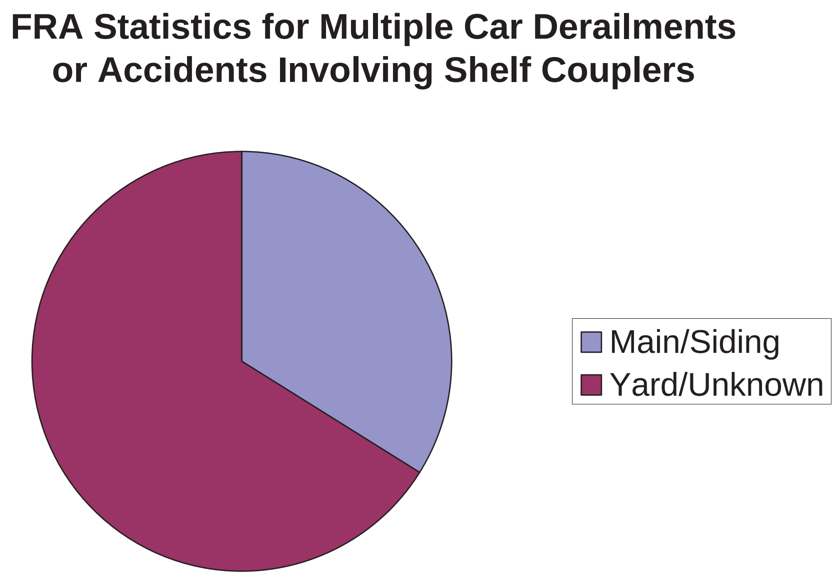

Figure 6: A significant proportion of multiple car derailments occur in yards.

The data in Table 1 could be further broken down by separating the "cars overturned" column into two or three new columns. The first would capture the number of tanks cars overturned, and the second would capture the number of other car types overturned. The third column would capture the number of unknown car types.

The information available from the narratives in the FRA database sometimes clearly indicate the number of tank cars overturned; other times they simply indicate that "7 cars overturned". In some cases, actual car reporting marks are given. These car types could be identified by looking the reporting marks up in Umler, but the results would still be incomplete due to records whose narratives do not explicitly indicate car type or reporting marks. A similar problem would exist if the data were to be segregated into loaded and empty cars overturned. The narratives do not always explicitly indicate whether the overturned cars were loaded or empty.

One benefit of such a breakdown of the data would include a more accurate assessment of the extent to which tank cars overturn (relative to other cars types) when they are involved in a derailment or collision. A second benefit would be the identification of trends to indicate whether lading influences tank car rollover or not. Either of these benefits could serve to focus further research into the problem.

The statistics gathered from the FRA Office of Safety database included references in the narratives to rotary couplers. From the available data, rotary couplers were never directly implicated as the primary cause of any multiple car rollover derailments, but they were indirectly linked on fifteen occasions. It has been assumed that any mention of coal cars in the narrative fields implies that they are equipped with rotary couplers, although this may not be the case. Data by type of coupler are presented in Figure 5.

It can be seen from Figure 6 that half as many accidents occurred on mainlines/sidings than in yards or in unidentified locations. This could indicate that multiple car rollovers tend to occur at lower speed. It is difficult to identify trends in the causes for the rollovers from the narratives, as there is no consistent method of describing the accidents. Sometimes there is no mention of what caused (or may have caused) the accidents, but broken rails, incorrect cross-level and sideswipes are frequently mentioned. In a few cases, the narratives mention accidents where a car has lost one truck but remains upright while the train continues along the track, supported by shelf couplers. This effectively prevents the cars from coming apart and separating the air hoses.

NRC's library (CISTI) was asked to search for any reports or documentation that contained references to double shelf couplers and rotary couplers, related to derailments, for any rollover derailments that also mentioned couplers and for unit trains that had been involved in derailments. These CISTI searches located many reports containing information relevant to this project. The abstracts for all reports were reviewed, and several were then selected for further review by CSTT.

Information was also collected from the Transportation Safety Board of Canada in the form of accident investigation reports relating to unit train derailments that took place between 1995 and 2004. Other documents reviewed included:

- US National Transportation Safety Board reports relating to accidents involving tank cars or hazardous materials releases [9], [10], [11], [12], [13], [14], [15], [16]
- A discussion of coupler override mechanisms [17]
- A report on damage prevention in the railroad tank car industry [18]
- A report on coupler height mismatch [19]

Older reports and papers [20], [21], [22] that mention the use of double shelf couplers on tank cars often note that these couplers had been effective in preventing tank head punctures by preventing the cars from uncoupling during derailments. More recent reports [3], [4], [7], [10], [13], [16] seldom mention the use of double shelf couplers other than to indicate they are required equipment on tank cars.

The TSB reports (for derailments involving tank cars) seldom specifically implicate the double shelf couplers as the source of multiple tank car rollovers. This is true of accident reports by other agencies (FRA, NTSB, etc.). Other information that was reviewed relates to:

- The development of double shelf (and other tank car improvements) to reduce the occurrences of tank puncture and product release [23], [24].
- Derailment statistics for couplers, without indicating the type of coupler [25].

- Statistics demonstrating that double shelf couplers (and head shields) have reduced the number of tank shell ruptures [1], [23], [26], and [27].
- Generic requirements for alternative couplers [28].

These reports suggest that the double shelf couplers are responsible for increasing the total number of overturned cars in each derailment.

2.2 Double Shelf Couplers

Double shelf couplers have been used successfully to prevent tank car punctures caused by vertical coupler disengagement during derailments [1]. Both E-type and F-type couplers can have double shelves. Two mated F-type couplers cannot move vertically relative to one another (even without double shelves). An F-type coupler can also be mated with an E-type coupler and the double shelf mechanism will then prevent vertical disengagement. Without double shelves, a mating of two E-type couplers has no mechanism to prevent vertical disengagement. Figure 7 shows the components of an E-type double shelf coupler.

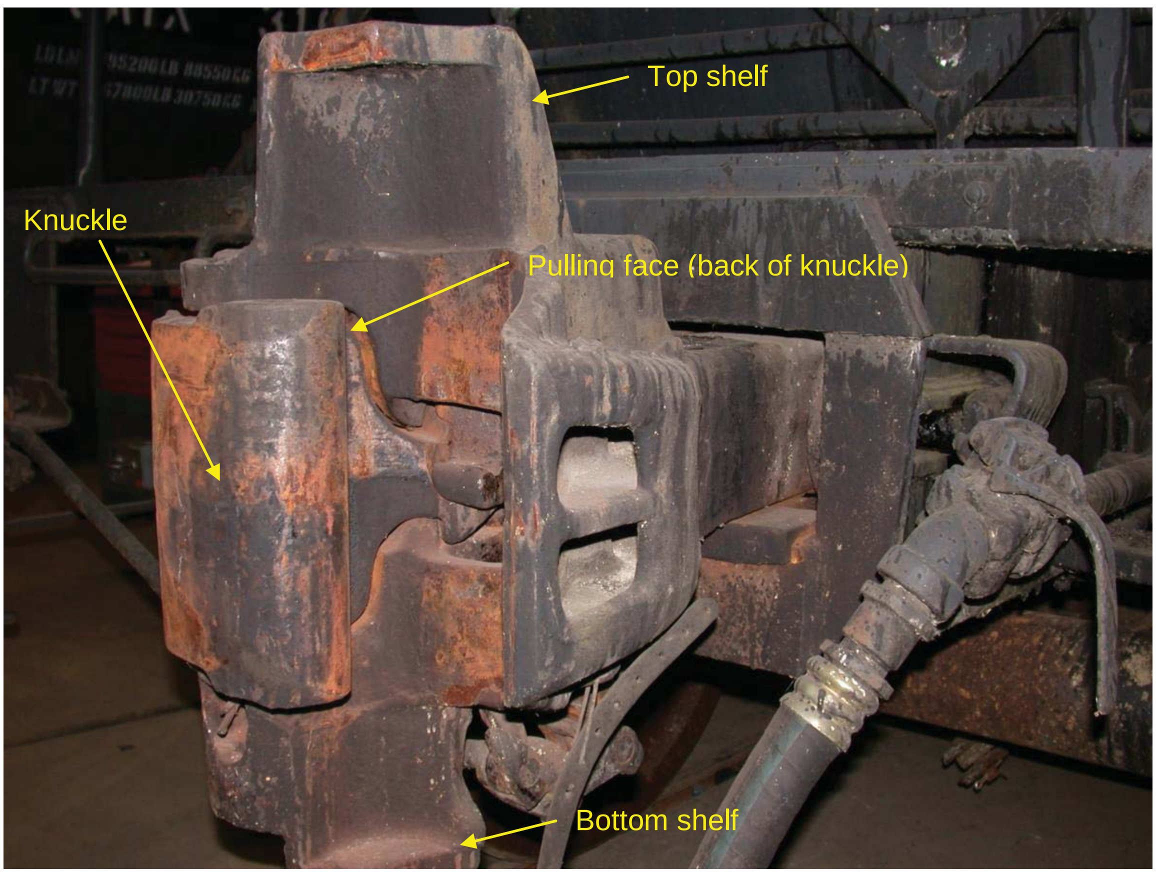

Figure 7: E-type double shelf coupler.

The basis of the heights of the shelves can be understood by considering the following. The centreline of couplers on rail cars can be between 31.0 and 34.5 in. above the top of rail. The couplers on new, unloaded cars can be no higher than 34.5 in. above the top of rail. The static height of the coupler above the top of rail decreases as the wheels wear (maximum of 1.5 in.), and will decrease again as the car is loaded (approximately 2 in.). Other things that contribute to reduced coupler height are wear of the coupler carrier, wear of the “wear plate” that is sometimes welded on to the bottom of drawbars, wear of the centre bowl liner and permanent set of the suspension springs. If the coupler height drops below 31.0 in., the coupler carrier is generally replaced to raise the coupler back up.

The distance between the working faces of the upper and lower shelves of an E-type double shelf coupler is 23.75 in., symmetrical about the coupler centreline. The height of the knuckle’s pulling face is 11.0 in. Therefore, one knuckle can slide vertically relative to the other by 6.375 in. before a knuckle will contact a shelf.

Assuming that two cars, at the opposite extreme limits of coupler height, could be joined together, they would have used 3.5 in. of the 6.375 in. clearance. The pulling faces of the knuckles would then be engaged over a length of 2.875 in. Note that continuous operation with less than approximately 3 in. of knuckle engagement is undesirable due to the risk of fracturing the corner of knuckle’s pulling face and subsequent car separation.

If these cars are rolling along a rough track, they can bounce or pitch on their trucks. This motion could consume an additional 1.25 – 1.5 in. of the clearance. This leaves a minimum of 1.375 in. of clearance, under poor or rough operating conditions, to ensure that unwanted contact between a knuckle and shelf does not occur.

CSTT has made the following observations based on cars in the field:

- Contact between the vertical faces of the shelves can occur in service, as evidenced by rub marks where contact has occurred on these faces.
- There was no appreciable wedge rise on the truck suspensions.
- There can be a significant gap between the vertical faces of the shelves of coupled couplers when the cars are in draft. However, when the cars are in buff, this gap can be reduced to zero (resulting in the rub marks as seen on the shelves of some couplers).
- Some of the cars were at different heights, resulting in coupler height mismatches. Some couplers also appeared to have drooped within their draft sills. In these cases, the yoke key had also rotated within the clearance of its slot. Drooped couplers contribute to the coupler height mismatch. An example is shown in Figure 8.
- The combination of coupler height mismatch and zero clearance gap (or negative gap) between the vertical faces on the coupler shelves can potentially cause binding between the couplers when the cars are in buff and undergoing vertical dynamic motion.

- Constant contact side bearings (long travel) are used.
- Both E- and F-type couplers are used.

CSTT also learned:

- The AAR issued a circular [29] requiring tank cars to have long travel constant contact side bearings installed when shopped for qualification. The preload on these side bearings should by design be 4,500 lbs. Prior to this tank cars typically used roller side bearings.
- Centre plate locking pins are generally not being used on unit train cars. This is not part of the AAR standards, and is therefore not in general use in the rail industry. However, some customers do require it.
- In rollover derailments where the car bodies have separated from their trucks, the centre pins tend to remain in the truck bolster and show no evidence of damage.

CSTT contacted FRA to discuss the issue of multiple tank car derailments associated with double shelf couplers. According to the FRA, these types of derailments are very uncommon in the United States, as tank cars are seldom run in unit train service. They felt that these types of derailments could increase in the future as shipments of ethanol by rail [30] increase.

FRA felt that the use of rotary double shelf couplers might be useful in unit tank train service, as they have performed well in other unit train service. F-type couplers would offer the further advantage of reduced slack between couplers. An E/E coupling has 2 in. of slack, whereas an F/F coupling has 0.625 in. of slack. Reduced slack would improve longitudinal train dynamics. F-type couplers also have increased lateral gathering range, approximately 4 in. versus 2 in. for E-type couplers. This is beneficial for yard operations, as it should reduce the number of accidents that result from bypassed couplers.

Figure 8: Double shelf couplers mated together showing height mismatch due to different coupler centreline heights, and a drooped coupler on the left car.

3 DERAILMENT MECHANISMS

In order to provide solutions to minimize multiple tank car rollovers, the mechanisms that propagate the derailment from car to car through the couplers need to be understood. After reviewing derailment reports and photographs, two mechanisms have been identified that likely work in combination.

The first mechanism investigated is rollover due to a moment transferred between couplers. The second mechanism is dominated by the transfer of vertical motion from a car that is derailing to the adjacent car that is still on the rails. Although this study focuses on empty car derailments, the methodology and general conclusions will hold even for loaded cars. Free surface effects due to the ability of a liquid to move inside a tank would not affect these calculations, as the change in lateral location of the centre of gravity is insignificant. The longitudinal location of the centre of gravity can be shifted as liquid can slosh toward one end of the tank. This will affect longitudinal train dynamics, but not the analysis presented.

3.1 Moment Transfer

The rollover resistance of a single empty tank car to a torsional load applied to the car body through the couplers is presented below. Older tank cars frequently used roller side bearings with a ¼ in. allowable clearance. However, in 2005 the AAR required tank cars to use long travel constant contact side bearings (CCSB) [29].

Roller side bearings are normally unloaded, as a gap (e.g. ¼ in.) is left between the wear plate on the car body bolster and the rollers. These side bearings have a simple design and have demonstrated long service lives. CCSB are preloaded, often by a compressed elastomer. The advantages of CCSB include better car body roll stability and hunting control.

Rollover resistance calculations were performed with both roller side bearings and long travel CCSB. The results for each case are presented in the following sections. Detailed calculations are supplied in Appendix A. The tank car used in the calculations in this report has a tank with an inner diameter of 108 in. and a 7/16 in. wall thickness. The car body has an empty weight of 57,500 lbs. The rollover resistance of other tank car configurations (e.g. tank diameter, etc.) are not examined here, but can be computed in a similar fashion.

3.1.1 Roller Side Bearings

CSTT performed engineering calculations to estimate the quasi-static external moment applied to the tank car body that would precipitate a rollover. The selected model was that of a single tank car, positioned on level track. To simplify the calculations, standard roller side bearing clearances were assumed, no lateral forces were applied to the system, and locking centre pins were not used. Symmetry at both ends of the tank car was assumed. There were several stages of rollover moment resistance, which varied piecewise-linearly with suspension roll angle, depending on the body roll angle with respect to the truck bolsters. Each stage is described below and illustrated in Figure 9.

<u>Stage 0:</u>

The car body is sitting level on its centre plate in the centre bowl, and no external moment is applied.

<u>Stage 1:</u>

Because of an externally applied moment on the tank car body, the body roll angle increases until the centre plate is just about to rock onto its bevelled edge. Although the body does not roll with respect to the truck bolster, the bolster rolls on the suspension springs because the distribution of the car body weight on the centre plate is moving towards one edge of the plate.

Stage 2:

The body roll angle increases until the body just contacts the side bearings (25 in. away from truck centre) without loading them. Note that the truck bolster does not undergo any additional roll during this stage, because the load-bearing contact point (the bevelled edge of the centre plate) does not change position. However, the body rolls with respect to the truck bolster to use up the allowable side bearing clearance.

Stage 3:

At this stage, the body and truck bolster roll together until the bevelled edge of the centre plate just separates from the centre bowl. All of the car body weight is now carried on the side bearings. Therefore, the weight on the side bearings increases from no load to the full load of the car body.

Stage 4:

The body continues to roll under the externally applied moment about the side bearings until the weight vector of the car body passes through the side bearing. At this point, the car body is balanced on the side bearings in a meta-stable position. The applied moment required to keep it in this position is zero; therefore, if the car body were perturbed from this position in either direction it would either roll off or back on to the truck.

Stage 5:

The car body rolls off the trucks onto its side on the ground. Before striking the ground, the car body bolster contacts the side frame.

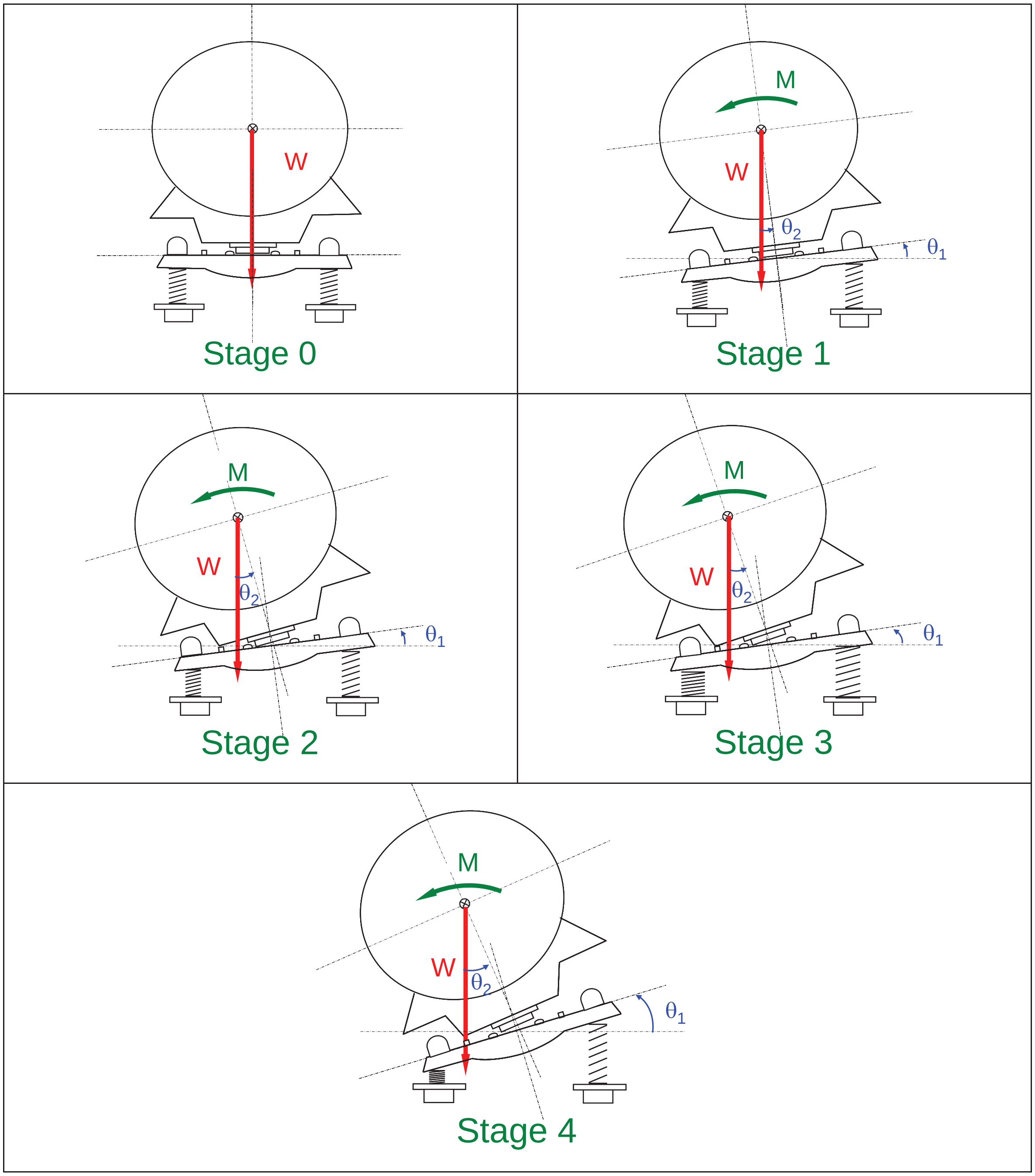

Figure 9: Stages of tank car rollover due to overturning moment – roller side bearings.

Table 2 shows the external quasi-static moments that must applied to the body at each stage in order to roll it to the positions described. The progression of the external moment is plotted in Figure 10.

Table 2: Applied external moment and roll angles for tank cars with roller side bearings.

Stage j	Applied body moment M_j (in-lbs)	Bolster roll angle (rads)	Body roll angle (rads)	Bolster roll angle (deg)	Body roll angle (deg)
0	0	0.0000	0.0000	0.00	0.00
1	367,667	0.0023	0.0023	0.13	0.13
2	331,445	0.0023	0.0158	0.13	0.90
3	1,377,881	0.0087	0.0222	0.50	1.27
4	0	0.0087	1.0786	0.50	61.80

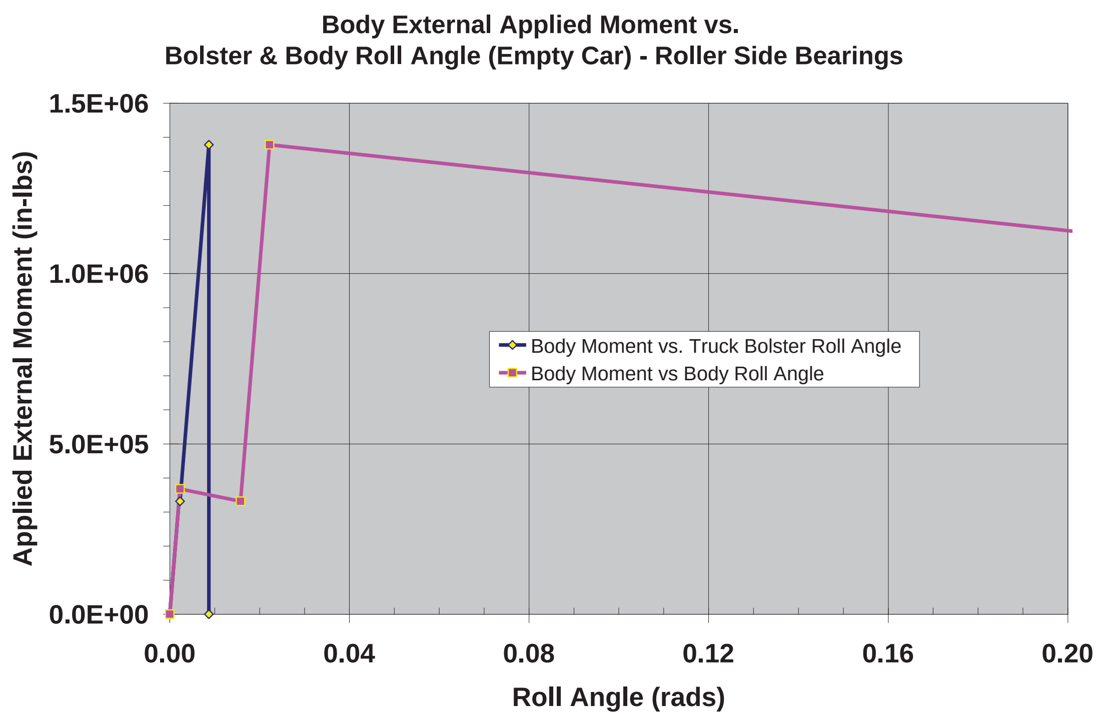

Figure 10: Applied external moment required for rollover of tank cars with roller side bearings.

It is evident that the tank car body has a very large self-stabilizing moment, once the body weight is carried on the side bearings. The maximum moment is almost 1,380,000 in-lbs, which helps it to resist body rollover. Although this moment decreases to zero once the weight vector passes through the side bearings, the car body must roll nearly 62° from level for this to occur. This is much higher than a maximum roll of 6º peak-to-peak (3º from horizontal) that is specified in AAR Manual of Standards, Section C Part II, Chapter 11 [31]. If the car were to be coupled at each end, the extra roll stabilizing moment from the adjacent cars would increase the rollover resistance of the middle car even further. Quasi-statically, the externally applied moment for a single car to initiate a sequential rollover of several cars would have to be extremely high. Large lateral inertial forces must also be applied to the car body in order to develop the rollover moments. Inertial effects would also be important when calculating the roll angle of the truck bolster. Inclusion of these contributing effects would be impractical for hand calculations; dynamic modeling would be needed. The amount of rotary slack within the coupler knuckles should also be included in the model. A discussion with a representative from ASF revealed that the amount of knuckle slack in rotation is not known [32]. A relatively simple physical test could be performed to determine the magnitude of this slack.

3.1.2 Long Travel Constant Contact Side Bearings

The rail industry began replacing roller side bearings with long travel CCSB when tank cars were brought to repair facilities. Therefore, CSTT performed the same hand calculations presented in the previous section, but this time the addition of long travel constant contact side bearings were assumed. The model was of a single tank car, positioned on level track. No lateral forces were applied to the system, and locking centre pins were not used. There are several stages of rollover moment resistance, which vary piecewise-linearly with suspension roll angle, depending on the body roll angle with respect to the truck bolsters. The stages of rollover are described below and illustrated in Figure 11.

<u>Stage 0:</u>

The car body is sitting level on its centre plate in the centre bowl, and no external moment is applied. Vertical load is shared by centre bowl and side bearings. Each side bearing carries an equal share of the vertical load.

<u>Stage 1:</u>

Because of an externally applied moment on the tank car body, the body roll angle increases until the centre plate is just about to rock onto its bevelled edge. Although the body does not roll with respect to the truck bolster, the bolster rolls on the suspension springs because the distribution of the car body weight on the centre plate is moving towards one edge of the plate.

<u>Stage 2:</u>

The car body rolls (on the bevelled edge of its centre plate) relative to the truck bolster until the CCSB on one side of the trucks becomes fully unloaded. The load is thus shared by the side bearings on the side of the truck bolster that the car body is rolling towards and the centre bowl.

<u>Stage 3:</u>

The car body continues to roll (on the bevelled edge of its centre plate) relative to the truck bolster until the CCSB on one side of the trucks go solid. The CCSB on the opposite side of the trucks are no longer in contact with the car body. The force carried by the side bearings in contact with the car body increases, and the force carried by the centre bowls decreases.

Stage 4:

Body and bolster roll together until the centreplate separates from the truck centre bowl and all vertical reaction force is through the CCSB contact at 25 in. from the truck's longitudinal centre line. The centre bowl carries no vertical load and the truck bolster roll angle stops increasing.

Stage 5:

The car body continues to roll under the externally applied moment about the CCSB until the weight vector of the car body passes through the CCSB. At this point, the car body is balanced on the CCSB in a meta-stable position. The applied moment required to keep it in this position is zero, and if the car body were perturbed from this position in either direction, it would roll off or back on to the truck.

Stage 6:

The car body rolls off the trucks onto its side on the ground. Before striking the ground, the car body bolster contacts the side frame.

Table 3 shows the external quasi-static moments that must be applied to the body at each stage in order to roll it to the positions described. The progression of the external moment is plotted in Figure 12. The peak moment resistance to rollover is quite similar for both roller side bearings and long travel CCSB, so we can conclude that the type of side bearings should not significantly affect the amount of rollover resistance of an empty tank car. A very large torsional force must be transmitted through the coupler connection in order to roll over an empty tank car.

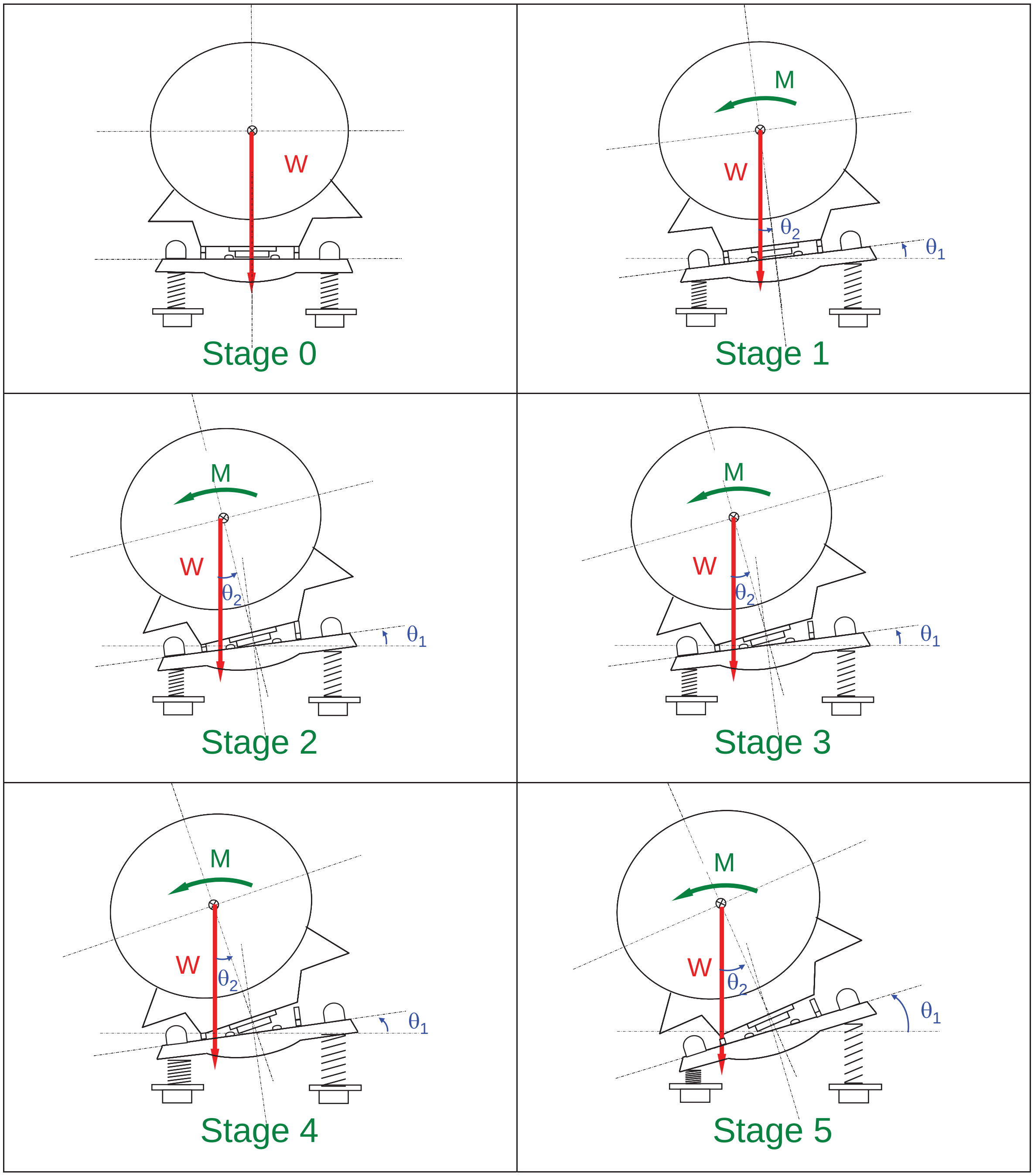

Figure 11: Stages of tank car rollover due to overturning moment – long travel constant contact side bearings.

Table 3: Applied external moment and roll angles for tank cars with long travel constant contact side bearings.

Stage j	Applied body moment M_j (in-lbs)	Bolster roll angle (rads)	Body roll angle (rads)	Bolster roll angle (deg)	Body roll angle (deg)
0	0	0.0000	0.0000	0.00	0.00
1	252,571	0.0016	0.0016	0.09	0.09
2	672,248	0.0045	0.0267	0.26	1.53
3	932,302	0.0063	0.0401	0.36	2.30
4	1,323,578	0.0087	0.0425	0.50	2.44
5	0	0.0087	1.0786	0.50	61.80

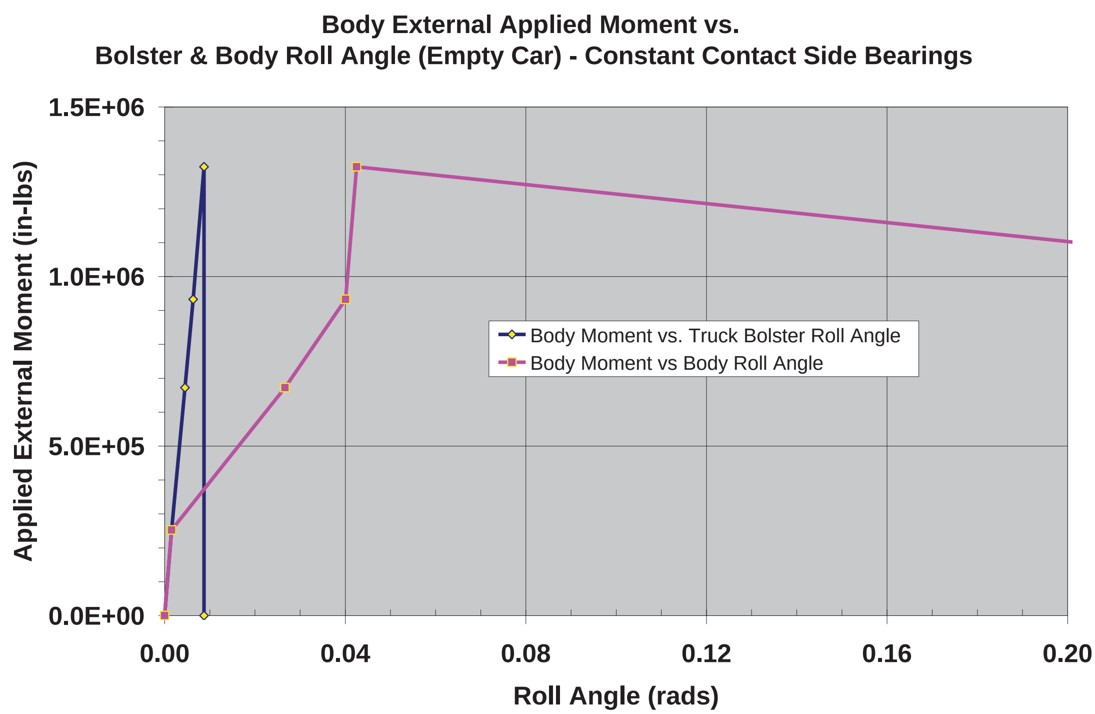

Figure 12: Applied external moment required for rollover of tank cars with long travel constant contact side bearings.

3.2 Vertical Force Transfer

In addition to the large external moment required to roll an empty tank car body off its trucks, further evidence exists to suggest that an additional mechanism must also be involved to create the resulting roll over condition. The mechanism that causes overturning of the car immediately adjacent to the initiating car is thought to have a significant vertical component of motion imposed upon it. This idea is supported by several key pieces of evidence from the derailment photos.

It was noted in the Clara City derailment (Figure 3 and Figure 4) that the centre pins on many trucks were still in place and undamaged. Figure 13 shows a centre pin sitting in the centre bowl. It can be seen that there is a limited amount of lateral slack and resulting potential lateral movement with only 3/8 in. total clearance between the pin and the mating hole in the truck bolster (3/16 in. on each side of the pin). The centre pin is 15 in. tall and has a 1.75 in. diameter. The height of the pin is about 8 in. measured from the top of the pin to the inside of the centre bowl liner. This means that for the car body to roll off the truck without causing damage to the centre pin, at least 8 in. of vertical movement must be obtained without exceeding the allowable lateral movement of the pin.

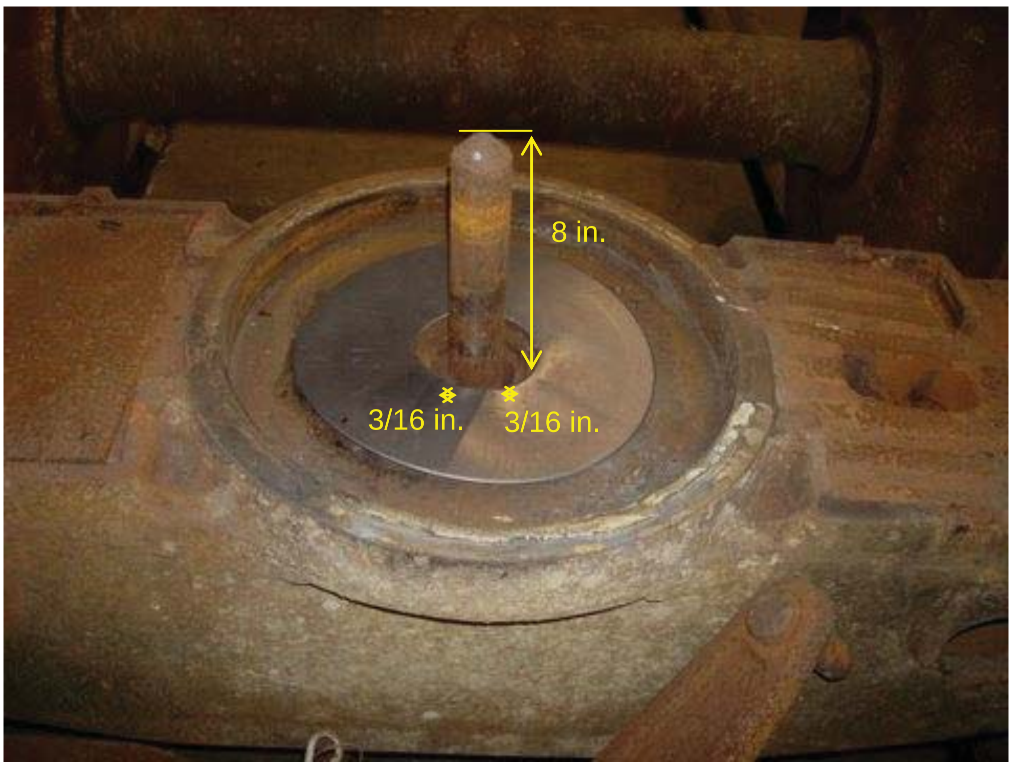

Figure 13: Centre pin height above centre bowl liner.

Considering that the car body rolls about the side bearings, there is not enough lateral clearance and resulting movement in the centre pin to achieve this vertical movement without causing noticeable damage to the centre pin. Figure 14 illustrates the maximum height of car body displacement relative to the centre pin so as not to damage the centre pin. Since the centre pin has a total of 3/8 in. lateral clearance (when the pin is centred it has 3/16 in. lateral clearance on either side) the maximum vertical lift that can thus be obtained is 3.1 in. at a rotation of 7º between the car body and truck bolster. Therefore, the torsion mechanism cannot fully account for the rollover effect since there is not enough clearance in the pinhole for the car body to roll off the trucks about the side bearing without additional vertical motion. Therefore, a significant vertical component becomes necessary for the rollover to occur.

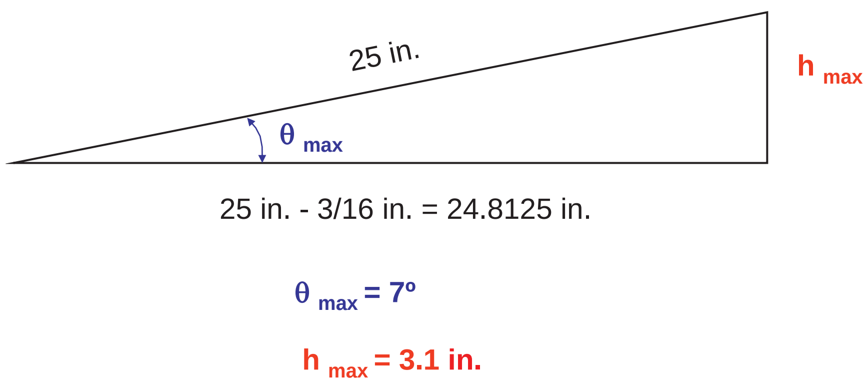

Figure 14: Maximum vertical displacement of the car body at the centre pin and rotation about the side bearings to avoid visible damage to the centre pins.

Figure 15 shows the convention used to describe the ends of derailed cars with respect to the point of derailment. For each tank car, the end closest to the point of derailment is referred to the “near” end and the end furthest from the initial point of derailment is referred to as the “far” end.

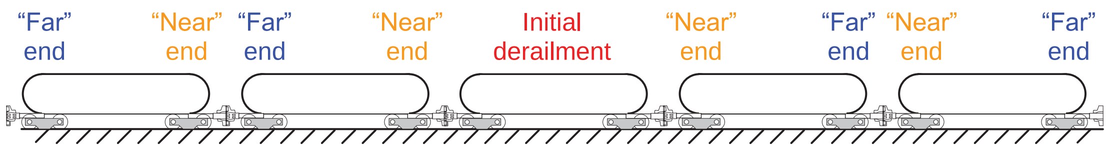

Figure 15: Definition of near end and far end of derailed cars with respect to the initiating point of derailment.

Figure 16 and Figure 17 are zoomed in versions of Figure 3. Both Figure 16 and Figure 17 show a pattern in that the near end trucks are undisturbed as they remained on the tracks after the derailment, with their centre pins undamaged and still in the trucks. The far end trucks, however either have rolled off the tracks with the car body, or have been displaced off the tracks with visible damage to and/or displacement of the centre pin. In Figure 17, truck springs are also observed to have come dislodged from the spring nest. This suggests that one truck bolster has lifted enough to allow the springs to drop out.

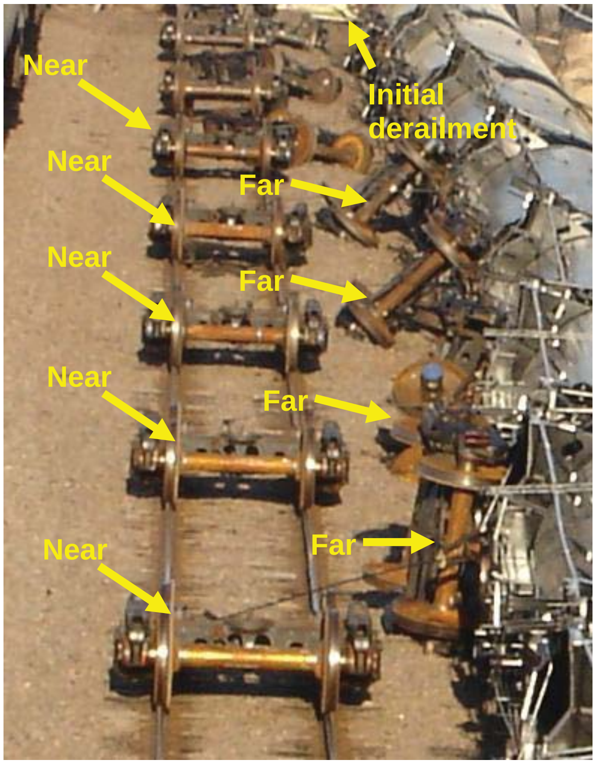

Figure 16: Pattern of near end and far end trucks after roller over derailment [8].

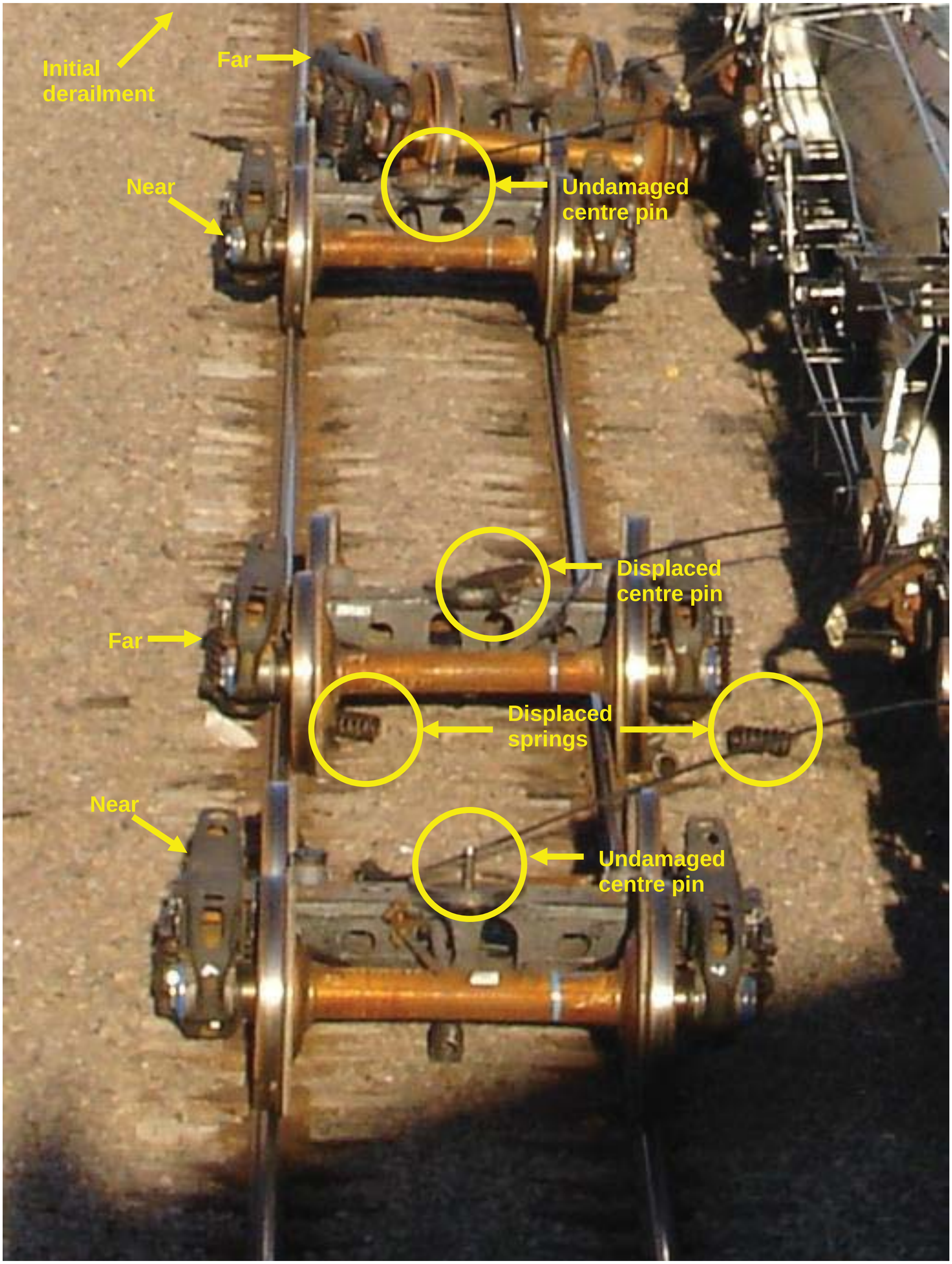

Figure 17: Every second truck remains on track with no damage to centre pin [8].

Additional evidence from derailment photos suggests that vertical force is a major contributor in transmitting rollover to multiple tank cars; this force can be high enough to cause carrier irons to fail. Figure 18 shows a carrier iron broken at the welds after a derailment involving a unit train in which 18 of 68 loaded tank cars derailed due to a subgrade failure [7]. In this image, the tank car is rolled over onto its side. As the initiating car derails, its coupler tries to lift the near end of an adjacent car, the coupler on the initiating car is forced downward upon its carrier iron and in the case of Figure 18, the force was large enough to break the carrier iron off at the weld.

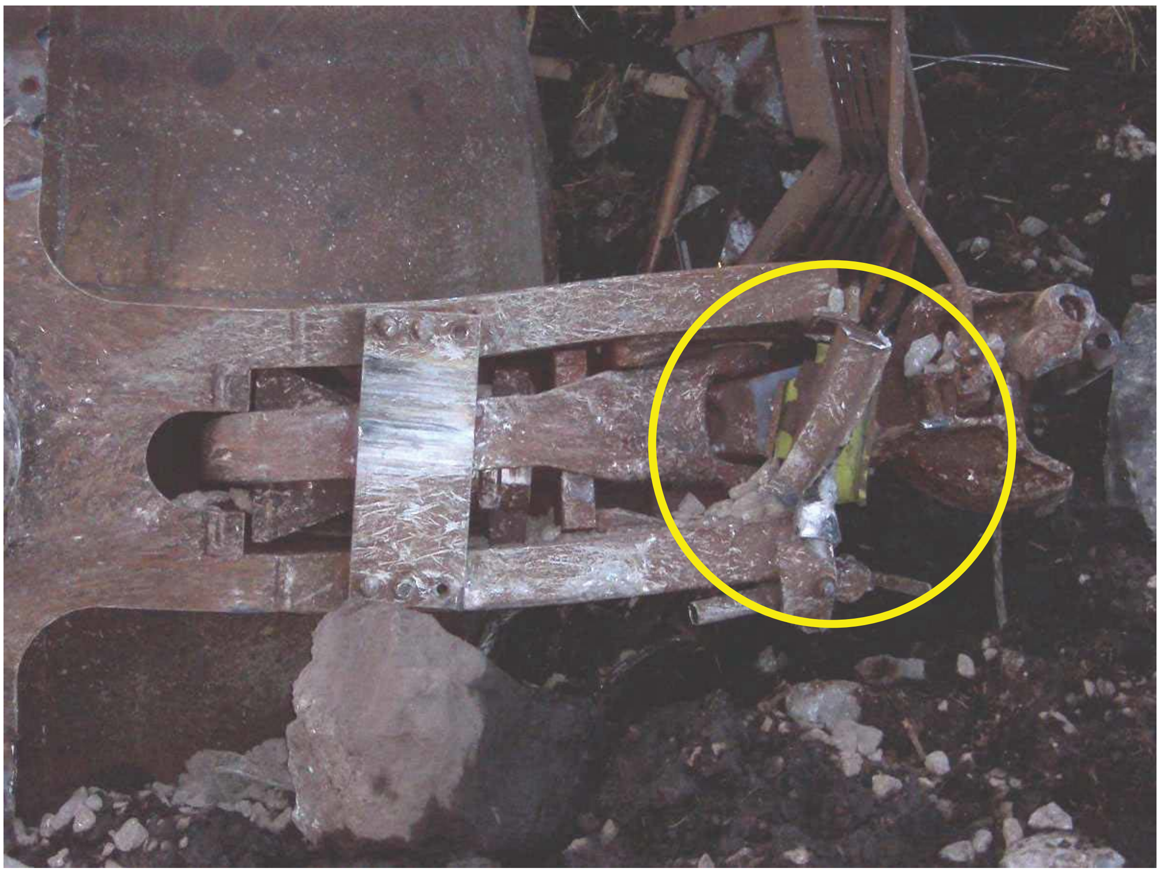

Figure 18: Broken carrier iron after derailment suggests a large vertical force was transmitted through the couplers during derailment.

The end walkway of several tank cars can be seen in derailment photos to have been bent in an upward position. One example is shown in Figure 19. The photograph was taken after the same accident [7] was cleaned up, and this loaded tank car, which had derailed and rolled over, was placed upright on the tracks. This also suggests a significant vertical force being involved in the rollover mechanism.

Figure 19: Coupler and end walkway bent up after derailment.

The existence of a significant vertical force involved in the rollover mechanism and the resulting pattern of trucks after such a derailment, as shown in Figure 16 and Figure 17, are explained in the following paragraphs.

When a tank car that initiates a roller type derailment is rolled onto its side, the rotation of the car body causes the height of the coupler to increase (see Figure 20) by at least 19.5 in. The double shelf E-type couplers can travel vertically relative to one another by 6.375 in. until they hit a shelf. This means that in order to go from the initial position (upright on track) to the final position (rolled onto side) the minimum required vertical motion of the coupler on the initiating car is about 13 in. as shown in Figure 20.

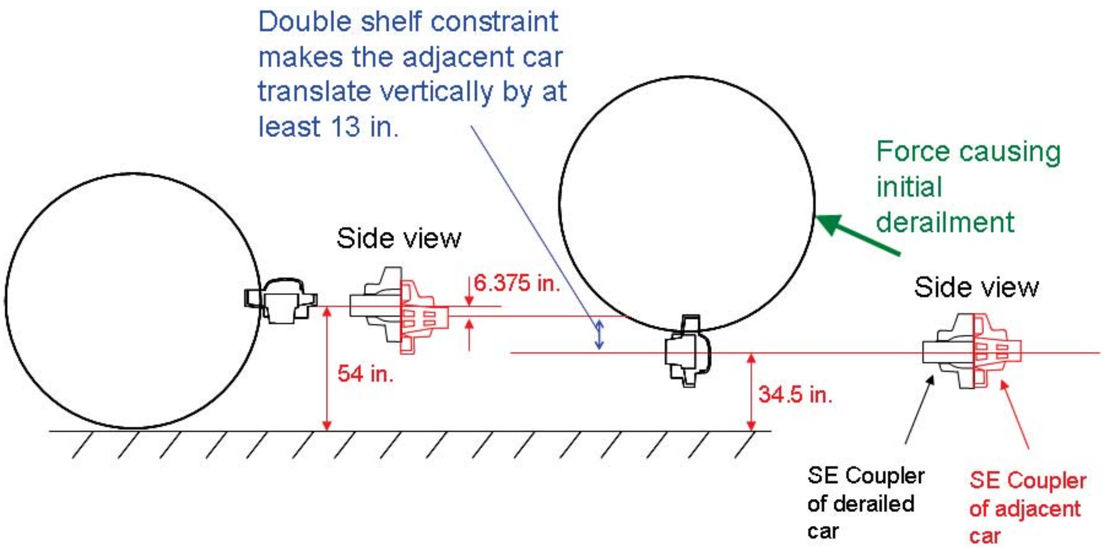

Figure 20: Vertical motion contribution to rollover derailment of empty tank car with double shelf couplers.

In order for the derailment initiating tank car to end up lying on its side, a plausible coupler path is shown in Figure 21. This potential path of the coupler demonstrates that the coupler must travel a greater vertical distance than suggested by Figure 20. In addition to a vertical component, a horizontal and a rotational component are involved in the couplers motion. The presumed path of the coupler on the derailing car implies the application of not only a torsional force on the adjacent coupler, but a vertical and horizontal force as well. With the two mechanisms (moment and vertical) acting together, the vertical lift at the near end of the car will reduce the required external moment to roll the car body off the near trucks without damaging the centre pin.

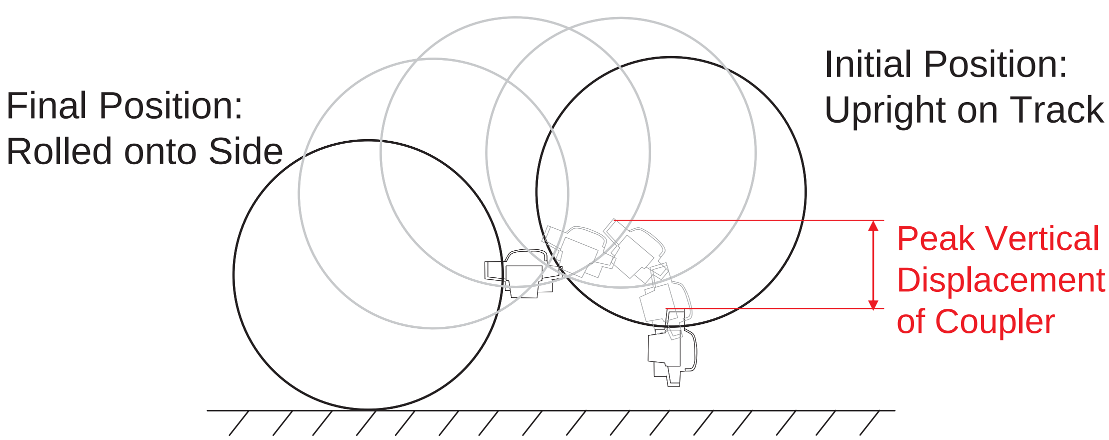

Figure 21: Possible path of coupler during rollover derailment.

Due to the large vertical displacement of the coupler along that path, the knuckle of the coupler on the car initiating the derailment lifts vertically on the adjacent car coupler through the top and bottom shelves of the coupler pair. Figure 22 shows that when the tank car initiating the derailment lifts vertically on the adjacent cars through the coupler shelves, the near ends of the adjacent cars are lifted sufficiently to clear the centre pins. To lift the near end of the car body off the centre pin requires a vertical force of about one-half the total weight of the empty car body (~30,000 lbs).

The near ends of the adjacent cars are able to roll over onto their sides without damaging the centre pin. The far ends of the cars, however, are not able to clear the centre pin and thus as these cars derail, the back end of the cars exert a horizontal force on the centre pins that damages them, and/or causes the trucks to become displaced from the track. Once the adjacent cars have derailed, the adjacent tank cars themselves become initiating cars, this process repeating itself like dominos.

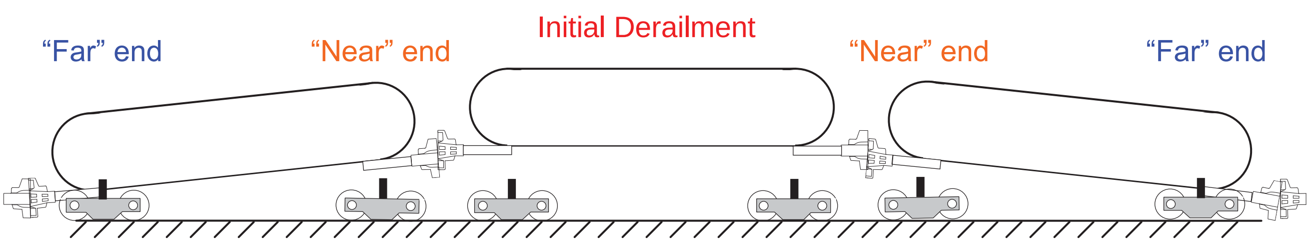

Figure 22: Car in the middle initiates the rollover derailment, and transmits large vertical forces to the connected couplers, causing the near ends of the adjacent cars to lift and clear the centre pins.

Once the near end of the adjacent car is lifted off and clear of its centre pin and then begins to roll off the trucks, the far end of the car must follow. However, without the same vertical lift as the near end, the rear end simply rolls, damaging the centre pin and often taking the truck with it. This can be seen in Figure 16 and Figure 17 where the near truck remains on the track with no signs of damage to the centre pin while the far trucks are either displaced from the track, but remain upright (often showing damage to the centre pin), or they have rolled off the track with the car body.

3.3 Summary of Mechanisms

Initiation of rollover derailment usually involves a lateral impact to the first derailing car. For a tank car with roller side bearings that is not coupled to any other cars a moment of at least of 1,378,000 in-lbs is required (roller side bearings). If the impact point is at a height about ¼ of the tank diameter (27 in.), the lateral force required should be higher than 50,000 lbs. For a car that is coupled to cars on each of its ends, the required force should be greater than 100,000 lbs.

Similar values are obtained for a tank car with a larger tank diameter and therefore a higher centre of gravity (e.g. 119 3/8 in. inner tank diameter and 72,920 lbs light weight) which was calculated to have a nearly identical peak rollover resistance of 1,362,000 in-lbs (assuming roller side bearings). As well, the required lateral force to initiate rollover would be similar to that of the tank car studied in this report (108 in. inner tank diameter and 77,500 lbs light weight). For tanks of equal weight, the one with the higher centre of gravity will have a lower stabilizing moment, but the difference is relatively small.

The near end of the adjacent car can be lifted by more than 13 in. by the derailing car, easily clearing the centre pin. This vertical lift is due to the height difference between the standing car and rollover car, and because the couplers are prevented from disengaging by the double shelf couplers. The far end of the adjacent car does not see the same vertical lift and is not able to clear centre pin due to car body pitching.

The adjacent car rollover will provide vertical lift and rolling moment to the next adjacent car. As soon as the adjacent car is lifted with some roll angle, it immediately experiences a destabilizing couple from the uplift force on its draft sill and the car body weight acting down through the centre of gravity of the car body. The moment arm is the lateral distance between the center of gravity and the line of action of the vertical uplift force. Then the so called domino effect can occur.

4 SOLUTIONS

4.1 Rotary Couplers

Rotary couplers are commonly used in unit coal train service. This enables the lading to be easily unloaded by rotating the railcar without uncoupling adjacent cars. F-type couplers are used so that no relative vertical displacement occurs between the couplers during the rotation portion of the dumping process. However, the F-type coupler arrangement uses a spring loaded coupler carrier iron to provide for equivalent relative vertical displacement that occurs due to differences in coupler heights from load to empty condition, wheel diameter differences, centreplate wear, sagging springs, etc.

Therefore, F-type couplers allow relative vertical displacement between the adjacent car bodies until the coupler carrier iron springs go solid. Although the coupler carrier springs have limited stroke (~ 2 - 3 in.), there is a lever arm ratio to the coupler knuckle pulling face of roughly 2 to 1 (see Figure 23). Therefore, relative vertical displacements of 4 - 6 in. can be accommodated between car bodies even though there is no displacement at the coupler heads of the F-type couplers.

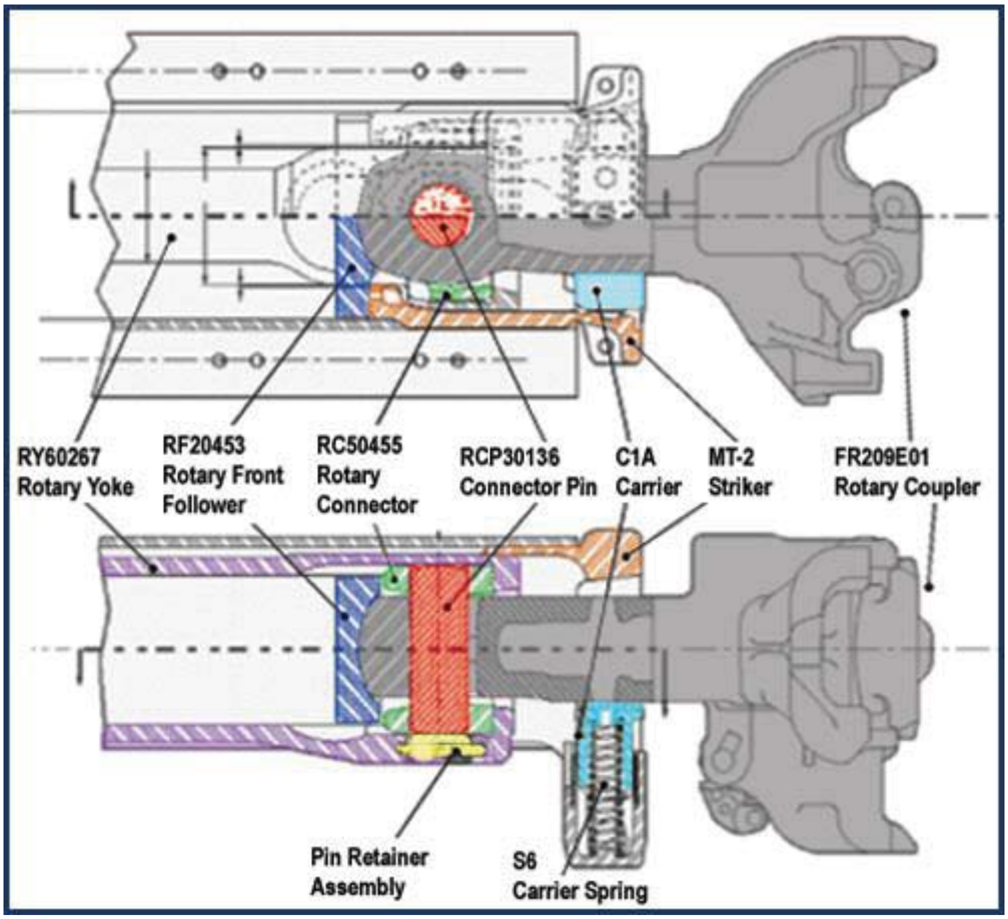

Figure 23: Details of a rotary coupler [33]

The installation of a rotary coupler at each coupling position would likely eliminate the ability of a derailing car to transmit a torque to the adjacent car through the coupling. However, a rotary coupler would still permit the transfer of horizontal and vertical force components. There are no AAR specifications for rotary couplers. Several manufacturers offer rotary couplers. Several rotary coupler connections are shown in Figure 24, Figure 25, and Figure 26.

Figure 24: Rotary coupler.

Figure 25: F-type rotary coupler (right) mated with non-rotary E-type coupler (left).

Figure 26: F-type rotary coupler (left) mated with non-rotary E-type coupler (right).

CSTT discussed the use and performance of rotary couplers with CPR's Mechanical Services Department [34]. From this discussion, CSTT determined that rotary couplers offer a few additional benefits compared to standard, non-rotary couplers, including:

- Reduced slack (F-type design), due to reduced clearance on the vertical yoke pin as compared to the horizontal yoke key used on E-type couplers
- More yaw angle available for the coupler to swing through, due to the vertical yoke pin. This provides an increased gathering range on these couplers, which helps when coupling cars and the couplers are misaligned.

However, there are logistical problems with the use of rotary couplers. If two rotary couplers are joined, they are unrestrained from rotating in their end sills because of the dynamic vibration that occurs while the train is underway. This vibration can cause the couplers to rotate such that they become out of phase with the car body orientation resulting in an increasingly inflexible connection. This distinction is made from the point of view of horizontal curving which could potentially result in derailments. Because of this, the AAR issued a bulletin [34] stating that two rotary couplers must not be coupled together, and that cars equipped with rotary couplers had to be striped (painted) at the rotary end to aid in train marshalling.

However, the problem of joined rotary couplers is still present in cases where bottom-dump coal gondolas are used in US service on trains originating from CPR's Canadian operations. These cars are equipped with rotary couplers in case they are delivered to a rotary dump unloading facility. These trains may be split apart and delivered to several facilities (groups of cars may also be split apart at some facilities) and when the cars are returned to Canada, they are unlikely to be marshalled in the same orientation as when they were originally delivered, so rotary couplers may be coupled together. CPR company rules require that cars be remarshalled once they re-enter Canada to prevent the occurrence of rotary-to-rotary couplings.

Locomotives are not equipped with rotary couplers therefore the first car that is joined to the locomotive must have a rotary coupler in its connection to the locomotive. Depending on the orientation of the rest of the unit train consist, the first car will couple to either a rotary or non-rotary coupler on the second car. To eliminate the possibility of two non-rotary couplers being joined the first car is equipped with rotary couplers at both ends, but the trailing coupler can be pinned so that it no longer rotates. This eliminates the risk of a rotary-rotary coupling between the first and second car.

In short, the logistics of dealing with rotary couplers can be difficult and time consuming. A mistake could lead to two rotaries being coupled together, possibly resulting in a derailment. CSTT discussed the use of rotary couplers with a North American car builder, and learned that rotary couplers are only used on three different car types: coal cars (some are equipped for both rotary and bottom dumping), ore cars, and high-side wood chip cars. Coal and ore cars are usually moved in unit trains but tank and wood chip cars are not, so the logistical problem of safely coupling two such cars exists. If cars are remarshalled at intermediate yards (whether in Canada or as part of an international move), an industry-wide planning mechanism would need to be put into place to prevent any undesirable couplings between tank cars and other rotary-equipped cars.

Nonetheless, use of rotary couplers could potentially reduce the possibility of multiple car rollovers if applied to tank cars. For unit train tank car service, a single rotary coupler per car would suffice. For other service, where tank cars may be coupled to cars not having rotary couplers, the tank cars would need to have a rotary coupler at each end of the car.

Since tank cars will never be intentionally rotated, the problem of unintentional coupler rotation due to vibration while the train is underway could be addressed by restraining the coupler against rotation by a weak pin or key passed horizontally through the drawbar. If a moment of sufficient magnitude were to occur, it would break the key, allowing the drawbar to rotate freely. Broken keys could be identified during train inspections.

The use of a rotary coupler as one of the two couplings between adjacent cars has both benefits and disadvantages. The benefit is that the car body suffering the rollover risk cannot apply a large rollover moment to adjacent cars, making it more difficult to propagate a rollover derailment. On the other hand, the disadvantage of having a rotary coupler at the coupling intersection is that the first car suffering a risk of overturning cannot benefit from the extra stabilizing roll moment available from adjacent cars through their couplers, and thus a single car can be more easily overturned.

However, use of rotary couplers will not be a guarantee against car body rollovers. In unit trains operating in rotary dumper service, F-type interlocking couplers are used. One of the two couplers at the coupling intersection between cars is a rotary coupler. Nevertheless, in these unit trains, several cars may derail in a train derailment, and roll over sequentially without uncoupling. An example of this is shown in Figure 27. The cars shown are rotary dump gondolas, used for coal service on a narrow gauge South American railway. As a result of a sideswipe collision at the entrance to a mainline siding, many (approximately fifteen) loaded cars overturned. These cars use F-type couplers with one rotary connection per car. In addition, the trucks are locked to the car body and the wheelsets are locked to the trucks.

Figure 27: Rotary couplers used in coal cars in South America.

If a rake of tank cars were to be joined with rotary couplers and one car were to roll over (yet remain coupled), it is important that the air hoses be routed such that they would break apart and cause the train to initiate emergency braking. If the hoses did not break apart, it seems likely that the end of the car body bolster would dig into the ties and ballast, creating a large draft force on the train. If the air hoses had not already separated, the draft force would have to be large enough for the train crew to feel it, or large enough to break a coupler knuckle (which would initiate emergency braking because of air hose separation at the broken knuckle).

There should be no issues with structural integrity with rotary couplers with double shelves, as the head of the coupler is an F-type design that is currently used, therefore the coupler head design need not change.

4.2 Locking Centre Pin

Typically, the bolster centre pins are not keyed to the car body or trucks, which is evident from Figure 16 and Figure 17. By keying or locking the centre pin to the truck and car body bolsters, additional resistance to rollover can be obtained. Note that the trucks would remain attached to the car bodies as in Figure 27. The weight of the trucks increases the stabilizing moment on the car body, thus making a rollover derailment more difficult. Once the car body has rolled sufficiently to pull up against the locking centre pin, no further relative roll angle between the car body and the truck bolster is possible. The instantaneous centre of rotation for the system is now about the wheel/rail contact points, and any additional roll of the system requires that the wheels on the other side of the truck lift off the rail.

The peak rollover resistance that can be obtained by locking the centre pin to the car body and the trucks has been calculated in Appendix A to be 1,984,000 in-lbs when the wheelsets are free and 2,305,000 in-lbs for when the wheelsets are locked to the trucks. The rollover resistance obtainable using a locking centre pin is compared with not using a locking centre pin in Table 4 and Table 5.

Table 4: Benefit of locking centre pins on rollover resistance of empty tank car – wheelsets not locked to trucks

	Rollover Resistance (in-lbs)		
	No Locking Centre Pin	**Locking Centre Pin**	**% Increase in Rollover Resistance From Locking Centre Pin**
Roller Side Bearings	1,378,000	1,984,000	44 %
Long Travel Constant Contact Side Bearings	1,324,000	1,984,000	50%

Table 5: Benefit of locking centre pins on rollover resistance of empty tank car – wheelsets locked to trucks

	Rollover Resistance (in-lbs)		
	No Locking Centre Pin	**Locking Centre Pin**	**% Increase in Rollover Resistance From Locking Centre Pin**
Roller Side Bearings	1,378,000	2,305,000	67%
Long Travel Constant Contact Side Bearings	1,324,000	2,305,000	74%

In order to roll off the tracks with a locked centre pin, the tank car must roll about the wheel/rail contact point position. This ultimately increases the total rollover resistance, although as seen in Figure 27, using a locking centre pin does not eliminate the possibility of multiple car rollovers. Note that the case shown in Figure 27 involved narrow (36 in.) gauge track, meaning there would be a smaller stabilizing moment compared to standard gauge track.

The use of locking centre pins is not a new concept, but it is often not used. Its use is not mandated and in the event of a train derailment, it is generally preferred to have the truck disengaged from the car body for the post-derailment clean-up process. In particular, locked centre pins are not used on tank cars. FRA representative felt that locking the trucks to the car bodies with the truck bolster pin might offer a near term solution to the problem of tank car rollovers [30].

4.3 Increased Shelf Height

In order to reduce the propensity for a derailing tank car to lift the near end of an adjacent tank car, and thus lead to a multiple tank car rollover derailment, an increase in the height of the top and bottom shelves may be a possible remedy.

The relative vertical displacement between couplers until the shelves contact the knuckles has an important effect on the transmission of the vertical force from one car to the next. At the current shelf height on an E-type coupler, the vertical force will start to be transmitted to the adjacent car after reaching a vertical displacement of 6.375 in. of the coupler on the derailing car (Figure 20). Two F-type couplers will have no allowable vertical movement between them, with or without shelves due to the interlocking design of the coupler heads. However, vertical car movement of the adjacent car bodies is accommodated by the spring supported coupler carrier irons used with F-type couplers.

By adjusting the height of the shelves on E-type couplers, one can essentially control when the vertical force will be transmitted between adjacent car bodies. With the shelves high enough (not considering practical limitations) it is theoretically possible that no transfer of vertical force would occur, and thus reduce the likelihood of the propagation of the rollover derailment from one car to the next.

Potentially this delay or elimination in the transmission of vertical force might prevent the car from lifting off the centre pin. This would depend entirely on the dimensions of the shelves. However, the dimensions of the shelves will be limited by practical considerations.

The derailment shown in Figure 27 was of a coal train using rotary F-type couplers. The rotary function allows no transfer of moment at the coupler between the two car bodies.

The F-type coupler allows for no relative vertical displacement between two mated coupler heads. Therefore, the vertical motion of a derailing car would propagate to an adjacent car much sooner than it would in the case of two E-type couplers with double shelves. This derailment provides further evidence of the importance of the transfer of vertical motion in rollover derailment events.

This increase in height would result in a reduced vertical translation of the near end of adjacent cars. However, the height of the shelves has practical limitations based on the height of the knuckle. The couplers may be susceptible to disengagement if the distance between the top of the knuckle and the top shelf is greater than the height of the knuckle (and similarly for the bottom side). Another practical consideration is the required clearances between the bottom shelf and the top of the railhead.

By combining the use of the rotary coupler principle to a double shelf coupler, (such a device is not currently commercially available) but with increased height shelves, the ability of a derailing car to derail an adjacent car through vertical uplift force and moment applied through the coupling can be greatly reduced. Unfortunately, this idea also works in a potentially detrimental manner. While the ability of a rollover car to derail adjacent cars is reduced, the ability of the adjacent cars to resist and prevent the rollover of the initiating car through the transmission of those forces and moments is also reduced. The combination of a rotary coupler function with increased shelf height will only have an effect for double shelf E-type couplers. For two mated F-type coupler heads, the change in shelf height will have no beneficial effect, as there will be no relative vertical movement at the coupling faces. The validity of these improvements needs to be confirmed and quantified by physical testing.

5 ECONOMIC ANALYSIS

In this section, an economic analysis is presented on the implementation of each of the two proposed solutions discussed in this report to mitigate multiple tank car rollovers. The first proposed solution is the use of rotary couplers with double shelves. The second is the increase in height of the top shelf of the double shelf couplers currently used. All dollar amounts are in US currency.

While the cost of each of the two options is different, the general steps involved in the replacement of the couplers are similar. In general, the steps involved are:

1. New coupler(s) purchased.
2. Car (or string of cars) pulled out of revenue service
3. Car sits idle in yard.
4. Car brought into shop.
5. Coupler replaced.
6. Car sits idle in yard.
7. Car returned to revenue service.

8. Old coupler reclaimed for scrap value.

5.1 Rotary Couplers

The economic analysis of the rotary coupler option assumes that each car initially has double shelf couplers. Since the rotary couplers will not be actually used in rotary dump service, they can be mated with either E-type or F-type couplers. Therefore, only one coupler needs to be replaced on each unit train car. For regular service, where tank cars may be coupled to cars that do not have rotary couplers, the tank cars would therefore need to have a rotary coupler at each end of the car. Table 6 shows the cost for replacing one coupler on each tank car of a unit train with a double shelf rotary coupler.

Table 6: Economic analysis of installation of double shelf rotary couplers (per car).

Item	Time	Cost/Time	Total Cost
Regular Rotary Coupler & Yoke ($) (x1)			$ 2,012
Premium for Shelves ($)			$ 128
Labour Rate ($/hr)		$ 95	
Labour (hrs)	1.4		
Labour Cost ($)			$ 133
In and Out Cost ($)			$ 1,000
Coupler Salvage Value ($)			-$ 72
Loss of Revenue/Day ($/day)		$ 261	
Number of Days out of Revenue Service (days)	3		
Loss of Revenue($)			$ 783
TOTAL COST / CAR ($)			**$ 3,983**

The price of the rotary coupler is assumed to be the cost obtained from a price list from one manufacturer of rotary couplers, plus a premium for double shelves. The premium is assumed to be the difference in price between an F-type coupler with no shelves and an F-type coupler with double shelves. The premium is intended to reflect the cost of adding the shelves to the coupler.

The labour rate was taken from the AAR Office Manual of Interchange Rules. The number of labour hours to replace each coupler was also given in the Office Manual of Interchange Rules. The total labour cost is simply the labour rate multiplied by the estimated labour hours.

It is assumed that the replacement of couplers will take place in a shop. It may be possible for the replacement to be done in a yard, but this may present safety issues. The "in and out" cost was assumed to be the cost to get the car in and out of the shop to do the replacement of the couplers. The salvage value of the couplers is calculated based on the weight of couplers and knuckles, and yoke (for rotary couplers) multiplied by the steel credit from the Office Manual of Interchange Rules.

The loss of revenue that occurs during replacement of couplers has been calculated as follows:

$$\text{Loss of Revenue} = \frac{\text{Total Freight Operating Revenues}}{\text{Total Number of Freight Cars in Service}} * \frac{\text{Number of Days out of Service}}{365}$$

The total freight operating revenues and total number of freight cars in service are taken from the Statistics Canada publication "Rail in Canada 2006" [35]. Data for the year 2006 was used. The number of days out of revenue service was assumed to be three days, and is intended to account for transportation time to and from the car shop, time to replace coupler(s) and time that the car sits idle at the shop.

The total cost per car calculated in this manner comes to $3,983. This cost however, is intended to reflect the worst case, when a car is brought into the maintenance facility solely for the installation of a double shelf rotary coupler. If the installation occurs while the car is in the maintenance facility for other issues, the total cost can be greatly reduced as the in and out cost and much of the loss of revenue can be attributed to the initial reason for bringing the car to the maintenance facility.

The cost could also be reduced if the removed coupler could have a service life in another application, and would not need to be sold only for salvage value. If the couplers were replaced only at the end of their service life, most of the total cost of replacement could be attributed to the end of the service life. Thus, only the premium of the increased heights of coupler shelves could be attributed to this replacement. Therefore if this replacement were only required for new couplers with existing couplers "grandfathered" until the end of their useful life, the cost would be significantly reduced.

5.2 Increased Top Shelf Height

The cost of implementing double shelf couplers with increased shelf height is calculated is the same manner as the previous implementation of double shelf rotary couplers. In this case, however, two couplers per car need to be replaced and this is reflected in the labour cost.

The premium or cost of increasing the height is assumed to be twice the premium for double shelves, calculated previously ($64 per shelf). In addition, the premium now is applied to two couplers. However, the cost of the original shelves is already included in the cost of the coupler, therefore the premium to increase the shelf height for two couplers is assumed to be $256 (4 x $64). The economic analysis is presented in Table 7.

Table 7: Economic analysis of installation of couplers with increased shelf heights (per car).

Item	Time	Cost/Time	Total Cost
Regular Top Shelf Coupler (F) & Yoke ($) (x2)			$ 3,136.00
Premium for Shelves ($)			$ 256
Labour Rate ($/hr)		$ 95	
Labour (hrs)	2		
Labour Cost ($)			$ 190
In and Out Cost ($)			$ 1,000
Coupler Salvage Value ($)			-$ 144
Loss of Revenue/Day ($/day)		$ 261	
Number of Days out of Revenue Service (days)	3		
Loss of Revenue($)			$ 783
TOTAL COST / CAR ($)			**$ 5,220**

The total cost is $5,220 per car. This cost can be significantly reduced if the replacements are done when the car is already in the shop for other repairs or scheduled maintenance. Therefore, the in and out cost and much of the loss of revenue could be attributed to the other repair.

If the couplers were replaced only at the end of their service life, most of the total cost could be attributed to this reason. Thus, only the premium of the increased heights of coupler shelves could be ascribed to this replacement. If this replacement were only required for new couplers, the incremental cost would be reduced to a few hundred dollars per car.

6 CONCLUSIONS

The importance of the safe transportation of dangerous goods by rail cannot be overstated. The introduction of double shelf couplers and head shields has reduced incidents of tank car puncture during derailments. However, due to the designed feature of double shelf couplers to remain coupled during a derailment one notable effect has been an increase in the size of each derailment where double shelf couplers are involved. Whereas derailments involving standard couplers might only involve one or a few cars, the use of double shelf couplers leads to sometimes many additional cars being derailing and rolling over. Empty tank cars are particularly susceptible to this problem.

Double shelf couplers have been successful in preventing tank car punctures caused by vertical coupler disengagement during derailments. However, since the introduction of double shelf couplers on tank cars there have been many documented cases where the derailment of a small number of tank cars precipitated the rollover of many more adjacent coupled tank cars. Due to the engagement of the double shelf couplers, large torsional forces can be transferred through the couplers and when one car overturns; adjacent cars can be caused to overturn. This can result in larger derailments.

After reviewing derailment reports and photographs, two mechanisms have been identified that likely work in combination to cause multiple car derailments. The first mechanism investigated is rollover due to a moment transferred between couplers. The second mechanism is dominated by the transfer of vertical motion from a car that is derailing to the adjacent car.

The presumed path of the coupler on the derailing car implies the development of not only a moment on the adjacent coupler, but also vertical and horizontal forces. The combination of the two mechanisms (moment and vertical) acting together results in vertical lift at the near end of the adjacent car which reduces the required moment to roll the car body off the near trucks without damaging the centre pin.

Rollover resistance calculations were performed for two cases: roller side bearings and long travel constant contact side bearings. The results for each case are presented. It is evident that the tank car body has a very large self-stabilizing moment once the body weight is carried on the side bearings. The peak rollover moment resistance of an empty tank car were calculated to be 1,378,000 in-lbs and 1,324,000 in-lbs for roller side bearings and long travel constant contact side bearings, respectively.

The installation of a rotary coupler at each coupling would likely eliminate the ability of a derailing car to transmit a moment to the adjacent car through the coupling, but would still permit the transfer of horizontal and vertical forces. In order to reduce the propensity for a derailing tank car to lift the near end of an adjacent tank car and lead to a multiple tank car rollover derailment, the height of the top and bottom shelves could be increased.

Another means to obtain additional rollover resistance is by locking the centre pin to the truck and body bolsters and by locking the wheelsets to the truck. The weight of the trucks increases the stabilizing moment on the car body, thus making a rollover derailment more difficult. In order to roll off the tracks with a locked centre pin, the tank car must roll about the wheel/rail contact point position. This ultimately increases the total rollover moment resistance of an empty tank car by 44 – 74 %.

The economics of using rotary couplers with double shelves and using double shelves with increased heights has been evaluated. An economic assessment of the worst case cost for the implementation of double shelf rotary couplers and non-rotary increased height double shelf couplers was estimated to be $3,983 and $5,220 per car, respectively. These costs would be reduced substantially if the replacements were made when the car was in the shop for other reasons.

This study has shed some light on the problem of multiple tank car rollover derailments. Based on data from various sources two main mechanisms have been identified; moment transfer and vertical force transfer through couplings. Several remedies to control or limit these mechanisms have been proposed. The next stage should involve the evaluation of these proposed solutions through full scale physical testing.

7 FUTURE WORK

CSTT proposes to perform two types of physical testing (quasi-static and dynamic), on three different coupler arrangements. Testing will require a rake of three coupled empty tank cars. The cars will be fitted with instrumentation to measure:

- Vertical load applied to the coupled drawbars
- Moment applied to the coupled drawbars
- Secondary suspension deflections
- Car body roll angle with respect to ground
- Vertical and lateral displacements of car body with respect to truck bolster
- Coupler roll angle with respect to car body

The first test will be the baseline test, and will use standard double shelf E-type couplers at all coupled interfaces.

The second test will use a rotary double shelf E-type coupled to a standard double shelf E-type coupler at both coupled interfaces. Since this coupler is not commercially available, it must be fabricated by modifying an existing coupler.

The third test will use a modified double shelf coupler without any rotary capability. The modification will include an increase in shelf heights and an increased knuckle (pulling face) height to prevent disengagement. This type of coupler will be used on both sides of both coupled interfaces in the rake of cars. The modified design will be a fabricated coupler suitable only for proof-of-concept testing.

The same rake of three tank cars and the same coupler arrangements would be used to perform three dynamic tests. However, the end car would be rolled off of its trucks and allowed to fall such that it can pick up the second car and the second can pick up the third, causing them to both roll off of their trucks, thus simulating a domino rollover similar to the one that occurred at Clara City, Minnesota (Figure 3 and Figure 4).

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APPENDIX A – MOMENT RESISTANCE OF EMPTY TANK CAR TO ROLLOVER

This appendix contains the calculations to assess the moment resistance to rollover for an empty tank car. Two cases are presented. The first considers a tank car equipped with roller side bearings and the second assumes the tank car is equipped with long travel constant contact side bearings. In addition, the rollover resistance of an empty tank car with a locking centre pin is presented. The calculations presented are for standard gauge track.

A.1 Empty Tank Car with Roller Side Bearings

Table A defines the parameters and relevant values used in the calculations for an empty tank car with roller side bearings.

Table A1: Definition of parameters for moment resistance (roller side bearings).

Parameter	Definition	Units	Value
K_v	Vertical stiffness of a spring group	lbs/in.	23,688
b	Lateral distance from truck centre line to spring seat centre line	in.	39.8
h	Vertical distance between the centre of gravity of the car body and the top of the spring nest	in.	46.62
c	Side bearing clearance	in.	0.25
s	Distance from truck longitudinal centre line to side bearing centre line	in.	25
W	Weight of car body	lbs	57,500
x	Distance from centre plate centreline to start of bevelled edge of centre plate	in.	6.5
θ_{1j}	Truck bolster roll angle (with respect to horizontal) at stage j	rad	
θ_{2j}	Car body roll angle (with respect to horizontal) at stage j	rad	
M_j	External body moment applied at stage j	in-lbs	
F_{CB}	The total force carried by the centre bowls on the two trucks carrying the car body	lbs	

Stage 0:

The car body is sitting level on its centre plate in the centre bowl, and no external moment is applied. The tank car is illustrated in this state in Figure A1.

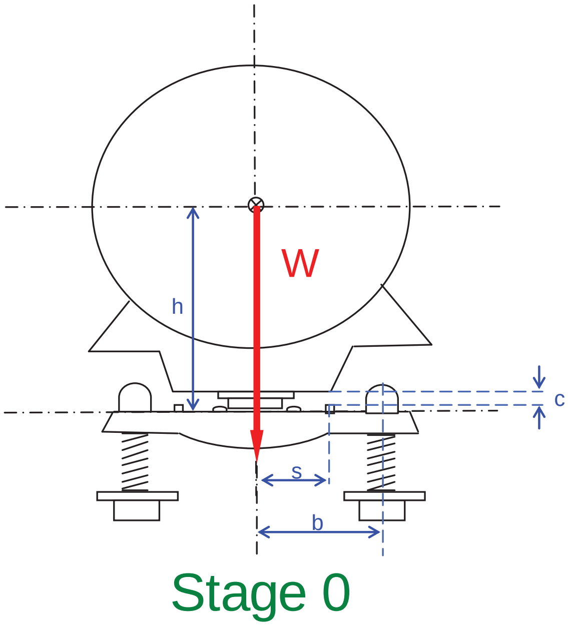

Figure A1: Tank car rollover stage 0 (roller side bearings).

Stage 1:

Due to an externally applied moment on the tank car body, the body roll angle increases until the centre plate is just about to rock onto its bevelled edge. Although the body does not roll with respect to the truck bolster, the bolster rolls on the suspension springs because the distribution of the car body weight on the centre plate is moving towards one edge of the plate. The tank car position is shown in Figure A2. Details of the centre plate, showing the distance to the bevelled edge are given in Figure A3.

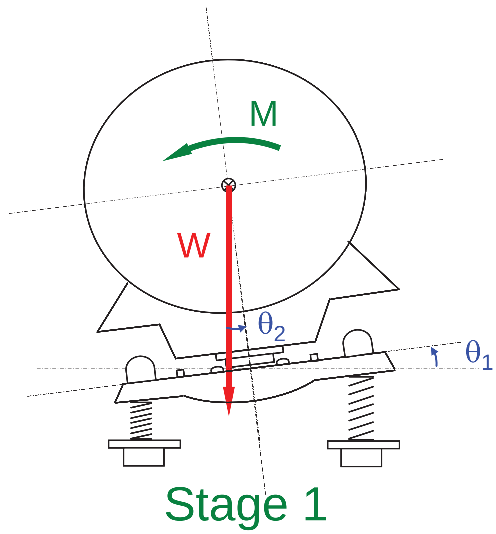

Figure A2: Tank car rollover stage 1 (roller side bearings).

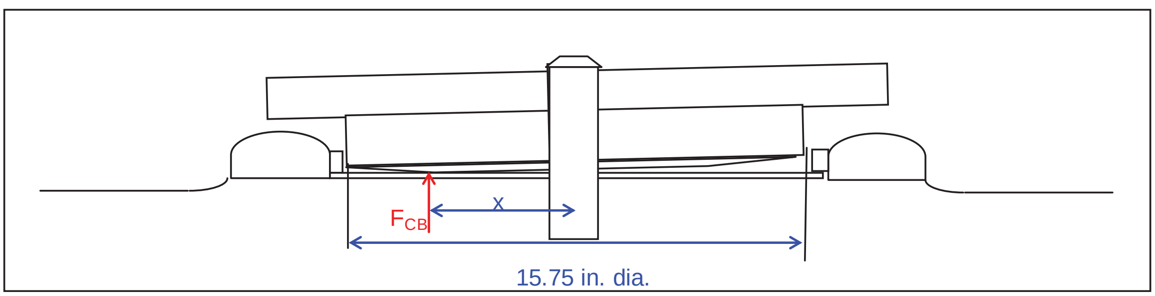

Figure A3: Details of centre bowl and centre plate.

The vertical force carried by the spring groups in the left (R_1) and right sides (R_2) of each pair of trucks on a car are given by

$$R_1 = \frac{W}{2}\left(1 + \frac{x}{b}\right) \tag{A1}$$

$$R_2 = \frac{W}{2}\left(1 - \frac{x}{b}\right) \tag{A2}$$

The deflections in the springs on each side of the trucks are given by

$$\delta_1 = \frac{R_1}{2K_v} = \frac{W}{4K_v}\left(1 + \frac{x}{b}\right) \quad \textbf{(A3)}$$

$$\delta_2 = \frac{R_2}{2K_v} = \frac{W}{4K_v}\left(1 - \frac{x}{b}\right) \quad \textbf{(A4)}$$

The roll angle of the truck bolsters and the car body are equal at this stage and can be computed from

$$\theta_{21} = \theta_{11} = \frac{\delta_1 - \delta_2}{2b} = \frac{Wx}{4K_v b^2} \quad \textbf{(A5)}$$

Finally, the moment that resists rollover of the car body at the end of stage 1 is given by

$$M_1 = W\left(x - h\theta_{21}\right) \quad \textbf{(A6)}$$

<u>Stage 2:</u>

The car body roll angle increases until the body just contacts the side bearings (25 in. away from truck centre) without loading them. Note that the truck bolster does not undergo any additional roll during this stage, because the load-bearing contact point (the bevelled edge of the centre plate) does not change position. However, the body rolls with respect to the truck bolster to use up the side bearing clearance. This stage is illustrated in Figure A4.

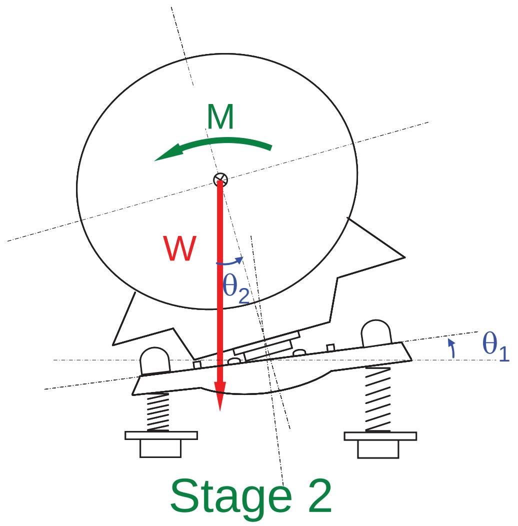

Figure A4: Tank car rollover stage 2 (roller side bearings).

The roll angles of the truck bolster and the car body can be found by

$$\theta_{12} = \theta_{11} = \frac{Wx}{4K_v b^2} \qquad \textbf{(A7)}$$

$$\theta_{22} = \theta_{12} + \tan^{-1}\left(\frac{c}{s-x}\right) \qquad \textbf{(A8)}$$

The moment which resists rollover at the end of stage 2 is calculated as

$$M_2 = W\left(x - h\theta_{22}\right) \qquad \textbf{(A9)}$$

Stage 3:

At this stage, the body and truck bolster roll together until the bevelled edge of the centre plate has just separated from the centre bowl. All of the car body weight is now carried on the side bearings. Therefore, the weight on the side bearings increases from no load to the full load of the car body weight. This stage is shown in Figure A5.

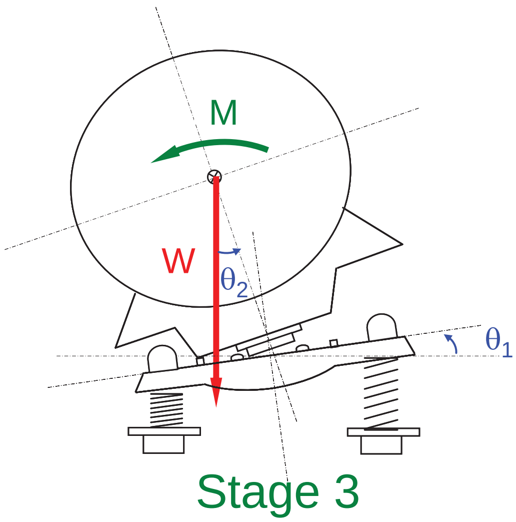

Figure A5: Tank car rollover stage 3 (roller side bearings).

The roll angles of the truck bolster and the car body can be found by

$$\theta_{13} = \frac{Ws}{4K_v b^2} \tag{A10}$$

$$\theta_{23} = \theta_{13} + \tan^{-1}\left(\frac{c}{s-x}\right) \tag{A11}$$

The moment which resists rollover at the end of stage 3 is calculated as

$$M_3 = W\left(s - h\theta_{23}\right) \tag{A12}$$

Stage 4:

The body continues to roll under the externally applied moment about the side bearings until the weight vector of the car body passes through the side bearing. At this point, the car body is balanced on the side bearings in a meta-stable position. The applied moment required to keep it in this position is zero; if the car body were perturbed from this position in either direction, it would roll off the truck or back on to the truck. Figure A6 illustrates this stage.

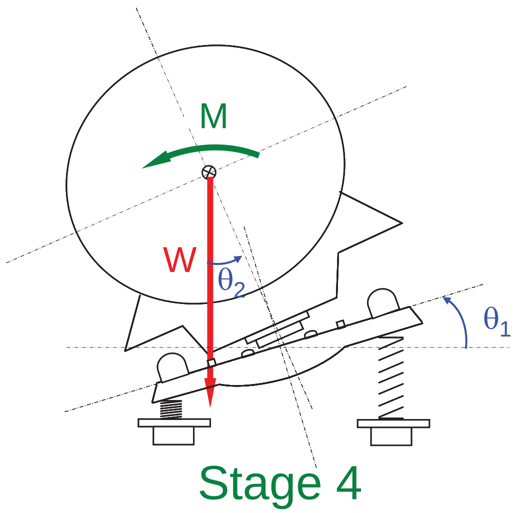

Figure A6: Tank car rollover stage 4 (roller side bearings).

The roll angles of the truck bolster and the car body can be found by

$$\theta_{14} = \theta_{13} = \frac{Ws}{4K_v b^2} \tag{A13}$$

$$\theta_{24} = \tan^{-1}\left(\frac{h}{s}\right) \tag{A14}$$

The rollover resisting moment at the end of stage 4 is zero.

$$M_4 = 0 \tag{A15}$$

Stage 5:

The car body rolls off the trucks onto its side on the ground. Before striking the ground, the car body bolster contacts the side frame.

A.2 Empty Tank Car with Long Travel Constant Contact Side Bearings

Table A2 defines the parameters and relevant values used in the calculations for an empty tank car with long travel constant contact side bearings (CCSB).

Table A2: Definition of parameters for moment resistance (long travel constant contact side bearings)

Parameter	Definition	Units	Value
K_v	Vertical stiffness of a spring group	lbs/in.	23,688
b	Lateral distance from truck centre line to spring seat centre line	in.	39.8
h	Vertical distance between the centre of gravity of the car body and the top of the spring nest	in.	46.62
t_{SB}	Constant contact side bearing travel	in.	0.625
s	Distance from longitudinal truck centre line to side bearing centre	in.	25
W	Weight of car body	lbs	57,500
x	Distance from centre plate centre line to start of bevelled edge	in.	6.5
θ_{1j}	Truck bolster roll angle at stage j	rad	
θ_{2j}	Car body roll angle at stage j	rad	
M_j	External body moment applied at stage j	in-lbs	
F_{CB}	The total force carried by the centre bowls on the two trucks carrying the car body	lbs	
F_{SB1}	The total force carried by side bearings on the left hand side of both trucks	lbs	
F_{SB2}	The total force carried by side bearings on the right hand side of both trucks	lbs	

Stage 0:

The car body is sitting level on its centre plate in the centre bowl, and no external moment is applied. Contact is made with side bearings. Figure A7 shows the tank car in this position. The vertical force at each centreplate is W/2 minus 9,000 lbs (4,500 lbs preload in each side bearing), due to the vertical preload force from the side bearings at nominal conditions.

$$F_{CB} = W - F_{SB1} - F_{SB2} \tag{A16}$$

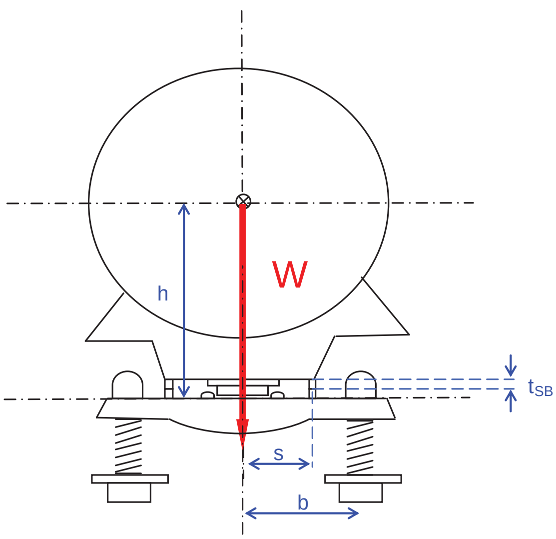

Figure A7: Tank car rollover stage 0 (constant contact side bearings).

Stage 1:

Due to an externally applied moment on the tank car body, the body roll angle increases until the centre plate is just about to rock onto its bevelled edge. Although the body does not roll with respect to the truck bolster, the bolster rolls on the suspension springs because the distribution of the car body weight on the centre plate is moving towards one bevelled edge of the plate. This configuration is shown in Figure A8. At this point, there is no relative roll angle between the car body and the truck bolster therefore, the force in each side bearing is equal to the preload (4,500 lbs).

$$F_{SB1} = F_{SB2} = 2 * 4{,}500 lbs = 9{,}000 lbs \tag{A17}$$

The load on the bevelled edge of each then becomes

$$F_{CB} = 57{,}500 - 9{,}000 - 9{,}000 = 39{,}500\,lbs \tag{A18}$$

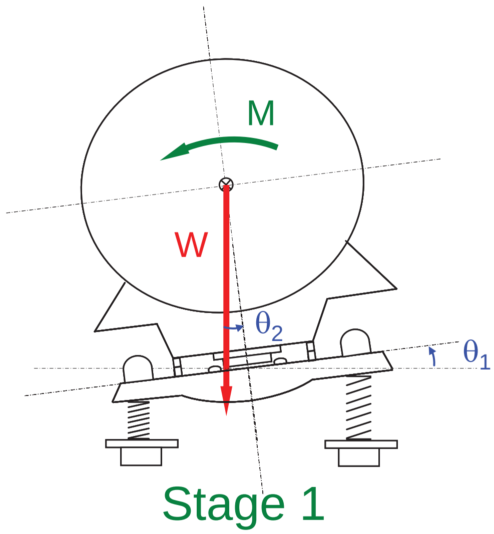

Figure A8: Tank car rollover stage 1 (long travel constant contact side bearings).

The vertical force carried by the springs in the left (R_1) and right sides (R_2) of both trucks are given by

$$R_1 = \frac{(b+s)F_{SB1} + (b+x)F_{CB} + (b-s)F_{SB2}}{2b} \tag{A19}$$

$$R_2 = \frac{(b-s)F_{SB1} + (b-x)F_{CB} + (b+s)F_{SB2}}{2b} \tag{A20}$$

The deflections in the springs on each side of the trucks are given by

$$\delta_1 = \frac{(b+s)F_{SB1} + (b+x)F_{CB} + (b-s)F_{SB2}}{4bK_V} \tag{A21}$$

$$\delta_2 = \frac{(b-s)F_{SB1} + (b-x)F_{CB} + (b+s)F_{SB2}}{4bK_V} \quad \textbf{(A22)}$$

The roll of the truck bolster and the car body are equal at this stage and can be computed from

$$\theta_{11} = \theta_{21} = \frac{xF_{CB}}{4b^2K_V} \quad \textbf{(A23)}$$

Finally, the moment that resists rollover of the car body at the end of stage 1 is given by

$$M_1 = F_{CB}x - W\theta_{21}h \quad \textbf{(A24)}$$

<u>Stage 2:</u>

Long travel side bearings could have three different deflection characteristics as they stroke from free height to solid conditions. Since the car body's centre plate bevelled edge rocks on and maintains contact with the truck bolster's centre bowl during side bearing closure, the moment arms for both side bearings are not equal. The one on the closing side being (25 – 6.5 =) 18.5 in. from the bevelled edge and the one of the extending side being (25 + 6.5 =) 31.5 in. from the bevelled edge. This provides a multiplication ratio of 1.7 between the side bearing extension and compression strokes. Therefore, there are three possibilities:

1. As one side bearing just goes solid (5/8 in. stroke to solid for the long travel CCSB), the other side bearing just extends to its free height.
2. As one side bearing goes solid, the other side bearing has not yet reached its free height. This is the most complex possibility, because the centreplate reaches zero load on the bevel, and the extending side bearing contributes a destabilizing moment.
3. Before one side bearing goes solid, the other side bearing reaches its free height.

Depending on the stroke characteristics of the side bearings used, one of these three conditions will occur. Data for the long travel constant contact side bearings used in this report suggest that case 3 will occur.

The car body rolls (on the bevelled edge of its centre plate) relative to the truck bolster until the CCSBs on one side of the trucks become fully unloaded. The load is thus shared by the side bearings on the side of the truck bolster that the car body is rolling towards and the centre bowl. This is illustrated in Figure A9.

$$F_{SB2} = 0 \tag{A25}$$

$$F_{SB1} = 2 * 10{,}000 lbs = 20{,}000 lbs \tag{A26}$$

$$F_{CB} = 57{,}500 lbs - 20{,}000 lbs = 37{,}500 lbs \tag{A27}$$

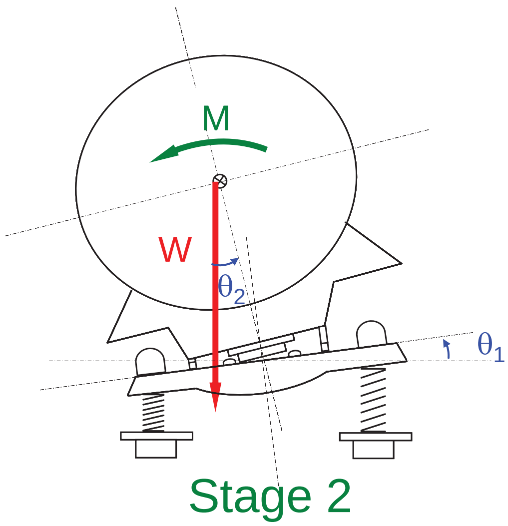

Figure A9: Tank car rollover stage 2 (long travel constant contact side bearings).

The roll angles of the truck bolster and the car body can be found by

$$\theta_{12} = \frac{sF_{SB1} + xF_{CB}}{4b^2 K_V} \tag{A28}$$

$$\theta_{22} = \theta_{12} + \tan^{-1}\left(\frac{t}{s - x}\right) \tag{A29}$$

Where t is the travel used up by the compressed side bearings when side bearings on opposite side become unloaded, ($t = 0.41\ in$).

The moment which resists rollover at the end of stage 2 is calculated as

$$M_2 = sF_{SB1} + xF_{CB} - Wh\theta_{22} \quad \textbf{(A30)}$$

Stage 3:

The car body continues to roll (on the bevelled edge of its centre plate) relative to the truck bolster until the CCSB on one side of the trucks go solid. The CCSB on the opposite side of the trucks are no longer in contact with the car body. The force carried by the side bearings in contact with the car body increases, and the force carried by the centre bowls decreases. This is illustrated in Figure A10.

$$F_{SB2} = 0 \quad \textbf{(A31)}$$

$$F_{SB1} = 2 * 18{,}000 lbs = 36{,}000 lbs \quad \textbf{(A32)}$$

$$F_{CB} = 57{,}500 lbs - 36{,}000 lbs = 21{,}500 lbs \quad \textbf{(A33)}$$

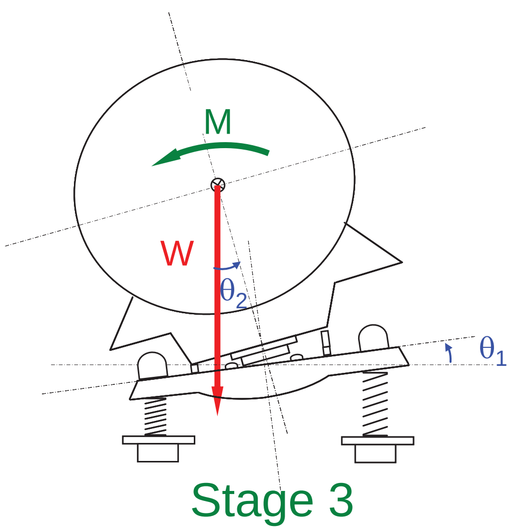

Figure A10: Tank car rollover stage 3 (long travel constant contact side bearings).

The roll angles of the truck bolster and the car body can be found by

$$\theta_{13} = \frac{sF_{SB1} + xF_{CB}}{4b^2K_V} \quad \textbf{(A34)}$$

$$\theta_{23} = \theta_{13} + \tan^{-1}\left(\frac{t_{SB}}{s-x}\right) \quad \textbf{(A35)}$$

The moment which resists rollover at the end of stage 3 is calculated as

$$M_3 = sF_{SB1} + xF_{CB} - Wh\theta_{23} \quad \textbf{(A36)}$$

<u>Stage 4:</u>

The car body and truck bolster roll together until the centre plate just separates from the truck centre bowl. The solid side bearings are now carrying the entire weight of the car body. The centreplate has just separated from the truck bolster, and is now carrying no load. Figure A11 illustrates the positions of the car body and truck bolster.

$$F_{SB1} = W \quad \textbf{(A37)}$$

$$F_{SB2} = 0 \quad \textbf{(A38)}$$

$$F_{CB} = 0 \quad \textbf{(A39)}$$

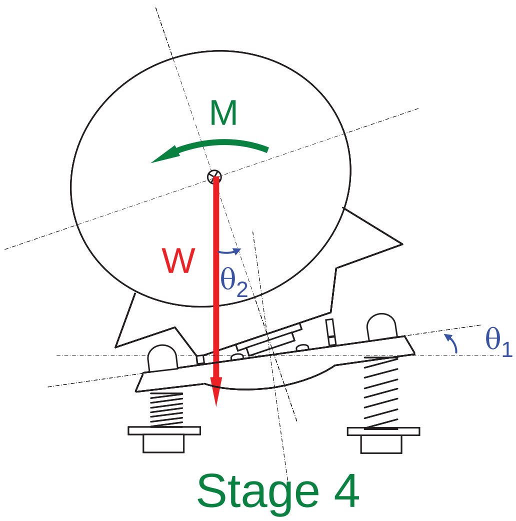

Figure A11: Tank car rollover stage 4 (long travel constant contact side bearings).

The roll angles of the truck bolster and the car body can be found by

$$\theta_{14} = \frac{Ws}{4b^2 K_V} \tag{A40}$$

$$\theta_{24} = \theta_{14} + \tan^{-1}\left(\frac{t_{SB}}{s-x}\right) \tag{A41}$$

The rollover resisting moment at the end of stage 4 is calculated as

$$M_4 = W(s - \theta_{24} h) \tag{A42}$$

<u>Stage 5:</u>

The body continues to roll under the externally applied moment about the side bearings on one side of the car until the weight vector of the car body passes through the side bearings. At this point, the car body is balanced on the side bearings in a meta-stable position. The applied moment required to keep it in this position is zero; if the car body were perturbed from this position in either direction, it would roll off the truck or back on to the truck. The tank car in this configuration is illustrated in Figure A12.

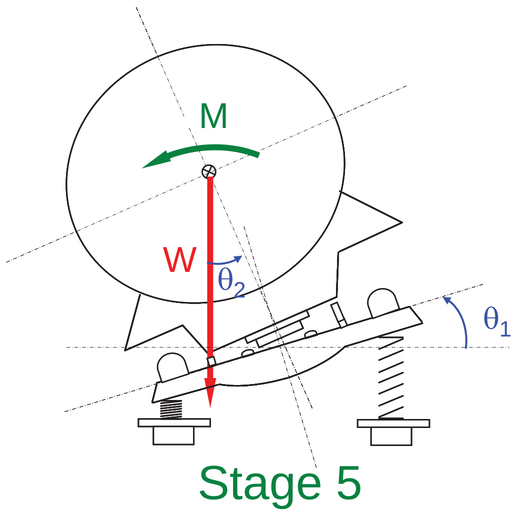

Figure A12: Tank car rollover stage 5 (long travel constant contact side bearings).

The roll angles of the truck bolster and the car body can be found by

$$\theta_{15} = \theta_{14} = \frac{Ws}{4b^2K_v} \tag{A43}$$

$$\theta_{25} = \tan^{-1}\left(\frac{h}{s}\right) \tag{A44}$$

The rollover resisting moment at the end of stage 5 is zero.

$$M_5 = 0 \tag{A45}$$

<u>Stage 6:</u>

The car body rolls off the trucks onto its side on the ground. Before striking the ground, the car body bolster contacts the side frame

A.3 Use of Locking Centre Pins

Additional rollover resistance can be obtained by locking the centre pin to the truck bolster and car body. By locking the car body and the truck together, before rollover occurs, the two bodies roll as one about the wheel/rail contact point. Further stabilizing moment can be obtained by locking the wheelsets to the trucks.

The calculations presented here assume that there is 3/16 in. vertical slack in the centre pin connection to the car body bolster and truck bolster. In other words, the maximum relative vertical displacement that can occur between the car body and the truck bolster is 3/16 in. The analysis presented here is valid for long travel constant contact side bearings and roller side bearings.

Stage 0:

The car body is sitting level on its centre plate in the centre bowl, and no external moment is applied. At this stage, the centre pin has no effect.

Stage 1:

Because of an externally applied moment on the tank car body, the body roll angle increases until the centre plate is just about to rock onto its bevelled edge. Although the body does not roll with respect to the truck bolster, the bolster rolls on the suspension springs because the distribution of the car body weight on the centre plate is moving towards one edge of the plate. The moment resistance at each stage can be calculated from equations presented in stage 1 for roller side bearings (Equation A1-A6) and long travel constant contact side bearings (Equation A17-A24).

Stage 2:

The car body rolls about the bevelled edge of the centre plate. This continues until the 3/16 in. slack in the centre pin is taken up, now the car body and truck roll as one about the wheel/rail contact point. The wheels on the other side of the car then lift up so that they no longer transmit any force to the ground. In addition, all of the weight of the car body and trucks are transmitted to the ground by the wheels that remain in contact with the rails. This state is illustrated in Figure A13.

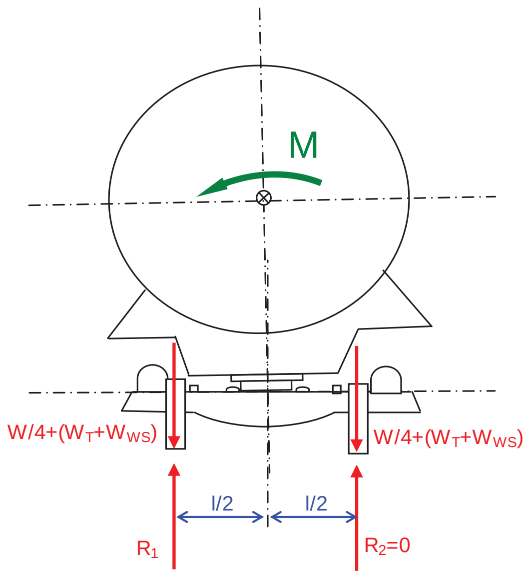

Figure A13: Free body diagram of tank car during rollover with locking centre pin.

The rollover resistance at this stage is the same for both types of side bearings. The rollover resistance is calculated from

$$M = 2R_1 \frac{l}{2} = R_1 l \qquad \textbf{(A46)}$$

Where *l* is the kinematic gauge (*l=59.5 in*.),

R_1 is the total reaction through the wheels on each truck that are still in contact with the rail

For the case where the wheelsets are free (the trucks rest on the wheelsets, but they are not fixed), R_1 is given by

$$R_1 = \frac{W}{2} + W_T \qquad \textbf{(A47)}$$

Where W_T is the weight of one truck (not including wheelsets).

$$W_T = 4{,}600\,lbs \qquad \textbf{(A48)}$$

When the wheelsets are locked to the trucks (when the trucks lift up, the wheelsets lift as well), R_1 is given by

$$R_1 = \frac{W}{2} + W_T + 2W_{WS} \tag{A49}$$

Where W_{WS} is the weight of one wheelset.

$$W_{WS} = 2{,}700\, lbs \tag{A50}$$

At this stage the peak rollover resistance occurs. The results of rollover resistance calculations at this stage are given in Table A3. These rollover resistance moments suggest that centre pins can offer a significant improvement in rollover resistance of empty tank cars.

Table A3: Rollover moment resistance with locked centre pins.

	Wheelsets Free	Wheelsets Locked
Rollover Resistance (in-lbs)	1,984,325	2,305,625

Stage 3:

The car body and trucks roll together about the wheel rail contact point, until the weight vector passes through the wheel rail contact point. At this point, the car body and trucks are balanced on the wheel rail contact point in a meta-stable position. The applied moment required to keep the system in this position is zero; if the system were perturbed from this position in either direction, it would roll off the tracks or back onto the tracks.

Therefore, the peak rollover resistance is obtained in stage 2, and thus Table A3 gives the peak rollover resistance for the two cases: wheelsets free and wheelsets locked.