Rollover of Heavy Commercial Vehicles

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16. Abstract

The state-of-the-art understanding of rollover of the commercial vehicle is reviewed. Accident statistics are presented which highlight the severity and lethal nature of rollover crashes. Physical and statistical evidence for the linkage between vehicle roll stability and the actual occurrence of rollover accidents is presented. The fundamentals of static roll stability are described in detail and then enhanced with discussion of dynamic considerations of the rollover process. The text concludes with a discussion of the evolving use of intelligent electronic systems and active vehicle control for reducing the occurrence of rollover. Appendices include a bibliography of the literature on heavy-vehicle rollover.

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1. Rollover Accidents and Vehicle Roll Stability

Rollover accidents are of special concern for commercial vehicle safety. Rollover accidents are especially violent and cause greater damage and injury than other accidents. Moreover, the relatively low roll stability of the commercial truck promotes rollover and contributes to the number of truck accidents. These ideas are generally confirmed by the accident record.[1]

Rollover and accident severity

Commercial truck rollover is strongly associated with severe injury and fatalities in highway accidents. According to data from the General Estimates System (GES)[2], rollover occurred in about 3.6 percent of U.S. truck accidents in 1995. However, from the *Truck and Bus Crash Fact Book* (T&BFB)[2] for the same year (the most resent available), rollover occurred in 7.9 percent of tow-away accidents involving trucks, 9.9 percent of injury accidents, and 12.3 percent of fatal accidents.[1][3]

In the preceding statistics, fatalities and injuries refer to anyone involved in the accident including occupants of other vehicles or pedestrians. The association of rollover with injuries to the truck driver is even stronger. Again from the 1995 T&BFB, death or incapacitating injury is about ten times more likely to occur to the truck driver in rollover accidents than in nonrollover accidents. Further, about 50 percent of truck-driver deaths and 47 percent of incapacitating injuries occurred in accidents in which the truck rolled over. (Statistics for other years are similar to those shown in figure 1.[2-4])

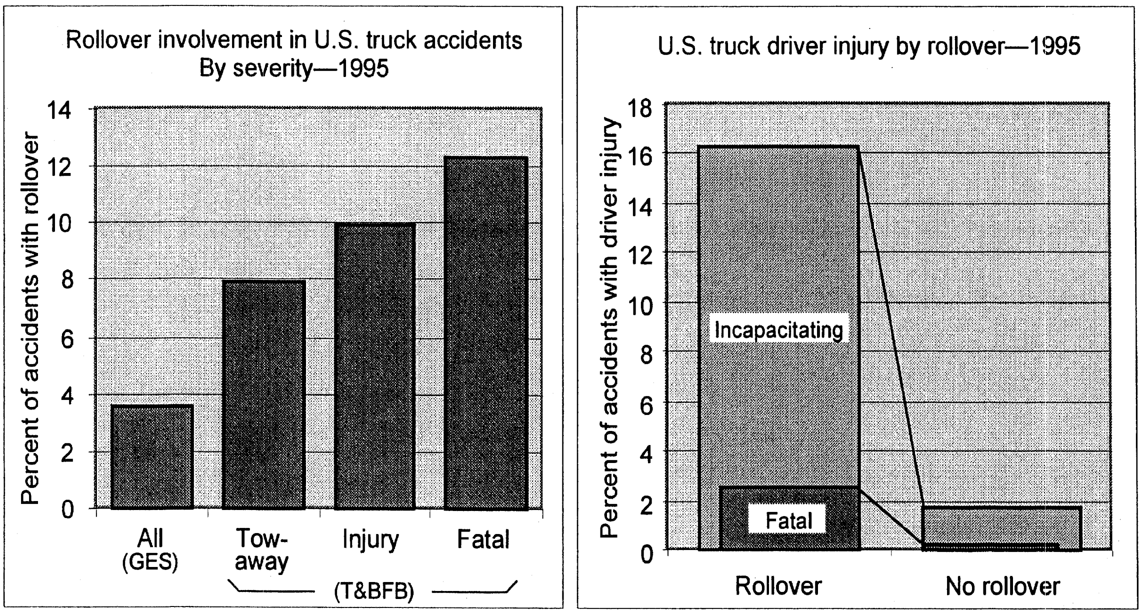

Figure 1. Rollover is strongly associated with accident severity and with serious injury to truck drivers [1]

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[1] That is, they are confirmed by the U.S. accident record. The points made in this section derive from statistical analyses of U.S. accident-data files. We would expect similar analyses of European data sources to yield similar results.

[2] See the descriptive notes on data sources in appendix A.

[3] Numbers in brackets refer to bibliographic references at the end of this text.

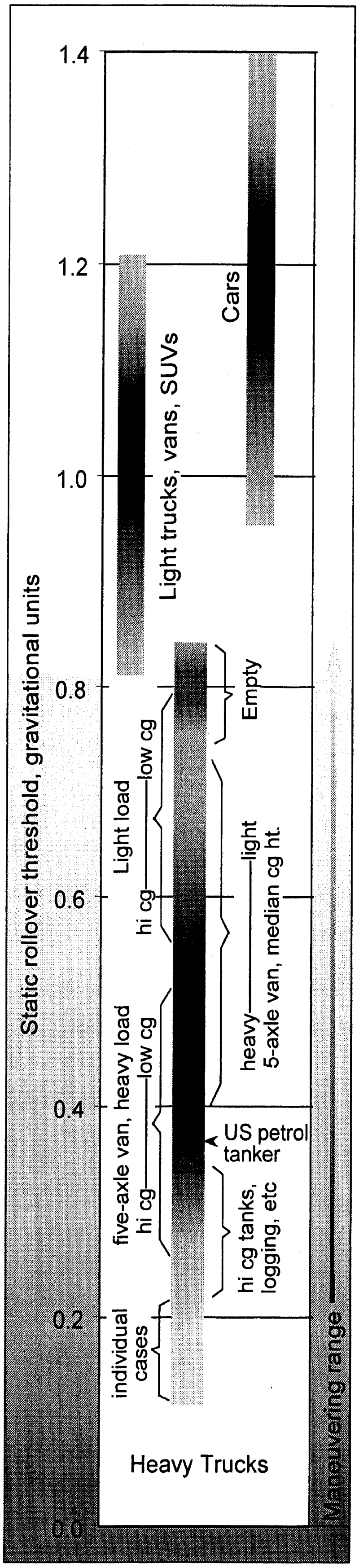

Figure 2. The rollover threshold of trucks extends deep into the maneuvering range

Roll stability and the occurrence of rollover accidents

The low level of basic roll stability of commercial trucks sets them apart from light vehicles and appears to be a significant contributing cause of truck rollover accidents. The basic measure of roll stability is the *static rollover threshold*, expressed as lateral acceleration in gravitational units (g). The rollover thresholds of passenger cars are virtually always greater than 1 g.[5] For light trucks, vans, and SUVs, this property lies in the range of 0.8 to 1.2 g.[5,6], but the rollover threshold of a loaded heavy truck often lies well below 0.5 g.

When loaded to legal gross weight, the typical U.S. five-axle tractor-van semitrailer combination has a rollover threshold perhaps as high as 0.5 g with a high-density, low center of gravity (cg) load, but as low as 0.25 g with the worst-case load—one which completely fills the volume of the trailer while also reaching legal gross weight.[7-9] The typical U.S. five-axle petroleum semitanker has a rollover threshold of about 0.35 g.[10] Rollover thresholds of common cryogenic tankers for the transport of liquefied gases are as low as 0.26 g.[11] El-Gindy and Woodrooffe found a variety of logging trucks operating in Canada to have thresholds ranging from 0.23 to 0.31 g.[12] Individual vehicles with rollover thresholds well below 0.2 g can occur occasionally.[e.g., 13][4]

Drivers regularly maneuver vehicles at well over 0.2 g. The AASHTO guidelines for highway curve design result in lateral accelerations as high as 0.17 g at the advisory speed.[21] Therefore, even a small degree of speeding beyond the advisory level will easily cause actual lateral accelerations to reach 0.25 g in everyday driving. On the other hand, tire frictional properties limit lateral acceleration on flat road surfaces to a bit less than 1 g at the very most. These two observations clearly imply that the rollover threshold of light vehicles lies above, or just marginally at, the extreme limit of the vehicles maneuvering ability, but the rollover threshold of loaded heavy trucks extends well into the "emergency" maneuvering capability of the vehicle and

[4] All these examples are estimates obtained from simulation or other calculations. However, sufficient tilt-table measurements are reported in the literature to confirm the general findings.[e.g., 14-20]

sometimes into the "normal" maneuvering range.

Nevertheless, it is relatively hard for truck drivers to perceive their proximity to rollover while driving. First of all, rollover is very much an either-or situation. It is something like walking up to a cliff with your eyes closed: even as you approach the edge, your perception is still one of walking on solid ground, until it is too late. Further, the actual rollover threshold of the commercial truck changes regularly as the load changes, so the driver may not have the chance to "get used to" the stability of his vehicle. Finally, for combinations especially, the flexible nature of the tractor frame tends to isolate the driver from the roll motions of the trailer, which might act as a cue to rollover.

These observations provide physical rationale for two safety hypotheses:

— *Heavy trucks are subject to a class of rollover accidents to which light vehicles are not susceptible, namely, rollover accidents caused directly by inadvertently operating the vehicle beyond the rollover threshold.*

— *Rollover in heavy-truck accidents is strongly related to the basic roll stability of the vehicle.*

The mass accident data support these hypotheses.

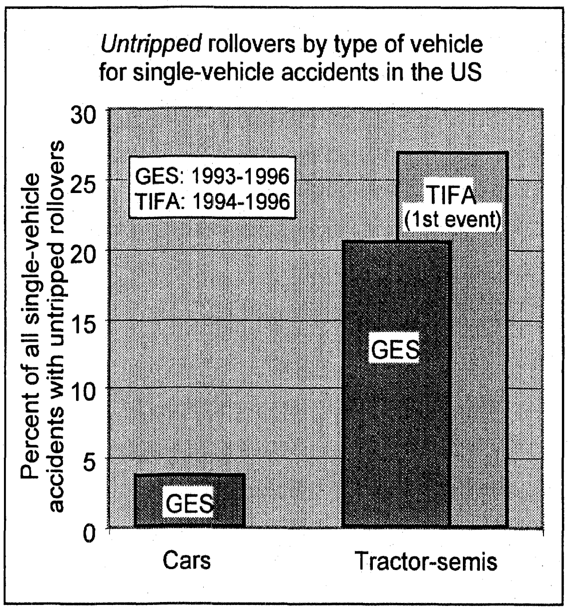

Figure 3. *Untripped* rollovers are common for tractor-semitrailer combinations but rare for cars

The accidents described in the first hypothesis could be identified as *single-vehicle accidents* in which the *first significant event* is an *untripped* rollover. According to the hypothesis, such accidents should be nearly nonexistent for passenger cars but more common for trucks.

As is often the case, the perfect accident file for such an analysis does not exist. However, GES files (which do not indicate first event) for 1993 through 1996 show that untripped rollovers account for more than 20 percent of the single-vehicle rollover accidents for tractor semitrailers, but they make up less than 4 percent of those accidents for passenger cars. Further, the Trucks In Fatal Accidents[5] files (which contain no comparable data for cars) for 1994 through 1996 show that *untripped, first-event* rollovers account for 26.8 percent of the single-vehicle rollover accidents in that file.

Accident data also confirm the second hypothesis. Figure 4 shows the results of an analysis of data from the Bureau of Motor Carrier Safety (BMCS)[5] from 1987-1991[6]. The

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[5] See the descriptive notes on data sources in appendix A.

[6] Ervin first presented analyses of this type in the1980s.[7-11] The analysis herein is limited to five-axle

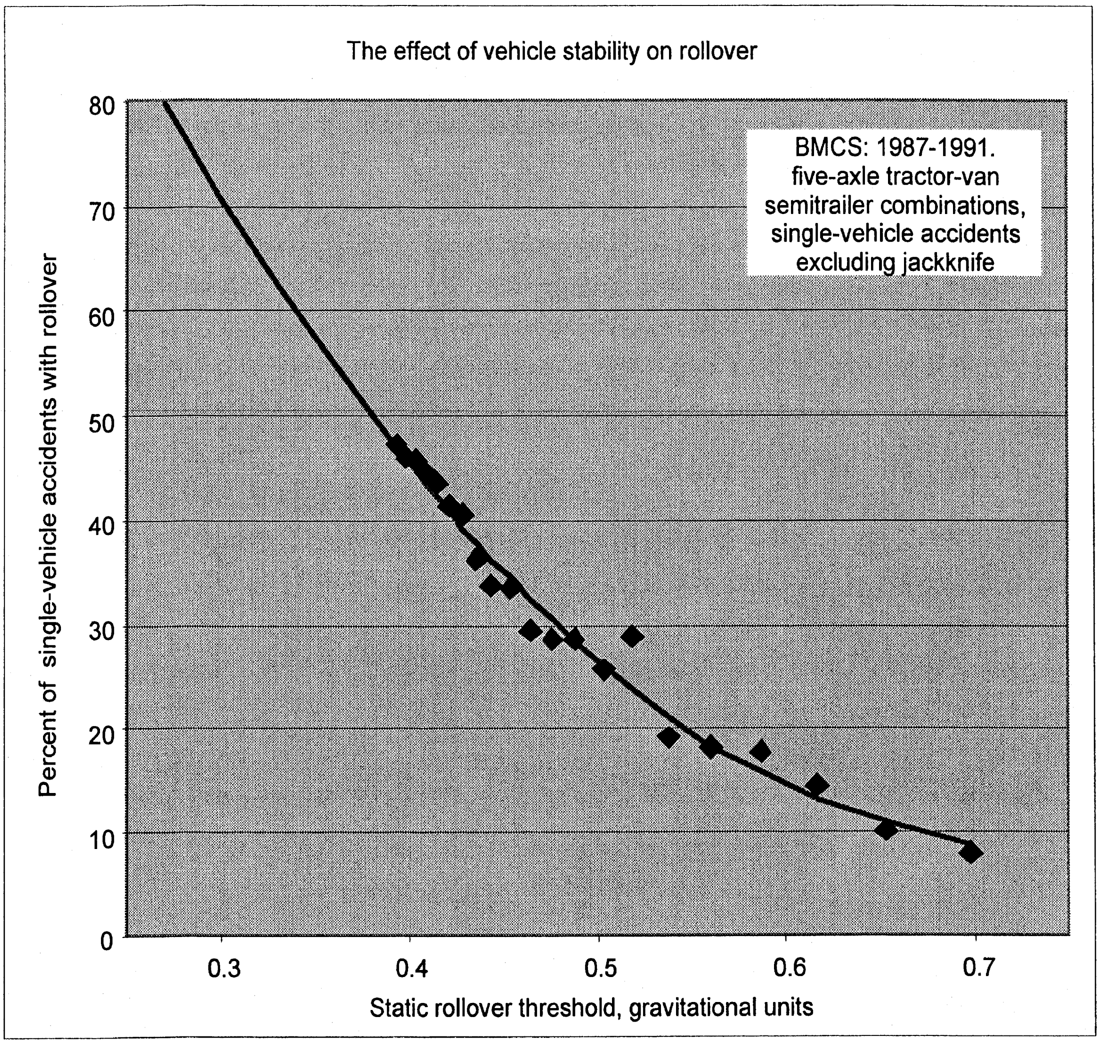


Figure 4. The chance of rollover is strongly influenced by the roll stability of the vehicle

figure reveals a strong, well-behaved relationship between roll stability and the chance of rollover in a single-vehicle accident. The relationship is nonlinear and of the form that one would expect. That is, as the vehicle becomes more and more stable, the chance of rollover asymptotically approaches zero. Conversely, as stability decreases, the sensitivity of the probability of rollover to stability increases rapidly and the function becomes quite steep.

The data of figure 4 can be combined with other data sources to allow at least a rough estimation of the influence of physical roll stability on rollover accident rates (i.e., rollovers per kilometer of travel). Data presented by Winkler[22] indicate that, in the 1988-1990 time period, the fleet of all tractor-semitrailer combinations in the U.S. averaged 8697 rollovers per year while traveling an average of 53,434 million kilometers a year (yielding an average of one rollover per 6.15 million kilometers). Data indicating

tractor-van semitrailers (the most common U.S. heavy vehicle) to facilitate estimation of rollover threshold. Threshold estimates are based on median cg heights and typical vehicle properties. Jacknife accidents are excluded to prevent this peculiar type of accident from distorting the high-stability end of the curve. A detailed explanation of the analysis is given in appendix B.

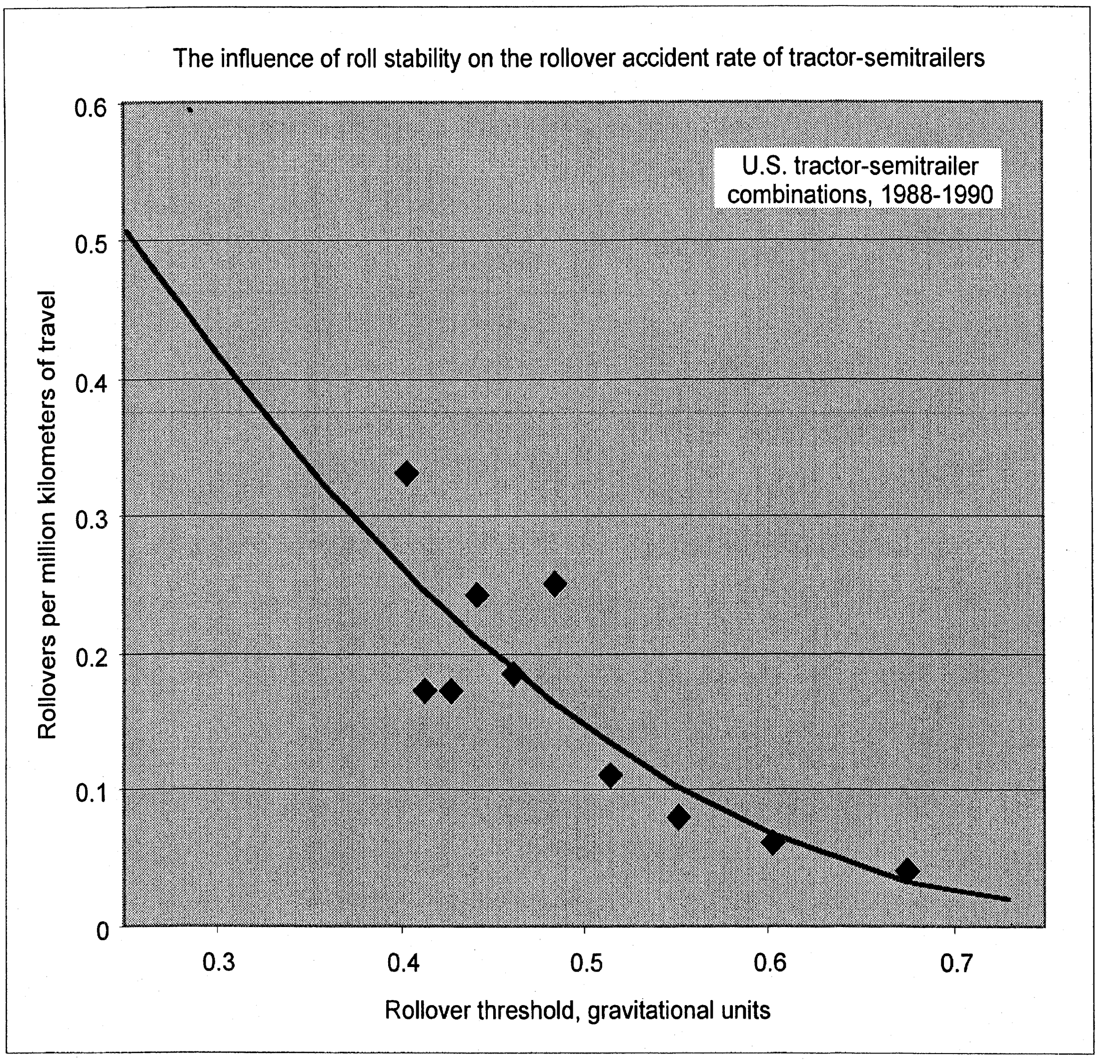


Figure 5. An estimate of the effect of roll stability on the rollover accident experience of tractor-semitrailers

the distribution of tractor-semitrailer mileage by gross weight is presented by Campbell [23]. Combining these data with figure 4 yields the estimates of rollover accident rate presented in figure 5. (See appendix B for details of the analysis.) As in figure 4, a second-order polynomial function is fit to the data and is extrapolated to low levels of stability.

Figures 4 and 5 should be viewed only as estimates. As discussed in appendix B, each is subject to broad assumptions about the applicability of the data. However, regardless of their absolute accuracy, the qualitative findings present in these figures are believed to be valid and significant. Namely, as roll stability declines to low levels, the probability of rollover in an accident increases rapidly until the vehicle becomes very likely to rollover in nearly any accident. Moreover, *for the low-stability vehicles for which rollover is such a great concern, relatively small improvements in physical stability can yield rather large improvements in rollover accident rate.*

2. The Mechanics of Static Roll Stabilty

Introduction and the simplified roll-plane model

All rollover events in the real world are dynamic events to some extent; none are truly quasi-static. However, the accident-data analyses presented in the previous chapter show that there is a very strong relationship between the basic, static roll stability of the heavy vehicle and the actual occurrence of rollover in accidents. Accordingly, this chapter will discuss the mechanics of quasi-static rollover in order to explain how this fundamental performance property derives from the mechanical behavior of the various components of the vehicle.[7]

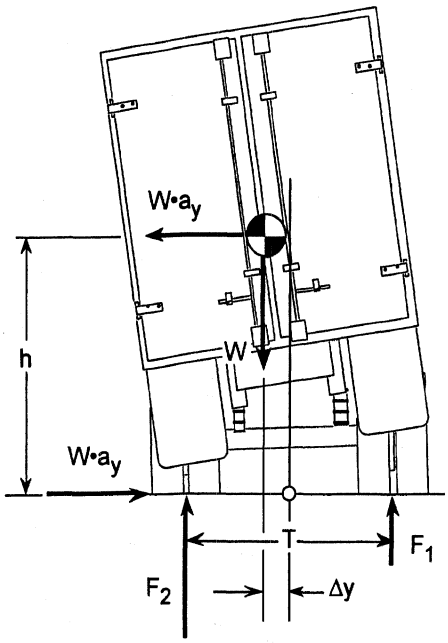


Figure 6. A simplified freebody diagram of a heavy vehicle in a steady turn

Figure 6 presents a simplified model of a heavy vehicle in a steady turn in which the vehicle, its tires, and suspenions have been "lumped" into a single roll plane. The nomenclature of the figure is as follows:

- a_y is lateral acceleration,
- F_i are the vertical tire loads, i=1, 2,
- h is the height of the cg,
- T is the track width,
- W is the weight of the vehicle,
- Δy is the lateral motion of the cg relative to the track,
- ϕ is the roll angle of the vehicle.

The equilibrium equation for roll moment about a point on the ground at the center of the track is:

$$W \cdot h \cdot a_y = (F_2 - F_1) \cdot T/2 - W \cdot \Delta y . \qquad (1)$$

Qualitatively, *there are two destabilizing (overturning) moments acting on the vehicle*:

- — a moment due to the lateral D'Alambert force acting through the cg, $W \cdot h \cdot a_y$, as a results of the external imposition of lateral acceleration
- — a moment due to the weight of the vehicle acting at position that is laterally offset from the center of the track, $W \cdot \Delta y$.

The first of these results from the external imposition of lateral acceleration while the latter results from the internal compliant reaction of the vehicle.

These two destabilizing moments are opposed by one stabilizing (restoring) moment which is due to the side-to-side transfer of vertical load on the tires, $(F_2 - F_1) \cdot T/2$. This moment is also due to the internal, compliant responses of the vehicle. The maximum

7 The analytical approach used herein (and especially the graphical form introduced in figure 8) was developed by Mallikarjunarao. It first appeared in [7] and has been presented in various levels of detail many times since. [e.g., 8, 9, 24]

possible value of this moment is W•T/2 which occurs when all load is transferred to one side of the vehicle, i.e., when $F_2 = W$ and $F_1 = 0$.

One way of interpreting equation 1 and the observation of two destabilizing moments, is that a vehicle's rollover threshold derives from (1) a reference *rigid-body stability*, which would result if Δy were zero, and (2) the degradation from that reference resulting from the lateral motion of the cg allowed by compliances within the vehicle.

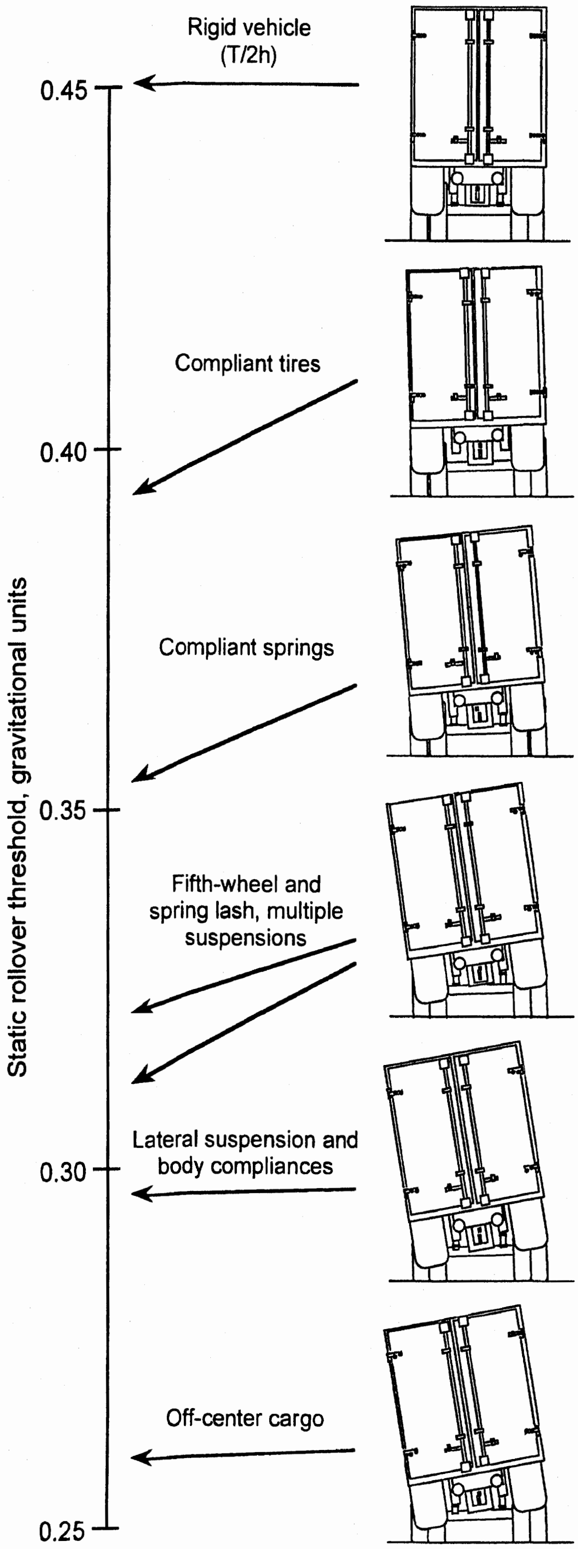


Figure 7. An example case showing various major influences which determine roll stability

Figure 7 presents a "case study" which illustrates how various properties of the vehicle contribute to the rollover threshold according to this view. The example is of a rather low-stability vehicle. Its heavy load and relatively high payload establish a *rigid-body stability of 0.45 g*. The roll motion allowed by the compliance of the tires and the suspension springs drop the stability level to about 0.36 g. Free play, or lash, in the suspension springs and fifth-wheel coupler allow more roll motion, further reducing stability. Less-than-optimum distribution of load on the suspensions lowers stability still more. Structural compliances in suspensions and the cargo body and the off-center positioning of the payload allow additional lateral translation of the cg.

The qualitative message of figure 7 is that roll stability is established by the summated effects of many compliance mechanisms. *While the effect of any one compliance may be small, virtually all compliances degrade stability. All the compliances combined can reduce roll stability to as little as 60 percent of the rigid-vehicle stability.*

The following paragraphs will review just how each of the mechanisms of figure 7 actually influences stability by progressively examining their individual influences on the behavior of equation 1.

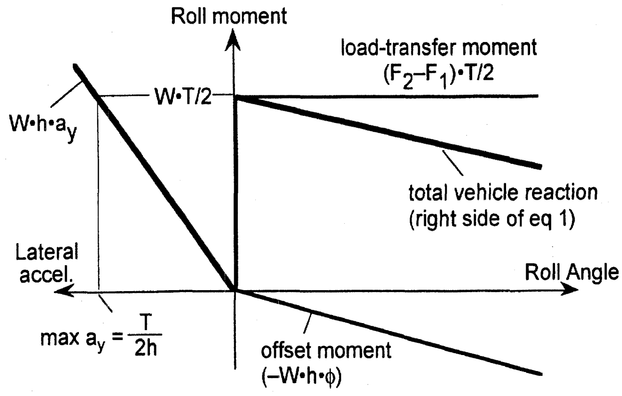


Figure 8. Graphic presentation of the roll-equilibrium equation for a rigid vehicle

(Small angles are assumed to allow linearity.)

Rollover of the rigid vehicle

Begin by considering a completely rigid vehicle. Figure 8 is a graphic representation of equation 1 for such a vehicle. Equation 1 has been arranged with the externally applied moment on the left side and the internal vehicle-reaction moments on the right side. The graph of figure 8 is arranged the same way. The left side of the equation is presented on the left side of the graph in a plot of roll moment (on the ordinate) versus lateral acceleration (on the abscissa to the left). The right side of the equation is presented on the right side of the graph in a plot of roll moment versus roll angle (on the abscissa to the right).

Because this vehicle is rigid, any finite roll of the vehicle results immediately in complete transfer of all vertical load onto the tires on one side of the vehicle. The unloaded tires would immediately lift from the ground. This is reflected in the plot of load-transfer moment shown as a horizontal line at the maximum value of W•T/2. However, the offset moment grows proportionately (and negatively) with roll angle as the cg translates laterally. This behavior is shown in the downward sloping plot of the offset moment. The sum of the load-transfer moment and the (negative) offset moment constitutes the total vehicle reaction as expressed by the right side of the equation. The graph shows that this combined function achieves its maximum value at zero roll angle. The negative slope of this plot at all finite roll angles indicates that the vehicle becomes unstable immediately as its tires lift from the ground. By projecting the maximum value of this right-side total onto the plot of the left side of the equation, it can be seen that the maximum lateral acceleration (in gravitational units) that can be sustained by this rigid vehicle in an equilibrium condition is the ratio of the half track (T/2) to the cg height (h). This well-know, *rigid-vehicle stability factor*, T/2h, *is the most fundamental vehicle property which influences basic roll stability.*

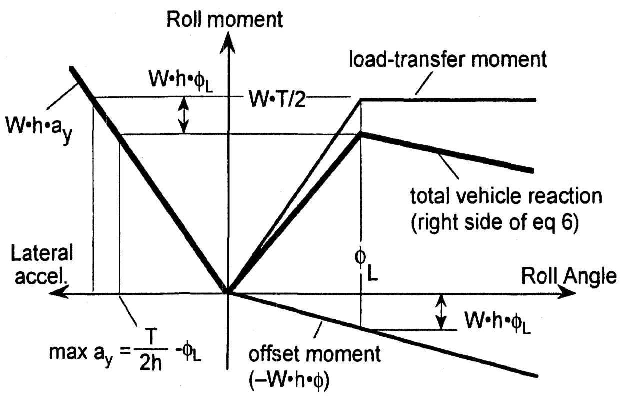


Figure 9. Graphic presentation of the roll equation for a vehicle with compliant tires and suspension

The vehicle with compliant tires

Now consider a vehicle with compliant tires represented by linear vertical springs. This vehicle would roll about a point located on the ground plane at the center of the track. In the process, the tire spring on one side compresses, increasing its load, while the tire spring on the other side extends, decreasing its load. Simultaneously, the cg translates

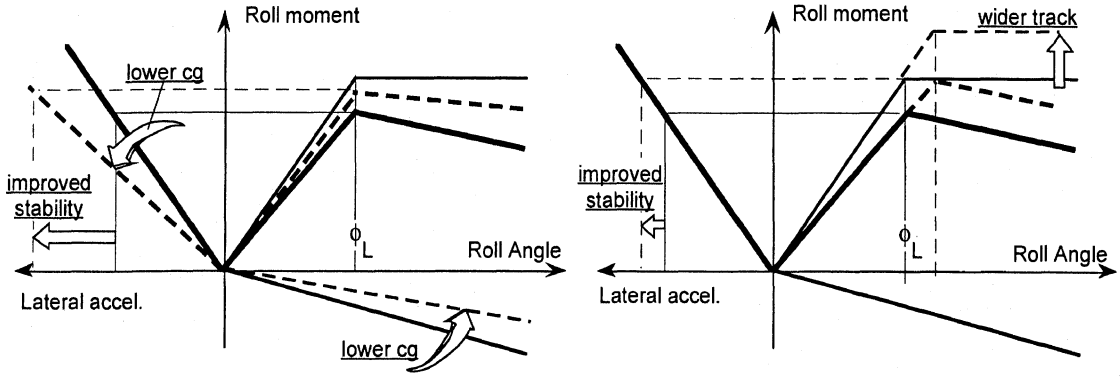


Figure 10. Improving roll stability with a lower cg and a wider track

laterally a distance equal to roll angle times the height of the center of gravity (cg height). Load-transfer moment is shown to develop progressively with roll angle. Full load transfer and the resulting maximum load-transfer moment are achieved at the "tire-lift-off angle," ϕ_L. At ϕ_L, the offset moment has grown to a negative value of $W \bullet h \bullet \phi_L$. The maximum value of the total-reaction moment is achieved just as tires lift at a roll angle of ϕ_L. This maximum is less than that which was achieved by the rigid vehicle because of the non-zero offset moment. At roll angles greater than ϕ_L, offset moment continues to increase but load-transfer moment is saturated. The resulting downward slope of the total vehicle reaction again indicates an unstable system.

The graphic form of figure 9 can be used to illustrate how the rollover threshold is influenced by both cg height and track width. Consider figure 10. The plot on the left shows that lowering the cg improves stability by reducing both destabilizing moments. Moment due to the lateral D'Alambert force through the cg is reduced directly by lowering its line of action. The lateral shift of the cg is also reduced, thereby reducing the associated moment. On the overhand, widening the track of the vehicle improves stability by increasing the stabilizing moment available from side-to-side load transfer.

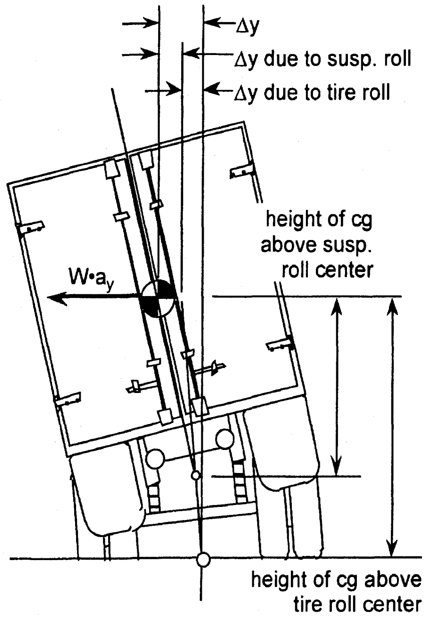


Figure 11. Tire and suspension roll motions occur about different centers

The vehicle with roll-compliant suspension

The effect of roll compliance of the suspension is very similar to the effect of tire compliance except that the additional suspension roll motion takes place about a roll center which is typically well above the ground. From figure 11 it is apparent that the height of this suspension roll center has two influences. (1) For a given roll angle condition, the lateral displacement of the cg is less if the suspension roll center is higher. (2) For a given cg height, roll moment acting on the suspension due to the D'Alambert force is less if the suspension roll center is higher. This, in turn, reduces body roll angle and the resulting lateral displacement of the cg.

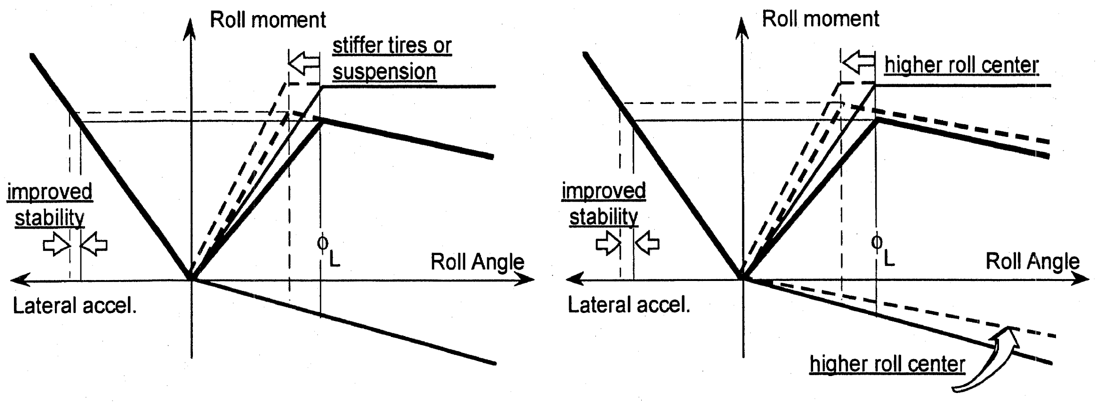


Figure 12. Improving roll stability with stiffer tires or suspension and by raising the suspension roll center

Figure 12 illustrates how stability is improved by (1) increasing tire or suspension stiffness, and (2) raising the suspension roll center. Increasing roll stiffness has the rather straightforward effect of causing the complete side-to-side load transfer to occur at a lesser roll angle. Consequently, at the point of tire lift, the destabilizing moment from the lateral displacement of the cg is smaller. This translates to improved roll stability.

The graph on the right shows that raising the roll center has two effects. (1) The amount of lateral translation of the cg per unit of roll decreases, thus lessening the offset moment, and (2) the stiffness of the system (load-transfer moment per unit of roll) increases. Both of these effects are the result of reducing moment about the suspension roll center by shortening the distance between it and the line of action of the lateral force through the cg.

The influence of lash in suspensions and in the fifth-wheel coupler

Many leaf-spring suspensions allow vertical free play, or lash, between the spring and body. The spring must pass through this lash as it transitions from compression to tension. Similarly, fifth-wheel couplers typically allow for some vertical free play between the kingpin and the coupler plate. As shown in figure 13, either or both of these can appear as free play in the rolling motion of the vehicle, especially for vehicles with high centers of gravity.

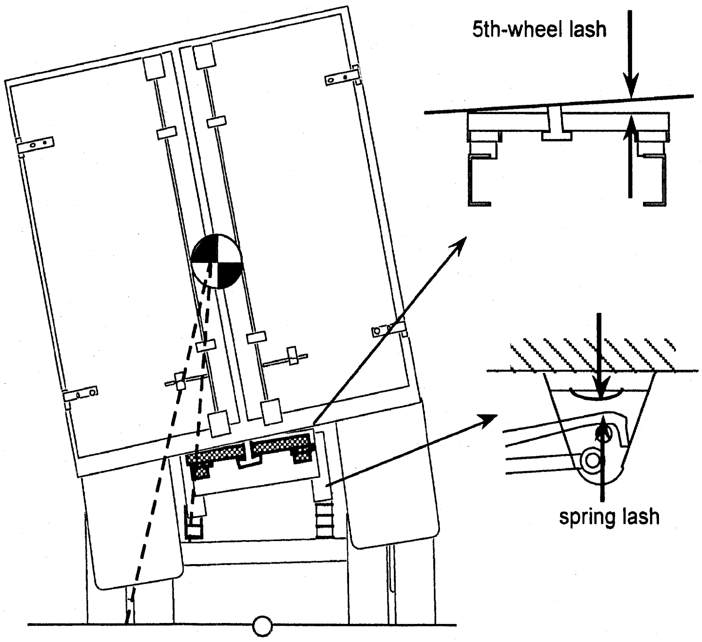


Figure 13. If the cg is high, the sprung mass attempts to "roll off" the suspension and fifth-wheel prior to rollover of the vehicle

Figure 14 shows the influence of free play on the roll stability of the vehicle. As the region of the lash is transitioned, the free-rolling motion of the vehicle simply

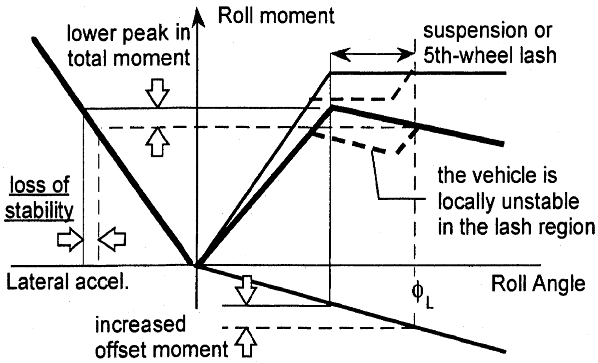


Figure 14. Suspension and fifth-wheel lash increase the roll angle required for complete load transfer

results in an increase in lateral displacement of the cg and the associated offset moment, but with no compensating increase in the load-transfer moment. Roll stability declines accordingly. Note that, as the system passes through lash, it is locally unstable in roll, as indicated by the negative slope of the total moment. After the zone of the lash, there is, in theory, a small zone in which the vehicle is again stable. In practice, however, momentum will typically carry the vehicle through this stable zone to rollover.

The influence of multiple suspensions

To this point, the discussion has assumed that all the tire and suspensions of the vehicle may be "lumped" and assumed to operate as a single suspension. Now consider figure 15 in which the load-transfer moments of the three suspensions of a tractor-semitrailer combination are shown individually. The relative performance of the three suspensions in the figure is typical of real vehicles. That is:

- — The trailer suspension exhibits the highest roll stiffness followed by the tractor drive-axle suspension and then the steer-axle suspension. The stiffness of the latter is quite low compared to the other two.
- — The loads carried by (and therefore, the maximum load-transfer moments of) the drive-axle and trailer-axle suspensions are similar and are each considerably more than the load carried on the steer axle.

Starting from the left side of the graph, consider the process as the vehicle gradually experiences increasing lateral acceleration and corresponding roll motion. At the beginning, all tires are on the ground, and the slope (effective roll stiffness) of the total system is determined by the sum of the stiffness of all suspensions less the negative influence of the offset moment.

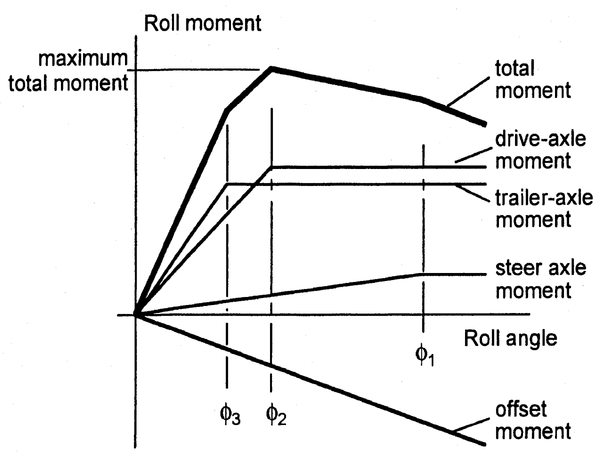
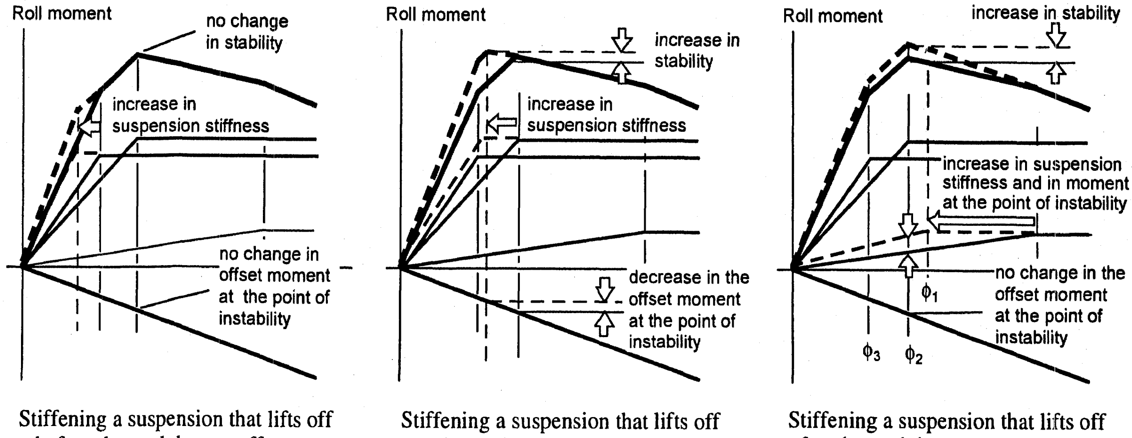


Figure 15. The load-transfer moments of the steer-axle, drive-axle, and trailer suspensions shown separately

Because of its greater roll stiffness, the trailer suspension transfers load side-to-side most quickly and is first to arrive at the point of tire lift (ϕ_3). When this occurs the stiffness of the trailer suspension is "lost" and the stiffness of the total system declines. However, the total stiffness is still positive, and the system is still stable even with the trailer tires off the ground. The process continues and the next tires to lift from the ground are the drive-axle tires (ϕ_2). At this point, the drive-axle stiffness is also lost. The remaining positive stiffness of the steer-axle suspension is less than the negative influence of the offset moment. As a result, the total-system stiffness is negative, and the vehicle is unstable. The peak in the total-system moment occurs at the roll angle ϕ_2 and represents the rollover threshold of the vehicle.

Generalizing on the presentation of figure 15, three classifications of suspensions can be identified: (1) suspensions whose tires lift off before (i.e., at a smaller roll angle) the peak of the total-system moment, (2) suspensions whose tires lift off at (and therefore define) the peak of the total-system moment, and (3) suspensions whose tires remain on the ground when the total-system moment peaks. Figure 16 illustrates the different influences brought about by changing the roll stiffnesses of these three types of suspensions. Starting from the left of the figure, we see that stiffening suspensions of the first type have no influence on rollover threshold. In essence, this type of suspension "delivers" all its available stabilizing moment to the system prior to instability. Delivering that moment even quicker can have no influence. It does not change the roll angle of the point of instability and it does not change the amount of stabilizing moment available from this suspension. However, from the second graph, stiffening the second type of suspension does result in an improvement in rollover threshold. Stiffening this suspension causes the point of instability to occur at a lesser roll angle. As a result the offset moment is less and stability improves. Finally, the third graph shows that stiffening the third type of suspension improves the rollover threshold by increasing the stabilizing moment which this suspension supplies at the point of instability.

Following the logic of figure 16, it can be shown that *the optimum situation for maximizing roll stability (although, not necessarily for best handling or other concerns) is for tire lift to occur simultaneously at all suspensions*. That is, for the example of figure 16, stiffening the drive-axles is productive only until the angle ϕ_2 is reduced to equal ϕ_3. Any additional stiffening of the drive-axle suspension would cause it to become a type-1 suspension for which increased stiffness is not effective. The same principle holds for the steer axle. Finally, noting that the angle of lift off is a combined function of the effective roll stiffness of the suspension and the load carried by the suspension (i.e., its maximum load-transfer moment), then to first order, *roll stability is optimized when load is distributed among suspensions in proportion to the distribution of roll stiffness.*



Stiffening a suspension that lifts off before the peak has no effect on roll stability.

Stiffening a suspension that lifts off at the peak improves stability by reducing the offset moment.

Stiffening a suspension that lifts off after the peak improves stability by increasing the load-transfer moment.

Figure 16. Increasing the roll stiffness of different suspensions has different influences on roll stability

Other mechanisms influence static roll stability

The majority of the lateral displacement of the cg (and thus, the majority of the destabilizing offset moment) usually results from roll motion due to the vertical and roll compliances of the tires and suspensions. However, there are many other compliances within the vehicle, each of which can contribute some additional lateral offset of the cg. Two such compliances which are known to be significant are illustrated in figure 17. These are lateral compliance of the suspension and lateral beaming of the vehicle frame. The figure also illustrates the obvious possibility that the cargo can be placed off center and thereby contribute to the lateral displacement of the cg.

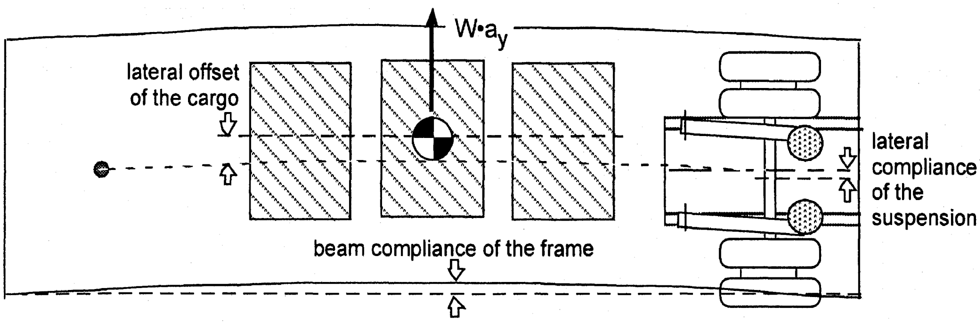


Figure 17. Examples of other mechanisms that can contribute to the destabilizing offset moment

The significance of lateral displacements such as these can be judged by comparing them to T/2. That is, the lateral displacement of the cg is, in effect, a direct reduction of the half-track. In round numbers, the half-track of an axle with dual tires is about 95 cm.[8] Thus, a 1-cm lateral deflection results in loss of stability equal to about 1 percent of the original rigid-body stability of the vehicle. Lateral suspension deflection may be on the order of 2 cm. Lateral beaming of the trailer may be 3 cm or more. A variety of other compliances may each produce displacements on the order of several millimeters, and of course, the lateral offset of the placement of the cargo can be very substantial.[9] While none of these displacements may seem significant individually, the total influence can easily account for the loss of a significant portion of the rigid-vehicle stability.

There is a general point of some significance which follows from this discussion. That is, *the roll stability of heavy vehicles typically derives from the summation of T/2H plus a large number of small influences resulting from various compliances.* The corollary observation is that *virtually all compliances degrade stability.* Thus,

[8] Ninty-five centimeters is the nominal half-track as measured to the *center of the dual-tire pair.* This is the appropriate point of reference—not the center of the outer tire nor the outer edge of the tire tread. As shown in figure 14, the point of roll instability of a tractor-semitrailer combination typically occurs when the light-side drive-axle tires first lift from the road surface. This occurs at only a few degrees of roll and with all the tires on the heavily loaded side of the vehicle, including the inside tires of the dual-tire pairs, still very firmly on the ground. At this point in the rollover process, however, the vehicle is already unstable and rollover is virtually an established fact. It is only much later in the process—and much too late to be of any significance—that the inside dual tires will lift off the road surface.

[9] Fluid cargos are of particular interest in this regard. They will be considered in the next chapter.

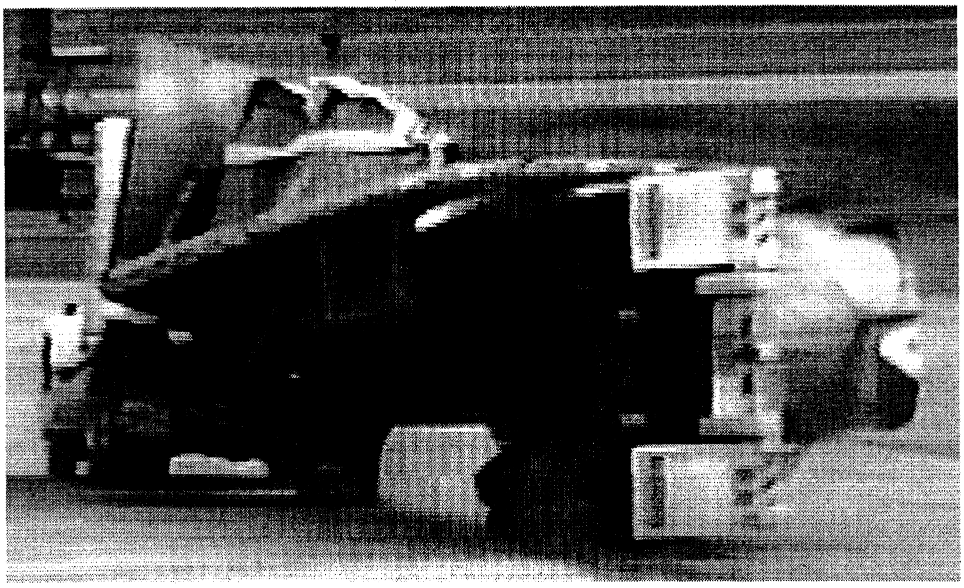


Figure 18. The rear end of a torsionally compliant flat-bed trailer rolls over nearly independently of the front end

engineering judgements as to whether individual compliances are "negligible" should not be made in isolation but should be considered in the context of all such compliances.

Finally, the torsional compliance of the vehicle frame stands out as a uniquely important element in establishing the roll stability of some vehicles, particularly those with flat-bed trailers. The photograph of figure 18 speaks clearly to the point. The photograph was taken on a test track. The rollover event was unintentional and occurred in nearly a quasi-static fashion.

The vehicle is loaded with four rolls of aluminum sheet. (The intention of the exercise was to test the cargo-restraint system.) Two rolls are at the extreme front of the trailer, directly over the drive-axle suspension. The other two rolls are at the extreme rear of the trailer directly over the trailer suspension. The frame of the trailer is so compliant that the front and rear of the vehicle are nearly independent bodies with respect to roll. The "front vehicle" benefits from the low cg height of the tractor mass and, as the picture shows, the "rear vehicle" has appreciably lower roll stability.

Figure 19 is an example of another vehicle whose stability might be expect to suffer due to torsional compliance of the frame. In this case, the central location of the load

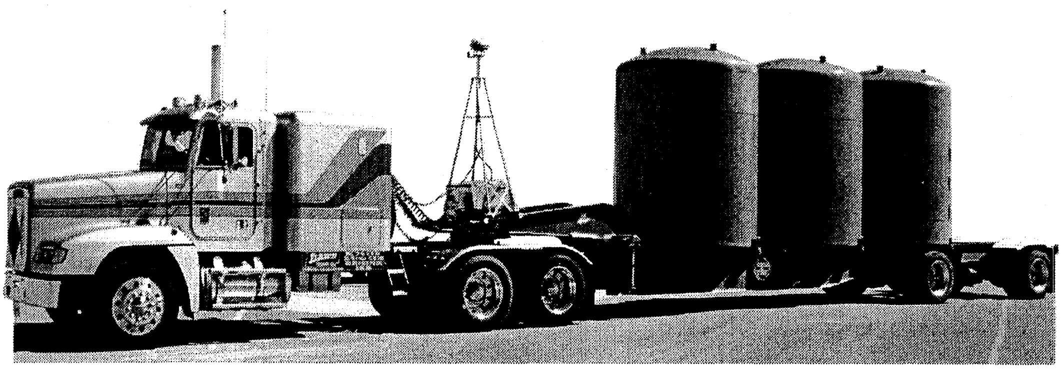


Figure 19. The actual roll stability of this vehicle is a small fraction of its "rigid-vehicle" stability owing to roll compliance of the trailer frame

whose cg lies high above the neutral axis of the compliant frame will result in a large lateral displacement of the cg and correspondingly large destabilizing offset moment.

Measuring rollover threshold with the tilt-table experiment

The tilt-table methodology is a physical simulation of the roll-plane experience of a vehicle in a steady turn. The method provides a highly resolute means of determining rollover threshold and examining the mechanism by which this limit is determined.

In this experimental method, the vehicle is placed on a tilt table and is very gradually tilted in roll. As shown in figure 20, the component of gravitational forces parallel to the table surface provides a simulation of the centrifugal forces experienced by a vehicle in turning maneuvers. The progressive application of these forces by slowly tilting the table serves to simulate the effects of quasi-statically increasing lateral acceleration in steady turning maneuvers. The tilting process continues until the vehicle reaches the point of roll instability and "rolls over." (The vehicle is constrained by safety straps to prevent actual rollover.)

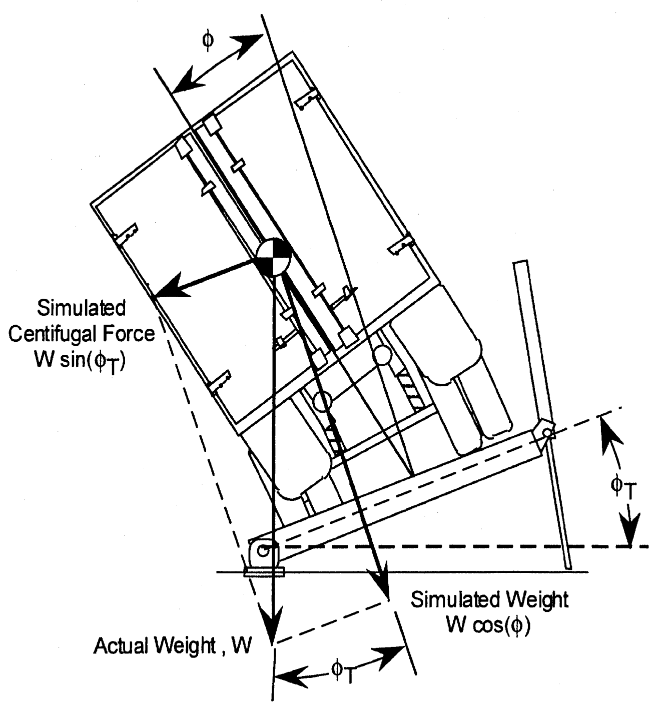


Figure 20. The tilt-table experiment

When the table is tilted, the component of gravitational forces parallel to the table surface, $W \cdot \sin(\phi_T)$, simulates lateral forces, and the weight of the vehicle itself is simulated by the component of gravitational forces that are perpendicular to the table (i.e. $W \cdot \cos(\phi_T)$, where W is the weight of the vehicle and ϕ_T is the roll angle of the table relative to the true gravitational vector). Thus, the forces acting during the tilt-table test are *scaled down* by a factor of $\cos(\phi_T)$. Since the important mechanisms of actual rollover depend on the *ratio* of the centrifugal forces to the vertical, gravitational forces, it is appropriate to take the ratio of the simulated lateral acceleration forces to the simulated weight to represent lateral acceleration when interpreting the results of a tilt-table experiment. That is:

$$a_{ys} \equiv \tan(\phi_T) = W \cdot \sin(\phi_T) / W \cdot \cos(\phi_T) \tag{2}$$

where:

a_{ys}	is the simulated lateral acceleration (in gravitational units)
ϕ_T	is the roll angle of the tilt table
W	is the weight of the vehicle.

The quality of $\tan(\phi_T)$ as an estimate of actual static roll stability depends, in part, on how closely $\cos(\phi_T)$ approximates unity. In the tilt-table experiment, both the vertical and lateral loading of the vehicle are reduced by the factor $\cos(\phi_T)$ relative to the loads they are meant to represent. Because of the reduced vertical loading, the vehicle may rise on its compliant tires and suspensions relative to its normal ride height, resulting in a slightly higher cg position and, possibly, a slightly *low* estimate of the static roll stability limit. At the same time, static lateral loading is also reduced by the factor $\cos(\phi_T)$. This may result in compliant lateral and roll motions of the vehicle that are relatively small, tending to produce a slightly *high* estimate of the static roll stability limit. The fact that these two influences tend to cancel each other is clearly advantageous. More importantly, for the moderate angles of tilt required to test large commercial vehicles, $\cos(\phi_T)$ remains sufficiently near to unity such that accurate representations of all loadings are maintained. (At a tilt angle simulating 0.35 g lateral acceleration, $\cos(\phi_T)$ is 0.94.)

A second error source in this physical simulation methodology involves the *distribution* of lateral forces among the tires of the several axles of the vehicle. Lateral forces developed at the tire-road interface must, of course, satisfy the requirements of static equilibrium of lateral force and yaw moments acting on the vehicle. However, many commercial vehicle units are equipped with multiple nonsteering axles which results in the system being statically indeterminate. Thus the distribution of lateral reaction forces among the axles is partially dependent on the lateral compliance properties of tires and suspensions. The compliance properties that are in play while the vehicle is sitting on the tilt table are not precisely those that are in play while the vehicle is in motion on the road. The significance of this error source is dependent on axle location, and the similarity, or lack thereof, of geometry among the redundant axles and suspensions. For many commercial vehicles, the close spacing and geometric similarity of the two axles of each tandem suspension tend to minimize these errors.

A third error source lies in the side slip angle of the tractor and the yaw articulation geometry of the vehicle. Although tilt-table experiments are conducted with these two yaw plane angles at zero, the negotiation of real turns at significant speed generally implies the existence of small, nonzero yaw plane angles. Some reflection on this matter (and that of the preceding paragraph) reveals that, in practice, static rollover threshold varies somewhat as a function of turn radius. In this light, the zero-yaw-angle condition is simply seen as one of many possible test conditions—certainly the one most easily implemented.

The most fundamental aspects of the mechanics of quasi-static rollover were presented earlier using very simplified models. Despite the high level of simplification, the validity of the ideas presented have been confirmed in numerous tilt-table experiments.[17,19,20]

Figure 21 presents data gathered in one such experiment.[17] The test vehicle was a five-axle tractor-van semitrailer equipped with air suspensions at the drive axles and the trailer axles. The figure is a plot of trailer-body roll angle (relative to the table surface, of course) as a function of simulated lateral acceleration (the tangent of the tilt angle). The annotations in the figure point out features in the data corresponding to many of the elements of the process which were discussed earlier in this chapter.

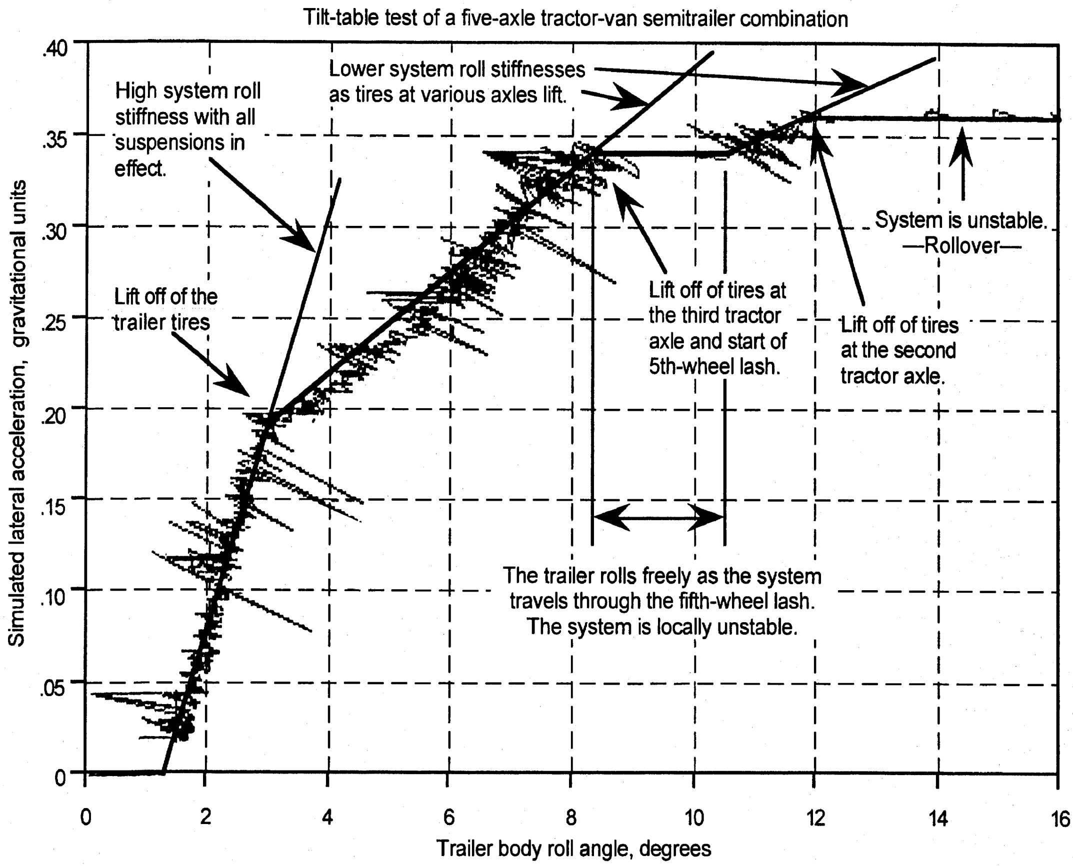


Figure 21. Tilt-table data showing some of the principal elements of quasi-static rollover mechanics

3. Dynamic Considerations in Rollover Of Heavy Vehicles

Introductions

The accident-data analyses presented in chapter 1 make clear that the static roll stability is the dominate vehicle quality affecting the chance of a given heavy truck being involved in a rollover accident. Chapter 2 reviews the mechanics of static stability. However, virtually all rollover accidents in the real world are dynamic events to some extent; none are truely quasi-static.

The influence of sloshing liquids and other moving loads[10]

In the majority of commercial truck operations, the load on the vehicle is fixed and nominally centered. In certain cases, however, the load may be able to move on the vehicle, with the potential of affecting the turning and rollover performance. The most common examples of moving loads are:

— bulk, liquid tankers, with partially filled compartments,
— refrigerated vans hauling suspended meat carcasses, and
— livestock.

The performance properties of commercial vehicles used in these applications may be influenced by the free movement of the load in either the longitudinal or lateral directions. This chapter will present material on the first two types of loads.

Sloshing-liquid loads

In the majority of commercial truck operations, the load on the vehicle is fixed and nominally centered. In certain cases, however, the load may be able to move on the vehicle, with the potential of affecting the turning and rollover performance. The most important of these is liquid cargo carried in tanks.

In the operation of a bulk-liquid transport vehicle, the moving load that can affect its cornering and rollover behavior is the presence of unrestrained liquid due to partial filling of the tank or its compartments. A compartment that is filled to anything less than its full capacity allows the liquid to move from side to side, producing the so-called "slosh" load condition. Slosh is of potential safety concern because (1) the lateral shift of the load reduces the vehicle's performance in cornering and rollover, and (2) the dynamic motions of the load may occur out of phase with the vehicle's lateral motions in such a way as to become exaggerated and thus further reduce the rollover threshold.

The motions of liquids in a tank vehicle can be quite complex due to the dependence of the motions on tank size and geometry, the mass and viscosity of the moving liquid, and the maneuver being performed.[e.g., 26,27,28] Fundamental analyses of sloshing liquids in road tankers appeared in the literature from the 1970s.[e.g., 29,30] A number of more elaborate computer studies arose in the late 1980s and early 1990s.[e.g., 31,32,33] This discussion is constrained to basic elements that provide insight on the mechanisms by which fluid motions influence rollover. The mechanisms of slosh are most readily

[10] The material of this section was largely created by Ervin and has been presented in [25].

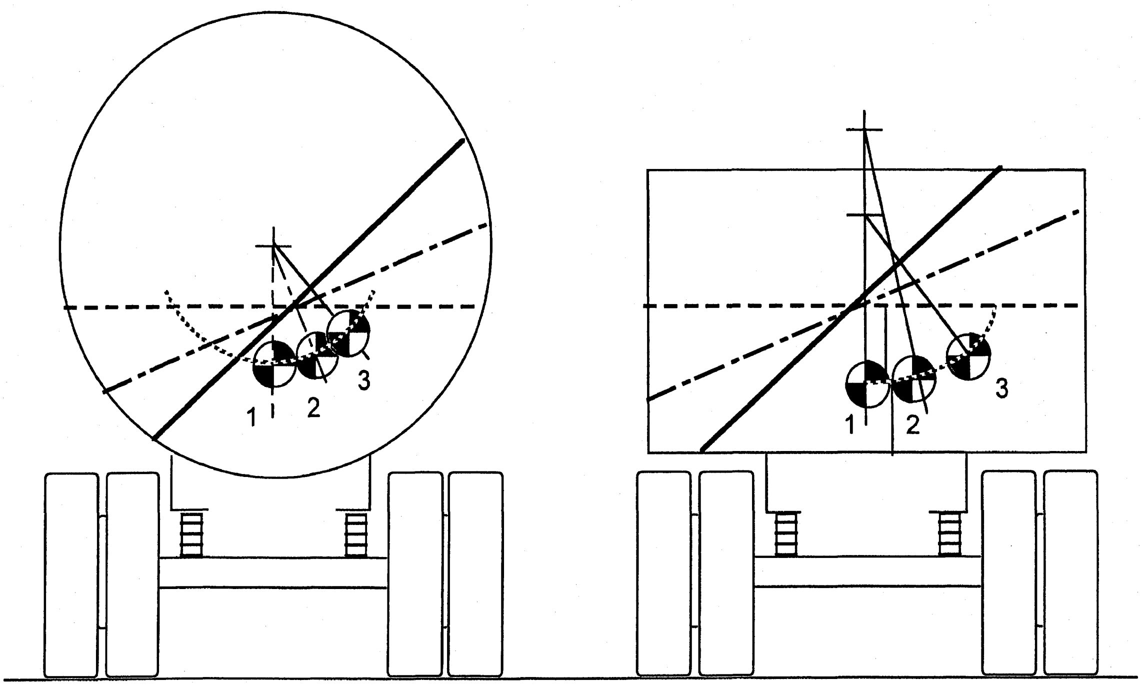
described in simple steady-state cornering, although it is in transient maneuvers that the most exaggerated fluid displacements take place.

Steady turning

When a slosh-loaded tanker performs a steady-state turn, the liquid responds to lateral acceleration by displacing laterally, keeping its free surface perpendicular to the combined forces of gravity and lateral acceleration. Figure 22a illustrates the position of a partial liquid load in a circular tank which is being subjected to a steady-state cornering maneuver. The mass center of the liquid moves on an arc, the center of which is at the center of the circular tank. In effect, the shift of the liquid produces forces on the vehicle as if the mass of the load was located at the center of the tank.

With more complex tank shapes, even the steady-state behavior becomes somewhat difficult to analyze. In particular, with unusual tank shapes it becomes more difficult to describe the motion of the liquid's center of mass as a function of lateral acceleration. As a contrast to the circular tank, figure 22b illustrates the behavior of liquid in a rectangular tank. At low lateral accelerations, the liquid movement is primarily lateral, centered at a point well above the tank center. Hence, its effect is similar to having a very high mass center. With increasing lateral acceleration, the mass center follows a somewhat elliptical path.

While the circular tank results in a vehicle with a higher load center, efforts to reduce the load height by widening and flattening the tank can be expected to increase vehicle sensitivity to slosh degradation of the rollover threshold. The effect is illustrated by the plot in figure 23 taken from Strandberg[30] showing rollover threshold versus load condition in steady-state cornering. For a circular tank, increasing load lowers the threshold continuously due to the increasing mass of fluid free to move sideways. In this



a. Circular cross section b. Rectangular cross section

Figure 22. Illustration of liquid position in steady-state turning for circular and rectangular tanks

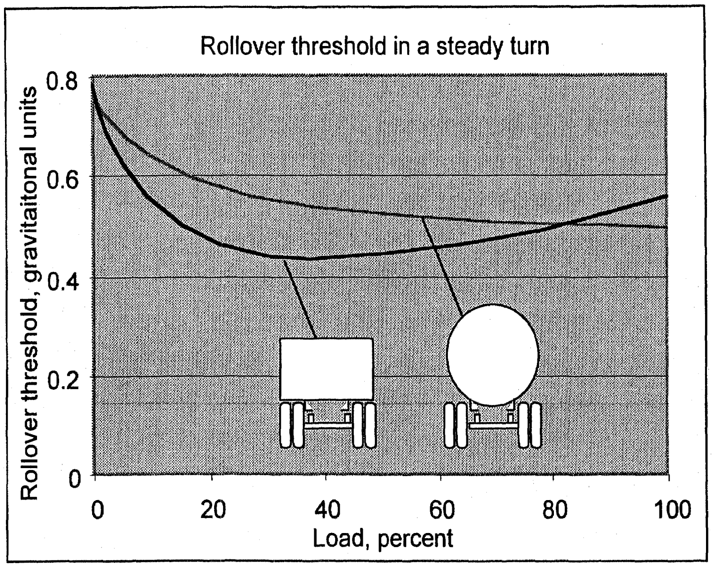


Figure 23. Rollover threshold in a steady turn as a function of the percentage of load of unrestrained liquid

case, the minimum rollover threshold occurs at full load. For a vehicle having a rectangular tank, higher levels of rollover threshold occur when the tank is either empty or full, although at intermediate load conditions the rollover threshold is severely depressed due to the greater degree of lateral motion possible for the unrestrained liquid. Thus, the rectangular tank shape (in contrast to the circular) can potentially result in rollover thresholds with slosh load that are less than that of the fully loaded vehicle.

Transient turning

In transient maneuvers such as an abrupt evasive steering maneuver (e.g., a rapid lane change), slosh loads introduce the added dimension of dynamic effects. With a sudden steering input, the rapid imposition of lateral acceleration may cause the fluid to displace to one side with an underdamped (overshooting) type of behavior. The difference between the steady-state and transient maneuvers are primarily a matter of the time involved in entering the turn. The steady-state type of behavior is observed when the turn is entered very slowly, whereas the transient behavior applies to a very rapid turning maneuver. The difference between the two is illustrated in figure 24, which shows the way in which the liquid surface moves in each type of maneuver. In effect, the liquid motion occurs much like that of a simple undamped pendulum. The response of the liquid mass to a step input of acceleration (as in the bottom illustration of figure 24) would be seen to displace to an amplitude which is approximately twice the level of the steady-state amplitude. In a lane-change maneuver in which the acceleration goes first in one direction and then

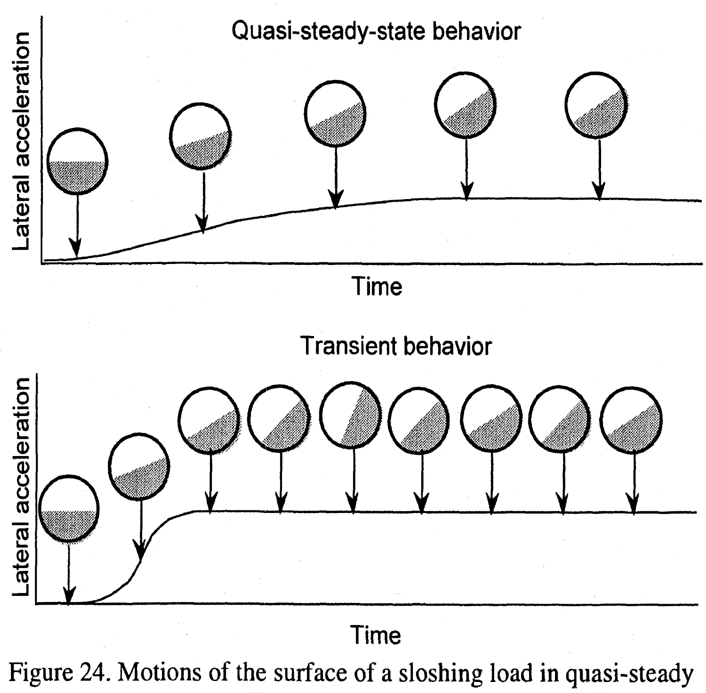


Figure 24. Motions of the surface of a sloshing load in quasi-steady and transient turning

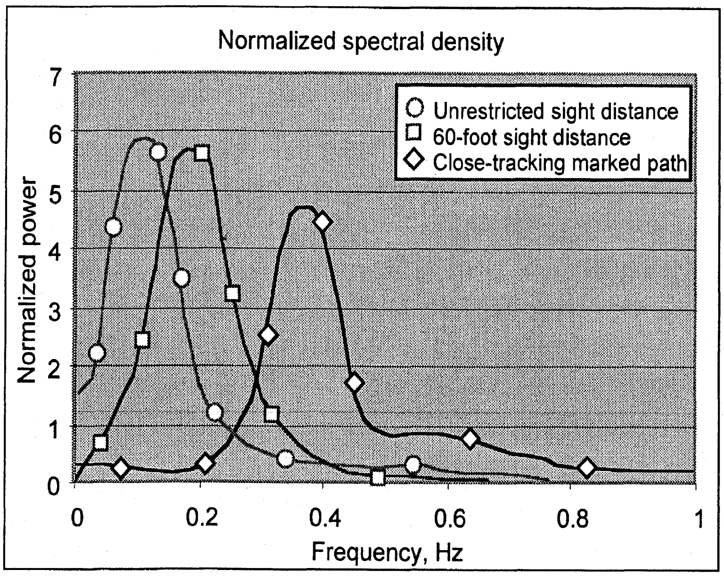


Figure 25. Frequency content distributions obtained from steering input time histories

the other, an even more exaggerated response amplitude can be produced.

In general, the degree to which the dynamic mode is excited depends on the timing of the maneuver. The unrestrained liquid will have a natural frequency for its lateral oscillation which depends on the liquid level and cross-sectional size of the tank. For a half-filled, eight-foot-wide tanker, this frequency is approximately 0.5 Hz (cycles per second); whereas, a six-foot-diameter circular tank (typical of an 8,800-gallon tanker) would have a frequency of approximately 0.6 Hz. As for dynamic systems in general, if the frequency content of input (lateral acceleration) stays below this natural frequency, the response is largely quasi-static, but if the input contains substantial power at or above the natural frequency, the response will be dynamic. Studies of driver steering behavior have shown that in a *demanding* steering task, such as an accident-avoidance maneuver, the steering input may have significant energy near the 0.5 Hz frequency.[34] Figure 25 shows the frequency content of steering motions measured under different driving tasks. The "tracking" task especially indicates that steering input at or near the 0.5 Hz frequency may be readily applied by a driver. Thus it is possible to excite these dynamic motions. For example, the two-second lane change used as a typical evasive maneuver for evaluating rearward amplification constitutes a lateral acceleration input closely matched to the slosh frequency.[35] Hence it must be concluded that dynamic slosh motions can be readily excited on a tanker of normal size, especially in the course of evasive maneuvers such as a lane change.

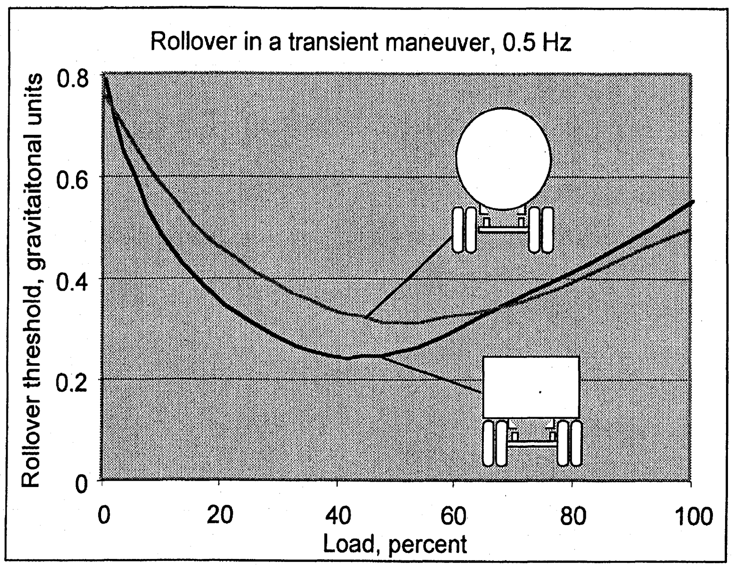


Figure 26. Rollover threshold in a transient turn as a function of the percentage of load of unrestrained liquid

In transient maneuvers, the rollover thresholds are depressed by this dynamic motion. Figure 26 shows the estimated rollover threshold as a function of load for unrestrained liquids in a transient maneuver. In the transient case, even the circular tank experiences reduced rollover thresholds when partially loaded due to the fact that the fluid can "overshoot" the steady-state level. Understandably, the elliptical tanker is even worse.

Though the results shown are derived from analytical studies, experimental tests of partially loaded tankers generally confirm these observations.[36]

Methods for dealing with partial liquid loads

In the vocational use of many liquid bulk haulers it is necessary to run at times with partial loads. This is especially true with local delivery tankers hauling gasoline and home-heating fuel. The question is what can be done to reduce the sensitivity and hence the potential risks of using these vehicles, once a substantial fraction of their load has been delivered? Of course, specifying a vehicle with suspension systems most resistant to rollover is a first step. However, at least two other aids are available.

Baffles. Baffles are commonly used in tank vehicles, except in special cases where provisions for cleaning prevent their use (such as bulk-milk haulers). However, the common baffle arrangement is a transverse baffle intended to impede fore/aft movement of the load. These transverse baffles have virtually no utility in preventing the lateral slosh influential to roll stability. To improve roll performance, longitudinal baffles are required. Properly designed, they can substantially reduce the slosh degradation of cornering and rollover performance.

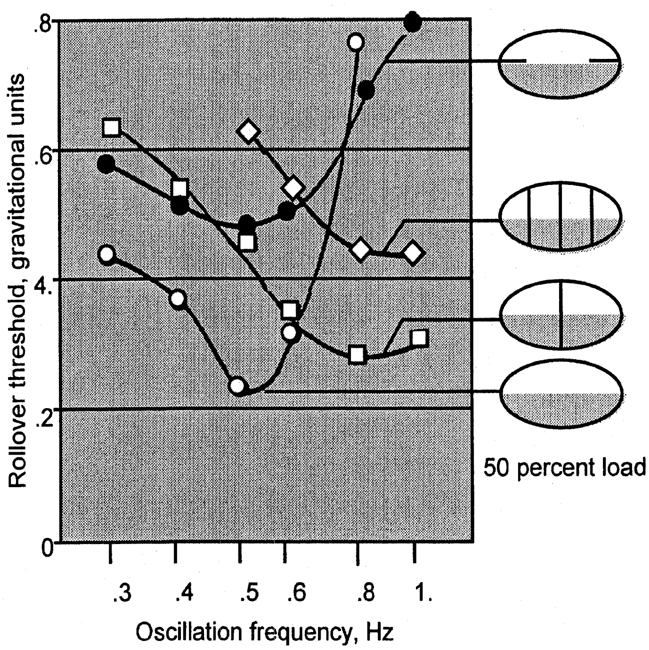


Figure 27. Approximate rollover limits as a function of harmonic oscillation frequency of an elliptic tank with 50 percent load and four types of baffles

Figure 27 shows the effect of different longitudinal baffling arrangements on an elliptical tank design. The plot shows the relative sensitivity to oscillation frequency at a 50 percent load condition in transient maneuvers. Note in the figure that the rollover limit is always less than that of an equivalent rigid load in the frequency range up to 0.5 Hz, representative of normal driving maneuvers. The unbaffled tank is most degraded by slosh at the resonant frequency of 0.5 Hz. Adding one vertical baffle on the centerline greatly improves the degradation in the low frequency range by pushing the resonant frequency to approximately 0.8 Hz (above the frequency of most steering inputs). Adding three vertical baffles further improves the rollover performance, largely by preventing the significant lateral movement of the liquid. Horizontal baffling also aids performance, presumably by interrupting the smooth flow of the sloshing liquid and reducing the overshoot motions.

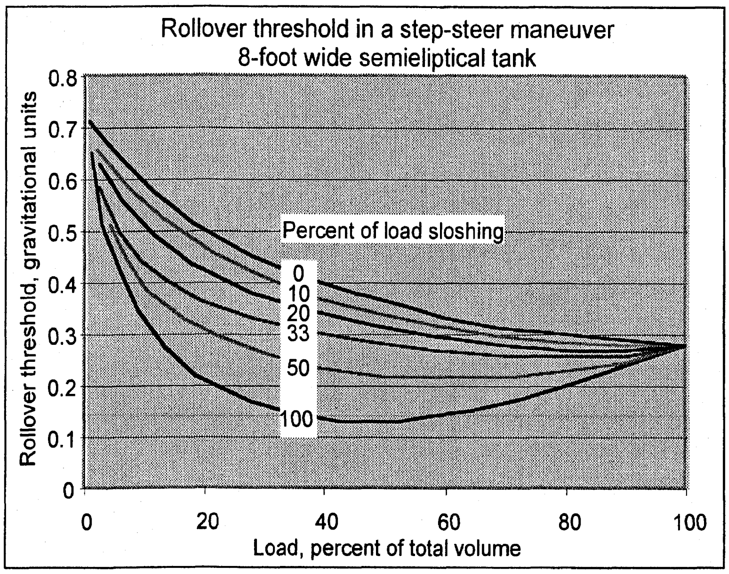


Figure 28. Rollover threshold as a function of load percentage and fractional sloshing volume

Compartmentalization. A second, and far more common, method for improving cornering performance with tankers under partial loading conditions is to subdivide the tank into separate compartments. Ideally, the compartments are completely emptied on an individual basis at a drop spot. In this case, the vehicle is never subject to a sloshing load. The only precaution in this type of use is that the delivery route be planned to empty from the rear of the vehicle first.

When it is not possible to completely empty each compartment, a reduced slosh sensitivity exists, but is often not significant as long as only a fraction of the total load is free to slosh. In these cases, the relevant parameters are the percent load being carried and the fraction of the load that is free to slosh. Figure 28 shows an estimate of the rollover threshold in a transient maneuver for an eight-foot-wide semielliptical tanker as a function of these parameters. The horizontal axis represents the percent (of capacity) to which the vehicle is loaded; whereas, the individual lines represent the fraction of the load that is free to slosh. The worst condition is at about 45 percent load with all of the load free to slosh (the 1.00 line). The key point illustrated in this figure is that, as long as the sloshing load is never more than 20 percent of the total load, the rollover threshold will not be less than that of the fully loaded vehicle.

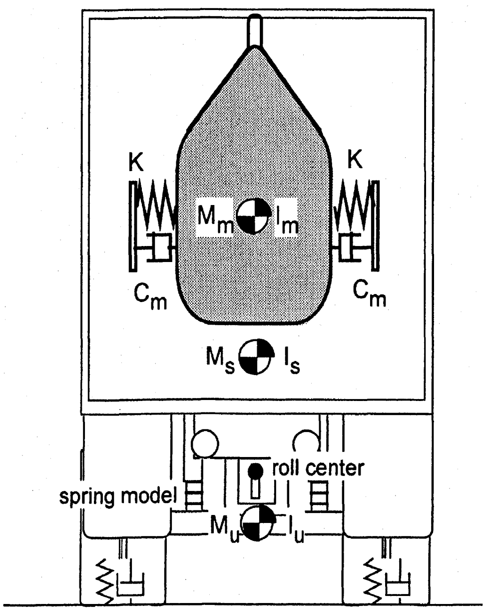


Figure 29. Analytical model of vehicle carrying hanging meat

Hanging-meat loads

The transport of hanging meat in refrigerated vans is another of the special cases of moving loads that has been subject to study, both analytically and experimentally. While these loads are similar to that of livestock in that there is a certain amount of lateral movement possible, they differ in that the loads are suspended from rails on the ceiling of the van. Thus, the load hangs much like a pendulum, free to move within a gap provided for air circulation inside the van, as modeled in figure 29. In addition to the fact that this load has an unusually high cg due to its suspension above floor level, it also has potential

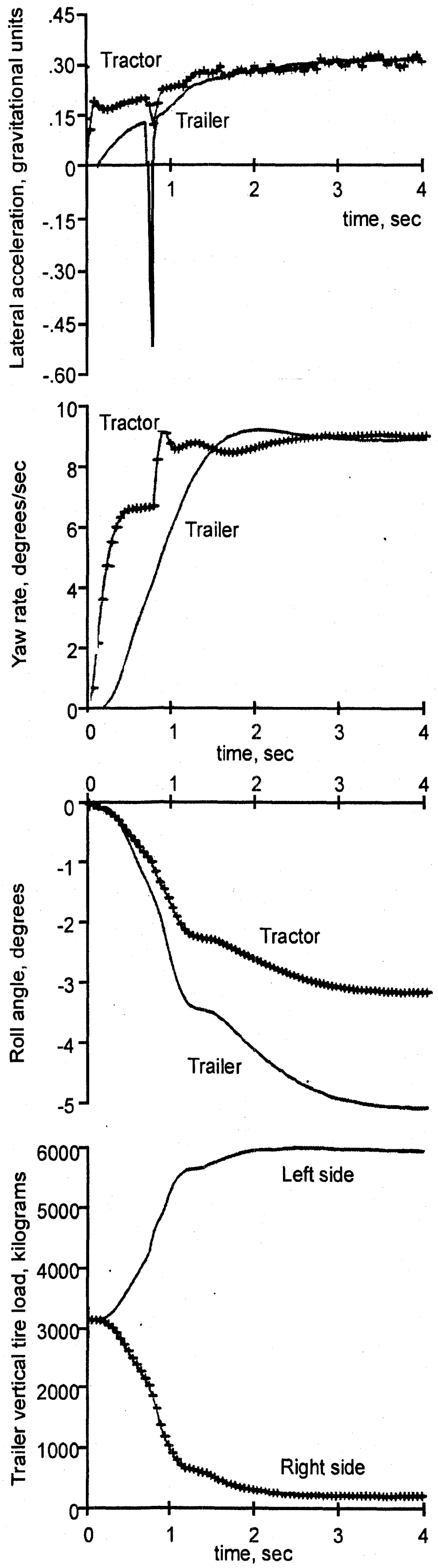


Figure 30. Example responses of a tractor-semitrailer with hanging-meat load in a step-steer maneuver

for two other effects:

— lateral displacement of the cargo cg in either quasi-static or dynamic fashion, and
— a dynamic influence associated with the impact of the load against the trailer wall.

The implications of the first point have been well covered, in principle, in the preceding material. In a quasi-steady turn, the load will swing sideways until it contacts the wall. The lateral air gap, then, represents a limit to the maximum lateral displacement of the cargo cg. This shift is the most significant influence which the mobile quality of the hanging load has on the roll stability of the vehicle.

Dynamically, the pendulum nature of the load establishes a natural frequency which is typically a large fraction of 1 Hz. Thus the dynamic influence of a potentially oscillating load are similar to those for dynamically sloshing liquids.

Regarding impact of the load with the wall of the trailer, figure 30 shows the time-response behavior of a typical tractor-semitrailer with a hanging-meat load in a step-steer maneuver as calculated from an analytical model with features as shown in figure 29. The first graph at the top of the figure shows that the trailer experiences a sharp disturbance in its lateral acceleration history at about 0.75 seconds into the step-steer turn. This is due to the impact of the hanging load with the trailer wall.

The second graph shows that the yaw rate of the tractor is substantially influenced by this impact. It can be expected, therefore, that the driver of the vehicle would be well aware of the impact of the load with the trailer wall and would find it necessary to apply some steering correction.

However, the third graph, which shows the roll angles of the tractor and trailer, and the fourth graph, which shows vertical load on the trailer tires, suggest that the direct influence of this impact on roll stability is minor. A relatively slight "ripple" is apparent in both plots around 1 second. However, the frequency content of the impact event is so high that it substantially exceeds the ability of the vehicle to respond in roll. The primary influence of the hanging meat on roll stability is the lateral translation of its cg per se.

Rollover in dynamic maneuvers

In practice, quasi-static rollover is nearly impossible to accomplish even on the test track. The analyses of chapter 2 assume that the lateral acceleration condition is a given and is sustained (i.e., the condition defining steady state). In practice, a test vehicle can approach rollover quasi-statically either by very slowly increasing turn radius at a constant velocity, or by very slowing increasing velocity at a constant radius. (The former is the more common practice). In either case the, the quasi-static condition can be made to hold reasonably well until the tires of the drive axles lift. At this point, however, the vehicle loses traction and typically "scrubs off" speed such that the lateral acceleration immediately declines and the drive wheels settle back onto the surface. The process may be repeated any number of times. Strictly speaking, there are at least two exceptions which can allow quasi-static rollover. (1) The vehicle may be equipped with a locking differential so that drive thrust can be maintained after lift of tires on the drive axles. (2) Highly compliant (flat bed) trailers may rollover at the rear without lifting drive-axle tires (figure 18). Regardless, in real world events, there is virtually always a dynamic component to the maneuver which, at the least, provides the needed kinetic energy to raise the cg through its apex height after tires of all axles (or at least all axles other than the steer axle) have left the ground. However, as can be seen in figure 31, for vehicles with high centers of gravity, the additional elevation of the cg which is required is not that great.

In the context of rollover, *dynamic* maneuvers are those in which the frequency content of the maneuver (and in particular, the lateral acceleration of the vehicle) approach the natural frequency of the rolling motion of the vehicle. A lightly loading tractor-semitrailer can be expected to have natural frequencies in roll in the range of 2 Hz or more—well above the frequency of steering input which the truck driver can muster even in emergency maneuvers. However, a heavily loaded combination with its payload cg at a moderate height and with suspensions of average roll stiffness is likely to exhibit a roll natural frequency near 1 Hz. A heavily loaded semitrailer with a high cg and with suspensions of less-than-average stiffness can have a roll natural frequency as low as 0.5 Hz. As discussed in the previous chapter (see figure 25), 0.5 Hz in particular is well within the range of excitation frequencies expected in emergency maneuvering. Thus, one can expect the potential for harmonic tuning and related

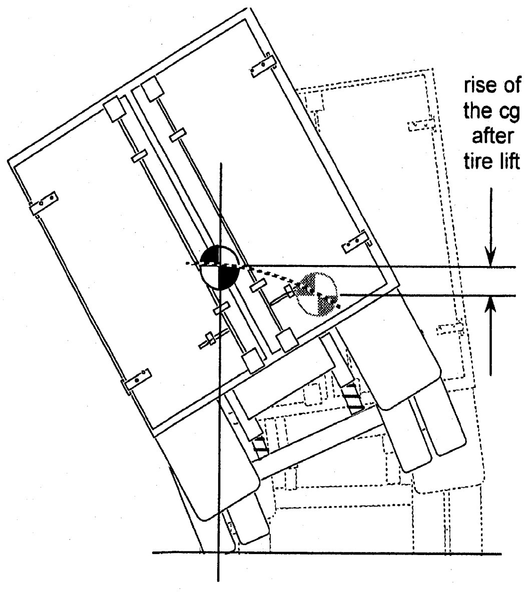


Figure 31. At the least, rollover requires the dynamic momentum required to lift the cg through its apex height.

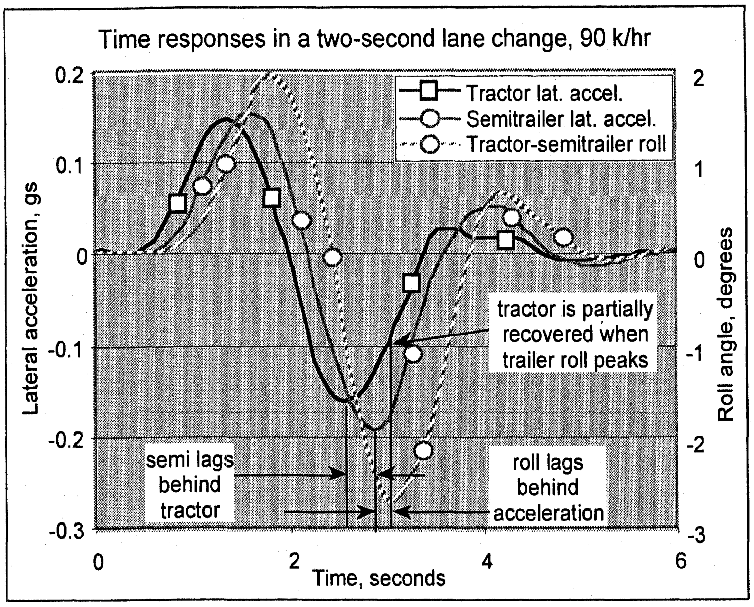


Figure 32. In a dynamic maneuver, the acceleration of the semitrailer lags the tractor and roll lags acceleration

resonant overshoot to promote rollover in transient maneuvers with higher frequency content.

However, higher frequency maneuvers also involve yaw dynamics which can complicate roll behavior. Figure 32 shows the response behavior of a tractor-semitrailer during a simulated, 2-second emergency lane change maneuver.[35] The figure presents time histories of lateral acceleration for the tractor and for the semitrailer and roll angle for the combination. When maneuvering at speed, the semitrailer tends to follow the path of the tractor rather faithfully. Particularly with longer vehicles, this implies a time lag between the actions of the tractor and the trailer. (This is more a result of the tractrix geometry which basically governs the motion of the trailer rather than a true dynamic phenomenon.) When the frequency content of the lateral motion approaches the roll natural frequency, roll motion can be expected to lag lateral acceleration. Both of these effects are readily apparent in the figure. With respect to rollover, it is significant to note that when the trailer reaches its maximum roll displacement, the tractor is well passed its peak lateral acceleration. Consequently, at this critical point, the tractor, with its relatively low cg, is more "available" to resist rollover than it would be in a demanding steady-state turn. Thus, in this maneuver, while roll dynamics are degrading roll stability, the yaw dynamics are compensating to some extent. The situation (even in this relatively simple maneuver) is complex and the net result depends on the tuning of the frequency content of the particular maneuver, the frequency sensitivities of the vehicle in yaw, and the natural frequency

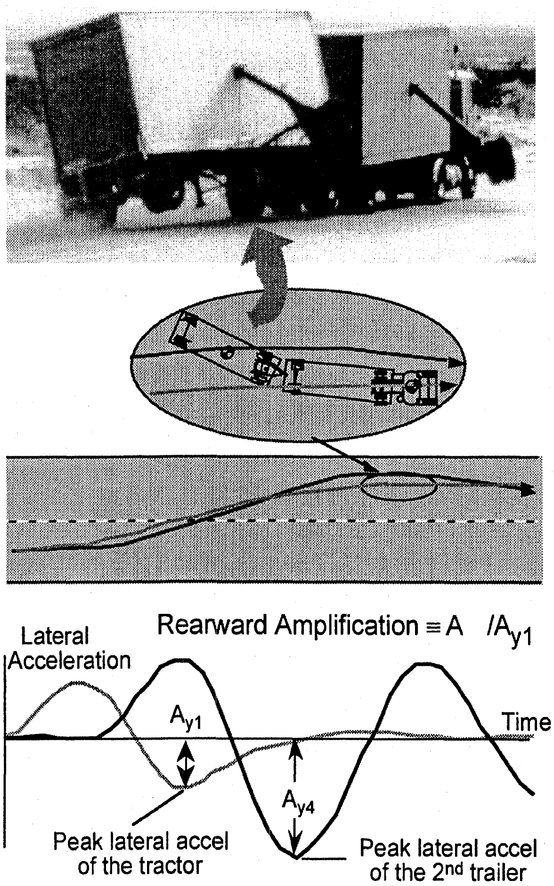


Figure 33. In rapid obstacle-avoidance maneuvers, rearward amplification may result in premature rollover of the rear trailer

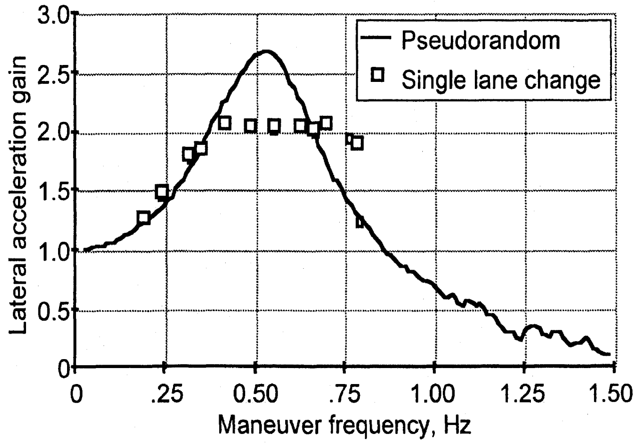


Figure 34. Rearward amplification of a Western double (see figure 33) at 88 k/hr as a function of maneuver frequency and test method[35]

and damping of the vehicle in roll.

Dynamics can play a unique role in the rollover of multiply articulated vehicles. As illustrated in figures 33, vehicles with more than one yaw-articulation joint (i.e., truck-trailer combinations, doubles, or triples) may exhibit an exaggerated response of the rearward units when performing maneuvers with unusually high frequency content. The phenomenon is known as rearward amplification and is often quantified, as shown in the figure, by the ratio of the peak lateral response of the rearward unit to that of the tractor.[35][11]

Figure 35, shows that rearward amplification is a strong function of the frequency content (and the type) of the maneuver. Because rearward amplification is nearly unity at low frequencies, these vehicles behave very well in normal driving. However, since rearward amplification tends to peak in the frequency range characteristic of quick, evasive maneuvers, these vehicles are also quite susceptible to rollover of the rear trailers during emergency maneuvering.

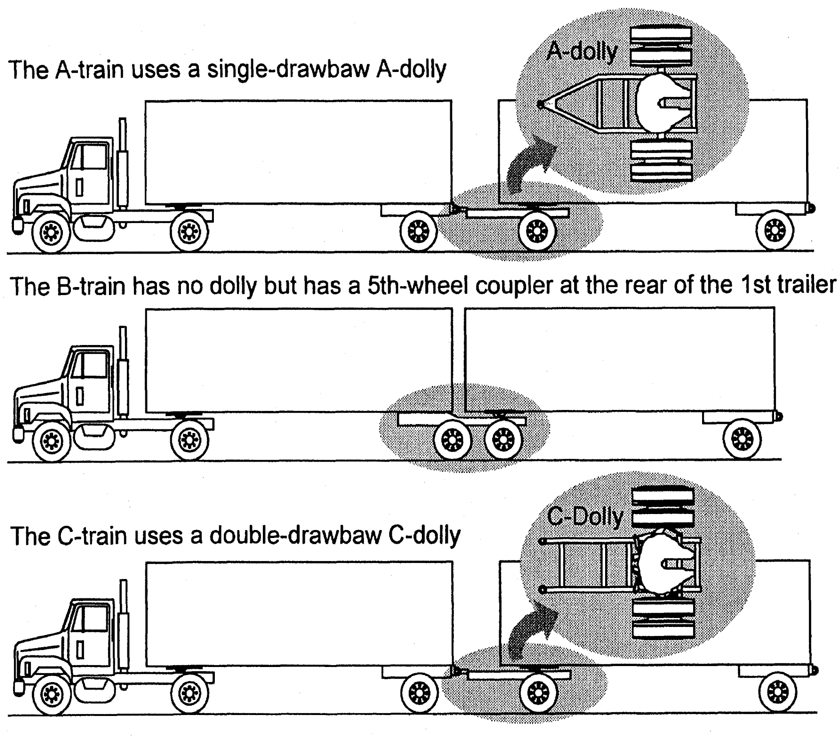


Figure 35. The B-train and C-train, originally introduced in Canada, exhibit less rearward amplification than the standard A-train

Numerous approaches to reduce rearward amplification of multitrailer vehicles have been proposed, most of which are

[11] The subject of rearward amplification is covered extensively in the literature. Early work in the 1960s and1970s [37-40] was followed by a large effort, primarily by UMTRI, in the 1980s and 1990s. This work ranges from linear analyses of closed-form [41], on the stability plane, and in the frequency domain [42], through extensive simulation and test-track experimental studies [43,44,45], to real-world field-test studies [46].

based on different arrangements for coupling trailers. (See [44] for a review of many types of innovative couplings.) The most successful have been the so-called B-train and C-train which are compared to the reference A-train in figure 35. Both of these vehicles eliminate the yaw and roll degrees of freedom associated with the pintle-hitch coupling between the semitrailer and the full trailer. Eliminating the yaw articulation indirectly improves roll stability by reducing rearward amplification. For example, the A-train in figure 35 would typically have a rearward amplification of about 2 (figure 34), but the rearward amplification of the B-train and C-train in the figure would typically be less than 1.5.

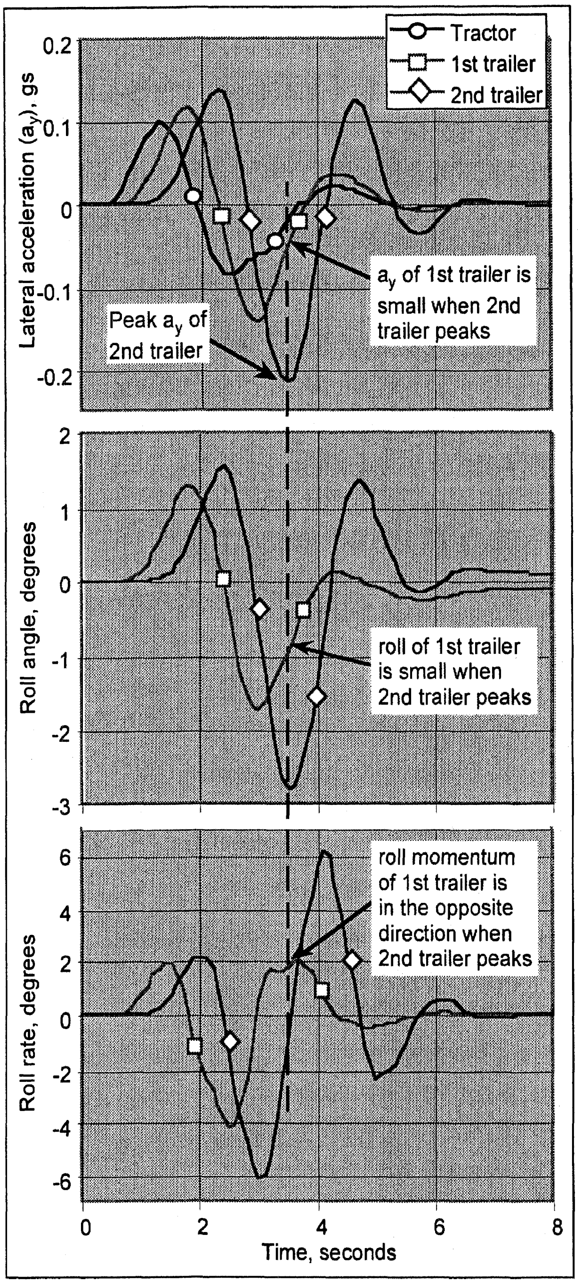


Figure 36. Time histories of an A-train double in an evasive lane-change maneuver

However, by coupling the two trailers in roll, the B- and C-train configurations dramatically improve dynamic roll stability directly. Figure 36 presents time histories of an A-train in a maneuver similar to that of figure 33. The figure shows that the lateral acceleration and roll motions of the two trailers are about 90 degrees out of phase. Thus, when the second trailer reaches its critical condition of maximum lateral acceleration and roll angle, the first trailer has passed its peak and returned to near-zero in these two measures, and actually has substantial roll momentum in the opposite direction. Thus, when these two trailers are coupled in roll as in a B- or C-train, the vehicle can perform very severe lane changes (i.e., with peak lateral accelerations of the *tractor* on the order of 0.5 g) without experiencing rollover as it is extremely difficult for one trailer to “drag over” its out-of-phase partner. [44] (Of course, the mechanical loads on the coupler and dolly frame may be very high in such maneuvers, introducing the risk of mechanical failure of these parts.)

In recognition of this powerful mechanism for improving dynamic roll stability, Ervin introduced a performance measure known as the dynamic load transfer ratio (DLTR). [43]

$$\mathrm{DLTR} = \left|\sum_{i=m}^{n}(F_{Li} - F_{Ri})\right| \Big/ \left|\sum_{i=m}^{n}(F_{Li} + F_{Ri})\right| \qquad (1)$$

Where:

F_{Li}	is the vertical load on the left-side tires of axle i
F_{Ri}	is the vertical load on the right-side tires of axle i
m	is the first axle of the roll unit
n	is the last axle of the roll unit

DLTR is defined for each roll unit of a vehicle, i.e., for each group of units of the vehicle which rolls independently of other units. (For example, the A-train of figure 35 has two roll units, the tractor semitrailer and the full trailer, but the B-train and C-train of that figure are each single roll units.) DLTR can be calculated for each instant during a dynamic maneuver according to equation 1. The maximum value so determined is the DLTR of the vehicle for that maneuver. DLTR is zero for a laterally symmetric roll unit at rest and is unity when all the tires on one side of a roll unit have lifted from the ground.

Within a given class of vehicle configuration, DLTR transfer has been shown to be largely a direct result of the static rollover threshold (SR) and rearward amplification (RA). In 1990, Winkler and Bogard showed that, over a broad range of A-train doubles, rearward amplification and static rollover threshold are very good predictors ($r^2 = 0.91$) of DLTR. [47] Similarly, McFarlane et al., obtained very strong correlations between DLTR and static rollover threshold within some 19 (fairly narrow) vehicle configurations. At the same time, the differences in the correlation models among the various configurations was very substantial. [48]

The broad messages to be taken from these findings are twofold: (1) the most powerful means for improving dynamic rollover stability of *a given vehicle configuration* is to increase static roll stability or, in the case of multiply articulated vehicles, to decerease rearward amplification; (2) dynamic roll stability varies a great deal between configurations, and DLTR provides an effective means to compare this property among all heavy vehicles.

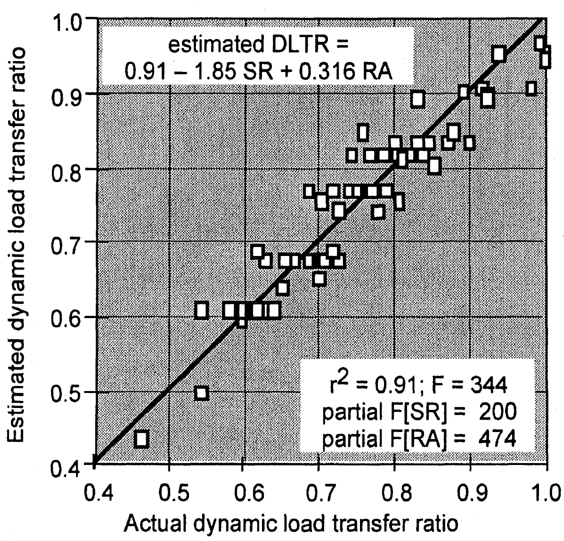


Figure 37. Estimated DLTR versus actual DLTR for a broad range of A-train doubles [47]

4. ROLLOVER AND THE INTELLIGENT HIGHWAY/VEHICLE SYSTEM

Modern electronics are beginning to be applied to the problem of heavy vehicle rollover in the form of so-called *intelligent systems* either on-board the vehicle or within the highway infrastructure.

Since a disproportionate number of commercial-vehicle rollover crashes occur on exit ramps (17 percent according to [49]), highway-infrastructure systems have concentrated on active signing for advisory speeds on exit ramps. The methods explored vary significantly in complexity. For example, Freedman et al. examined the effectiveness of speed-advisory signs employing flashing lights activated when any truck was observed entering the ramp at excessive speed. [50] On the other hand, Strickland et al., described prototype installations which selectively display the message "Trucks reduce speed," based on automated observations of speed, weight, and height of individual vehicles. [51]

At least three approaches aimed at reducing the occurrence of commercial-vehicle rollover through on-board systems are being pursued (at least at the research-and-development level).

Perhaps the most direct method is active roll control which is intended to directly improve the roll stability of vehicles during critical events. Kusahara et al., describe a prototype active roll stabilizer installed on the front suspension of a medium duty commercial truck. [52] Similar devices to be installed on all suspensions of either unit trucks or tractor-semitrailer combinations have been described and are under development at Cambridge University. [53, 54, 55]

Another approach employing on-board intelligence is the roll-stability-advisory (RSA) or rollover-warning systems. A "stability monitoring and alarm system" was advertised for application on commercial vehicles as early as the late 1980s. [56] More recently, RoadUser Research of Melbourne, Australia, has developed and installed a rollover-warning system in limited numbers for use on tank vehicles. The system produces an audible warning for the driver based on real-time measurement of lateral acceleration which is compared to a predetermined, worst-case static rollover threshold for the vehicle. UMTRI has developed a prototype RSA which includes a visual display to the driver comparing the current lateral acceleration of the vehicle together with the static rollover threshold of the vehicle in left- and right-hand turns. [20,57] The rollover thresholds are calculated in real time based on signals from on-board sensors. Thresholds for each new loading condition are determined after only a few minutes of normal driving.

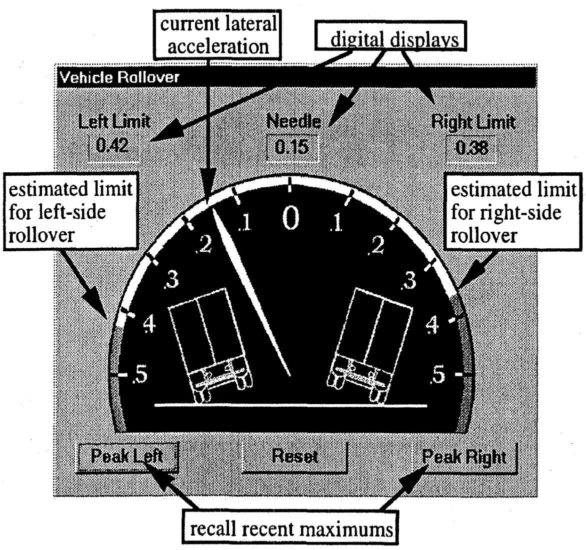


Figure 38. The driver display of the UMTRI prototype RSA

Another approach to the reduction of rollover crashes is active yaw control of the vehicle intended to prevent lateral acceleration from exceeding the rollover threshold of the vehicle. The approach uses selective application of individual wheel brakes to apply appropriate yaw moments and/or to simply slow the vehicle. Palkovics, in association with El-Gindy and others, has published a number of research articles based on this approach, and the ideas presented are being introduced in commercial applications. [58-61] UMTRI has developed and demonstrated a prototype system especially for reducing rearward amplification in multitrailer vehicles. [20,57] Development of this system continues with expectations of commercial application.

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APPENDIX A.

NOTES ON ACCIDENT-DATA SOURCES

GENERAL ESTIMATES SYSTEM: GES is compiled by the National Center for Statistics and Analysis (NCSA) within the National Highway Traffic Safety Administration (NHTSA). The file incorporates data from a probability-based, nationally representative sample of police-reported crashes. It covers all motor vehicle types, including medium and heavy trucks. All police-reportable crashes are included. Approximately 54,000 crashes are sampled each year. The police accident report (PAR) is the sole source of data. Frequencies based on the GES file reported in the tables in this report are national estimates, calculated using an appropriate weighting variable. Since GES is a sample file, estimates are subject to sampling error. The GES file includes data for every vehicle in a sampled crash whether it was towed due to damage or towed for some other reason.

FATALITY ANALYSIS REPORTING SYSTEM: FARS is compiled by the National Center for Statistics and Analysis within NHTSA. The file contains data on a census of fatal traffic crashes within the 50 states, the District of Columbia, and Puerto Rico. FARS includes records for all crashes involving a motor vehicle on a trafficway that resulted in the death of a vehicle occupant or nonmotorist within 30 days of the crash. Trained employees within each state code over 100 data elements from a variety of state documentary sources. These data are then transmitted to a central computerized database and compiled into the FARS file by NHTSA.

TRUCKS INVOLVED IN FATAL ACCIDENTS: The University of Michigan Transportation Research Institute produces the TIFA file. TIFA contains detailed information on all medium and heavy trucks involved in fatal crashes in the United States, including Alaska and Hawaii. TIFA consists of a random sample of straight trucks with no trailers and tractor-semitrailers (as recorded in FARS) and all remaining medium and heavy trucks involved in a fatal crash. The file combines information from the FARS file, police accident reports, and comprehensive telephone interviews conducted by UMTRI research staff. TIFA includes most FARS variables, supplemented with a detailed description of each involved truck collected by the TIFA interview process. Mississippi does not supply police reports, precluding the TIFA interview process, so truck configuration is derived from FARS variables for Mississippi cases.

TRUCK AND BUS CRASH FACTBOOK: T&BFB is complied by the Center for National Truck Statistics at the University of Michigan Transportation Research Institute. It covers crashes for trucks and buses in reportable crashes. A truck is defined as a vehicle equipped for carrying property and having two or more axles and six or more tires or a vehicle displaying a hazardous material placard. A bus is defined as a vehicle designed to carry at least 16 people. Reportable crashes are those with at least one of the following: a fatality; an injury requiring immediate transportation from the scene for medical attention; a towaway. The factbook is based on multiple data source: Motor Carrier Management Information System, General Estimate System, Fatality Analysis Reporting System, and Trucks Involved In Fatal Accidents.

BUREAU OF MOTOR CARRIER SAFETY: Until the end of 1991, the Bureau of Motor Carrier Safety (BMCS) required all regulated motor carriers in interstate commerce to report any accidents involving their vehicles which involved death, injury, or property damage exceeding $4400. Excluded are accidents involving only boarding or alighting from a stationary vehicle, loading or unloading cargo, or farm-to-market agricultural transport. Reporting was done by the carriers themselves, not by a policing agency. Reporting includes description of the vehicle by configuration, body type, number of axles, length, width, height, gross load, and payload. Accident description is minimal. Accidents are categorized as collision (with another vehicle) or noncollision (single vehicle). Only noncollision accidents are further characterized as rollover, jackknife, and other.

APPENDIX B.

ANALYSES OF THE INFLUENCE OF ROLL STABILITY ON ROLLOVER ACCIDENTS

Percent of accidents involving rollover as a function of roll stability

This analysis is based on BMCS accident data (see appendix A) from the years 1987 through 1991. The BMCS data source is the most appropriate for the purpose among those available in that:

- — the file is relatively large, covering all regulated U.S. motor carriers in interstate commerce;
- — the reporting criteria are sufficiently broad as to provide an accident sample which is relatively unbiased with respect to severity,
- — rollover events are identified, although only in single-vehicle accidents,
- — the description of the commercial vehicle in the accident is relatively extensive.

The disadvantages of the file include the fact that it is limited to interstate carriers. This probably biases the sample (relative to national norms) toward the better, restricted-access road network, rural roads, more experienced drivers, and better maintained vehicles. The data are also self-reported by the carriers, not by a policing agency, and are thought to probably contain more coding errors than other data files. The reporting system was terminated at the end of 1991.

Vehicle roll stability is not, of course, reported in the data. However, vehicle configuration, number of axles, body style, height, width, length, empty weight, cargo weight, and gross weight are all reported. In order to allow the best possible estimate of roll stability, this analysis is limited to accidents involving three-axle tractors in combination with two-axle van semi-trailers with trailer lengths in excess of 12 meters (40 feet) and total vehicle tare weights. This defines the most common commercial vehicle in the U.S. (By far the most common van-trailer lengths in the 1987-1991 time period were 48 and 45 feet.) Given this constraint, it is then possible to estimate rollover threshold based on (1) an assumption of typical tare-vehicle properties, including the relevant tire and suspension properties, and the weights and cg heights of unsprung and empty sprung masses; (2) a reasonable estimate of a representative height for payload cg; and (3) the weight data from the accident file. A presentation of the parameters used appears in [7].[12] Of primary importance is the choice of the nominal payload cg height of 203 mm (80 inches) applied to all loads. An extensive presentation of the rationale for this choice is also presented in [7]. As part of this rationale (and to cull obvious reporting errors from the BMCS data) the analysis is limited to vehicles weighing between 11.4 and 36.4 metric tons (25,000 and 80,000 pounds). The relationship between gross vehicle weight and roll stability which results appears in figure B-1.

[12] The parameters and rationale are, of course, appropriate to vehicles and practices of the 1980s. The results presented in figure B-1 are also.

The analysis is also restricted to single-vehicle accidents (in addition to being restricted to five-axle tractor-van semitrailer combinations). The BMCS reporting system first divides accidents into "collisions" and "noncollisions" (single vehicle). Further description of the accident as a rollover, jackknife, etceteras, is only provided for the single-vehicle subset of accidents. Thus the restriction to single-vehicle accidents.

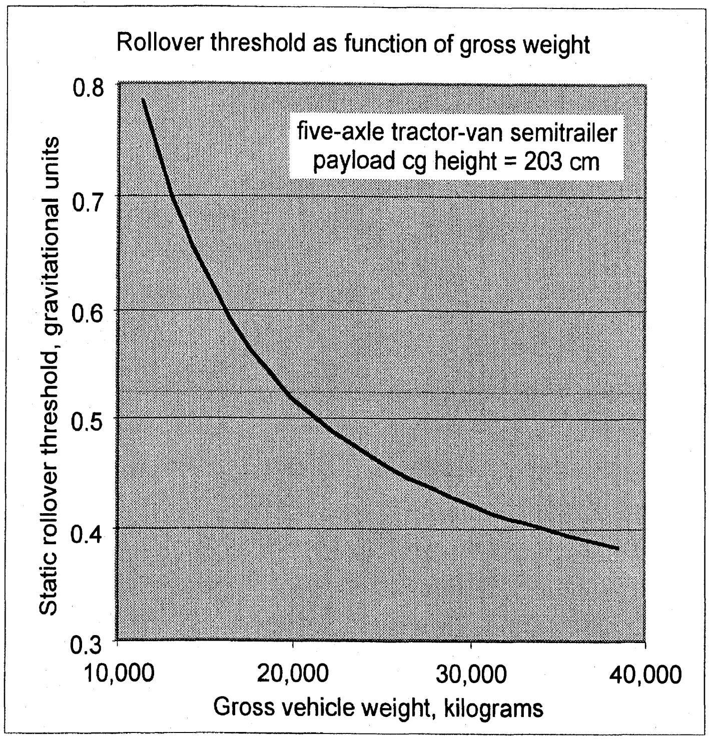

Figure B-1. Estimated rollover threshold as a function of gross vehicle weight

Table B-1 presents the numbers of single-vehicle accidents from the BMCS files of 1987 through 1991 for the vehicles of interest. Counts are provided for accidents characterized as rollovers, jackknifes, all other, and the total. The totals for all five years are also presented. In every case, the counts are presented as a function of gross vehicle weight in categories spanning 1140 kilograms (2500 pounds).

The summated data for all five years was used to prepare figure B-2. This figure shows the percentages of single-vehicle accidents (of the subject vehicle) which were categorized as rollover, jackknife, or others, all as a function of gross vehicle weight. The plot shows clearly that heavy vehicles are more susceptible to rollover, light vehicles are more susceptible to jackknife, and that other accident types are relatively insensitive to weight.

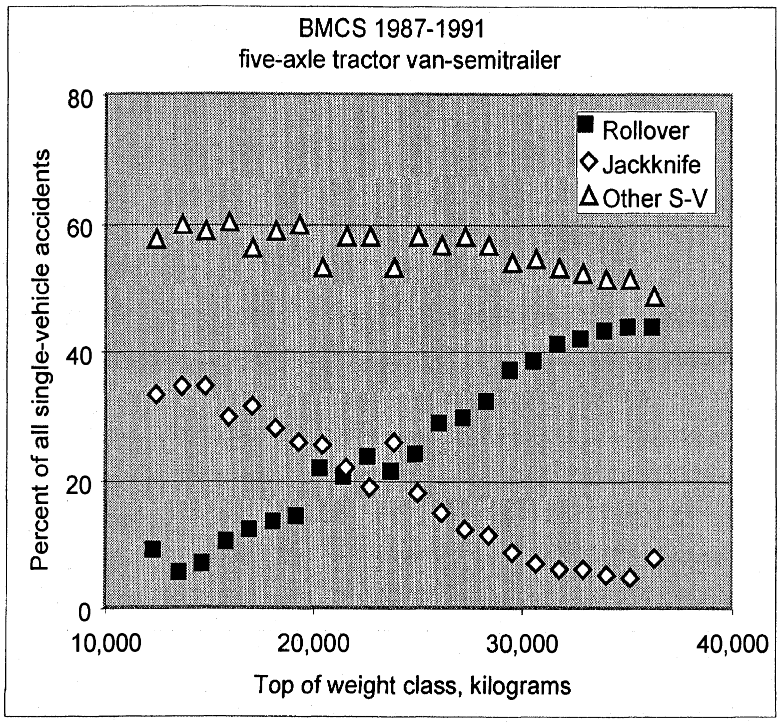

Figure B-2. Types of accidents as a percent of all single-vehicle accidents

Note that percentages for rollover accidents and jackknife accidents are interrelated. That is, while the physics of vehicle behavior certainly suggests that the rollover accidents should generally increase with weight because roll stability decreases with weight, it is also true that some of the relative decrease in rollover for light vehicles could result simply from an increasing number of jackknife accidents for light vehicles.

That is, the physics of vehicle behavior certainly suggests that rollover accidents should generally increase with weight because roll stability decreases with weight, but physics also suggests that jackknife accidents should increase for lighter vehicles because of issues of brake-force proportioning. Then mathematically, it is clearly possible that some of the relative decrease of rollover for light vehicles which appears in figure B-2 could simply be the result of increasing numbers of jackknife accidents. To properly examine the influence of physical stability on the tendency to rollover in accident events, it is appropriate to remove the influence of the jackknife accidents.

Figure 3 in the main text, then, was produced by (1) using the relationship of figure B-1 to determine a representative rollover threshold for the weight categories of table B-1, and (2) calculating the percentage of rollover accidents in single-vehicle accidents excluding jackknife accidents.

Rollover accident rate as a function of roll stability

In appendix G of [22], GES data were used to show that, during the 1988-1990 time period, tractor-semitrailer combinations in the U.S. experienced an average of 8697 rollover accidents per year. The same source shows that this group of vehicles traveled an average of 53,430 million kilometers per year (33,228 million miles per year). From a census survey conducted from 1980 to 1985, Campbell et al. produced the distribution of tractor-semitrailer travel by weight as shown in figure B-3.[23] These data can be combined with the BMCS data of table B-1 and the stability function of figure B-1 to estimate the influence of rollover threshold on rollover accident rate of tractor-semitrailers. The analysis assumes, of course, that the various data are compatible even though they are collected from various sources covering somewhat different times (all nominally in the 1980s, however) and come from different vehicle populations. In particular, it must be assumed that the distribution of rollover probablity for single-vehicle accidents of five-axle tractor-van semitrailers used by interstate carriers (the BMCS

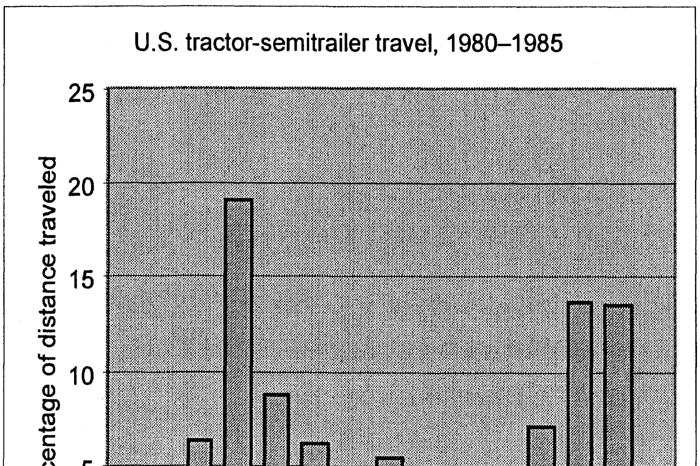

Figure B-3. Distribution of travel of tractor-semitrailer combinations by gross weight data) is applicable across the broader class of all accidents of all U.S. tractor-semitrailer combinations.

The analysis proceeds as follows: (1) Estimate that 95 percent of tractor-semitrailer rollovers occur to vehicles in the weight range covered by table B-1, implying that U.S. tractor semitrailers in that range experience a total of 8262 rollovers per year. (2) Use the five-year-total counts of rollover accidents from table B-1 to obtain the distribution of 8262 rollovers per year by weight. (3) Use the distribution of travel by weight in figure B-3 to obtain the distribution by weight of the 53,430 million kilometers traveled per year by tractor-semitrailers. (4) Normalize appropriate rollover yearly rates by travel yearly rates to obtain rollovers per million kilometers by weight. (The counts from two rollover categories are summed to match with one travel category. The process cannot be completed for some of the lightest and some of the heaviest categories.) (5) Translate weight to stability by the function of figure B-1.

The results of the analysis appear in figure 4 of the main text.

APPENDIX C.
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