# ROLL SENSING OF HEAVY TRUCKS

by


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# ABSTRACT

Rollover, according to accident statistics in North America, is not the most frequent cause of road accidents but is the most significant cause of injuries and fatalities within the trucking industry. The Fatal Accident Reporting System (FARS) for 1991 shows that 23.8% of the deaths occurred in this type of accident. Furthermore, this type of events caused significant damages to the environment and public health as 95% of the accidents involving bulk spillage of hazardous commodities were caused by rollover.

There are many factors which contribute to a rollover event: high center of gravity loads, loads which can shift laterally, excessive speed entering a corner or lane-changing and the need to execute emergency maneuvers to avoid an accident, to name a few.

Because of economic constraints, the only direction which can be acted upon with efficiency today for reducing rollover incidence seems to be the human component of the accidents. It was noticed that the human factor intervenes not only due to the errors but also due to the limitations of the human being.

It is reasonable to assume that incidence of some rollover accidents (those assessed as "preventable" or "potentially preventable" only) might be reduced if early warnings were given to the driver in an incipient stage of the progression towards an accident. The driver thus warned might be able to stop the progression and avoid an accident. Alternately, a warning device might educate a driver to avoid potential accidents.

The objective of this work is to develop a method for sensing the roll motion of a heavy truck. In order to reach the above objective, the mechanical factors associated with rollover are first explored. The roll behavior diagram for a one-axle and a three-axle heavy truck model are presented and discussed.

As it was found by several authors that the rollover threshold of a heavy truck is closely related to the tractor drive axle wheel lift-off, several possible approaches for detecting the progression towards this moment are examined. Both theoretical solutions and already existing devices are critically examined.

A solution for roll sensing by means of tilt sensors is then presented and supported with theoretical and practical arguments. The most important one is that a rollover warning device should be placed entirely on the tractor of a tractor-trailer combination as many tractors pick-up and drop-off a variety of trailers.

The work also presents static, dynamic and on-road tests done in order to assess the suitability of a commercial tilt sensor for the roll sensing purpose. A special arrangement of tilt sensors was mounted on a minivan featuring a truck-like rear axle and on-road tests were performed. All of the tests done show that roll sensing of a road vehicle may be done by means of an arrangement of tilt sensors. This information might provide a good indication of the roll state of a vehicle and, by extrapolation, of its progression towards rollover.

Obviously, further tests performed on heavy trucks are yet necessary before assessing the accuracy of the roll sensing method presented here. Other aspects such as reliability and, especially, cost should be also examined before concluding upon the feasibility of a rollover warning device based on the roll sensing method described in this work.

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# LIST OF SYMBOLS

| Symbol | | Units |
|---|---|---|
| $a_y$ | Lateral (centrifugal) acceleration (parallel to the road surface) | $m/s^2$ |
| $\delta$ | Tire deflection | m |
| $\delta_0$ | Tire initial deflection (when no roll, $\phi_u$=0) | m |
| $F_i$ | Vertical reaction from the ground on the wheels on the inside of a turn | N |
| $F_o$ | Vertical reaction from the ground on the wheels on the outside of a turn | N |
| $\varphi$ | Tilt table angle | rad |
| $\phi_s$ | Roll angle of the sprung mass (vehicle body) relative to the unsprung mass | rad |
| $\phi_u$ | Roll angle of the unsprung mass (vehicle axle) relative to the road surface | rad |
| $g$ | Gravitational acceleration | $m/s^2$ |
| $h$ | Difference between sprung mass center of gravity and roll center heights | m |
| $h_r$ | Height of roll center of the vehicle body | m |
| $h_s$ | Height of the sprung mass (vehicle body) center of gravity | m |
| $h_u$ | Height of the unsprung mass (vehicle axle) center of gravity | m |
| $k$ | Roll stiffness of the sprung mass relative to the unsprung mass due to the vertical stiffness of springs on the axle | Nm/rad |
| $k_{aux}$ | Roll stiffness of the sprung mass relative to the unsprung mass due to all factors other than vertical stiffness of springs | Nm/rad |

| | | |
|---|---|---|
| $k_r$ | Overall roll stiffness due to all factors (both $k$ and $k_{aux}$) | Nm/rad |
| $ks$ | Linear stiffness of a suspension spring | N/m |
| $k_t$ | Linear stiffness of a tire (dual tire) | N/m |
| *LTA* | Left trailing arm | -- |
| *LTR* | Lateral load transfer ratio | -- |
| *RTA* | Right trailing arm | -- |
| *SPR* | Side pull ratio | -- |
| $Sr$ | Lateral offset of the roll center from the centerline | m |
| $Ss$ | Lateral offset of the sprung mass center of gravity from the centerline | m |
| $S_u$ | Lateral offset of the unsprung mass center of gravity from the centerline | m |
| *SSF* | Static stability factor | -- |
| $T$ | Distance from the centerline to the center of a tire (dual tire) | m |
| $T_s$ | Lateral spring spacing | m |
| *TTR* | Tilt table ratio | -- |
| *VB* | Vehicle body | -- |
| $W_s$ | Weight of sprung mass (vehicle body) | N |
| $W_u$ | Weight of unsprung mass (vehicle axle) | N |

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To all of them I dedicate this work.

# Chapter 1

# INTRODUCTION

## 1.1. Rollover accidents involving heavy trucks

Statistics of road accidents in North America prove that, although rollover is not the most frequent of accidents involving heavy trucks, this category of road events causes the largest number of injuries and fatalities among the occupants of heavy trucks. While a report of the FARS (Fatal Accident Reporting System) for 1991 shows that only 4.1% of the large trucks involved in fatal crashes rolled over, it was noticed that 23.8% of the deaths occurred in this type of accidents (see Figures 1.1 and 1.2 and Appendix A).

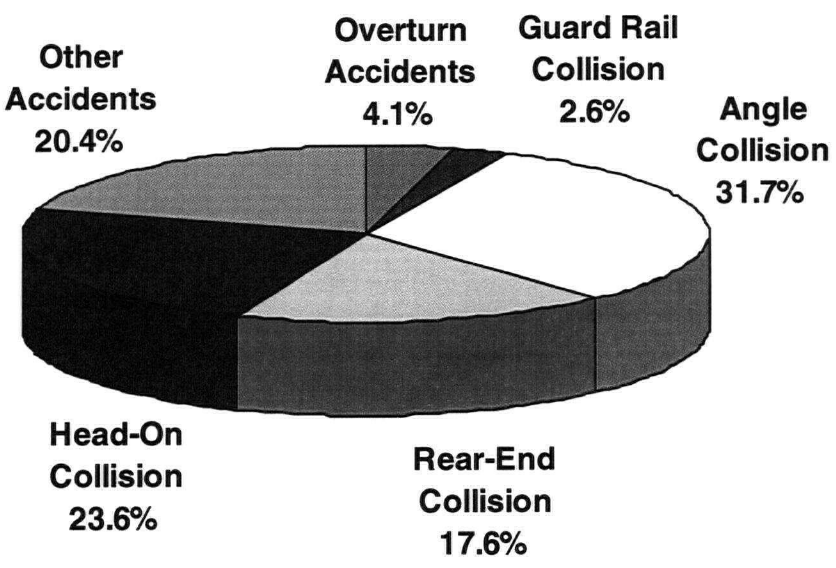



**Figure 1.1.** Statistics of accidents involving large trucks (from FARS 1991)

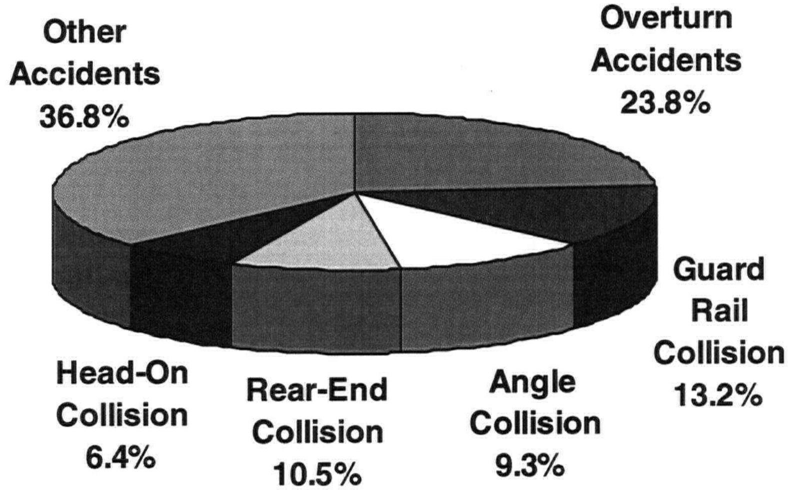



**Figure 1.2.** Occupant fatalities in accidents involving large trucks (from FARS 1991)

According to other statistics, the above numbers are even larger. Krall (1993) shows that after investigating 55,813 accidents of heavy trucks in the US in the period 1980-1990, it appeared that 8494 accidents (i.e. 15.2%) involved rollover collisions (as the primary or a subsequent event) and caused 4837 (i.e. 57% of) driver fatalities. Hinch et al. show in a study done for The National Highway Traffic Safety Administration (1992) that 10,000 people are fatally injured in rollover accidents in the US each year. This type of accident also causes serious damage to the roads and environment as was noticed that 95% of incidents involving bulk spillage of hazardous commodities were caused by rollover (Preston-Thomas and Woodrooffe 1990). Therefore studying the mechanics of this type of accidents with a view to reducing their incidence appears to be very important for present road transport.

Studies and investigations have tried to identify a correlation between rollover accidents and various factors. It was found that, although some other factors depend on the driver (alcohol, age, skills, etc.) or the specific environment (road, weather), the most important factor is the static stability factor *SSF* (defined as one half of the track width divided by the center of gravity height). It was also noticed that the loading condition (i.e. payload magnitude and position of the center of gravity) strongly influences *SSF* and the roll stability of a vehicle in general.

A vehicle's roll stability is characterized quantitatively by its rollover threshold. This measure is expressed in terms of lateral acceleration (or fractions of the gravitational acceleration $g$=9.81 m/s$^2$) and is seen as the maximum lateral acceleration which can be withstood quasi-statically by a vehicle without rolling over. In practice, determining the rollover threshold in an effort to prevent rollover accidents is almost impossible as this threshold is not only different for various vehicles but also often changes for the same vehicle. Investigations have demonstrated that the rollover threshold for all vehicles (and especially for heavy duty articulated ones) is strongly influenced by the position of their payload center of gravity height. Studies reported by Ervin (1983) showed that for heavy vehicles, the rollover threshold decreases by approximately 0.05$g$ for each 25 cm (10 in.) increase in the payload center of gravity height. Other data recorded by the Bureau of Motor Carrier Safety of the US Department of Transportation in the period 1976 to 1979 for 9000 accidents involving various heavy commercial vehicles showed that fully loaded vehicles with rollover thresholds of approximately 0.4$g$ were almost ten times most likely to roll over as empty vehicles with rollover thresholds of approximately 0.65$g$. A more recent study performed in the US and quoted by Preston-Thomas and Woodrooffe (1990) for 39,000 accidents involving passenger cars and light commercial vehicles substantiated the direct dependency between the loading condition and the roll stability of a vehicle although passenger cars usually have much higher rollover thresholds than heavy trucks.

Therefore one can conclude that rollover accidents of heavy trucks are a major cause of concern for road transports and, also, that trucks with large payloads and high centers of gravity are very susceptible to roll over.

## 1.2. Philosophy of a rollover warning device

As some authors have shown (Winkler et al. 1992) roll stability properties of a commercial vehicle can be broken down into two major areas:

- rigid vehicle stability (i.e. static stability factor *SSF*) which is influenced by design;
- roll and lateral compliances of the vehicle which are influenced by suspension characteristics.

From the referred accident statistics and according to other authors' studies and tests (Ervin 1983, Preston-Thomas and Woodrooffe 1990) it appears that improving the static stability factor *SSF* would be the most direct way to reducing rollover incidence. On the other hand, it is clear that improving *SSF* by widening the vehicles and lowering their center of gravity would imply a dramatic change in design for heavy commercial vehicles that is not likely to happen in the foreseeable future. Furthermore, there are also tens of thousands of heavy trucks already in service which could not be affected by such a change.

Some authors as Dunwoody and Froese (1993) proposed other solutions which address the suspension of the vehicle. The above authors described an active anti-roll suspension able to tilt the semi-trailer of a heavy truck inwards in a curve while negotiating a corner. They proved that the action of the active roll control system would have the same effect as lowering the position of the center of gravity and improving the static stability factor *SSF*. Calculations have shown an increase with 20-30% of the rollover threshold of a heavy truck using active roll control suspension compared with the same type of vehicle using conventional suspension. Although ingenious and possibly having highly positive implications for improving the road safety of heavy trucks, these types of solutions do not seem feasible in the near future because of the economic constraints; they would imply additional suspension elements and power units on the vehicle, as well as refined sensors and control units, leading to a reduction in payload and increase of general costs without any immediate financial gain for the operator.

Therefore the only direction remaining today for reducing rollover incidence seems to be the human component of the accidents. For example traffic accident statistics in British Columbia in 1991 showed that over 60% of the commercial vehicles accidents were due to human factors. Among these accidents a large proportion represent those due to unsafe speed (8.36% of the total), improper turning (3.75% of the total) and driver's inexperience (2.09 % of the total).

Restraining the area of discussion to rollover accidents only, it was noticed that the human factors intervene not only due to the errors but also due to the limitations of the human being. No matter how skilled and experienced the driver may be, how accustomed he/she may be with a specific heavy vehicle, the very large variations in payload magnitudes and positions of the centers of gravity as well as the always changing road conditions may impede the human operator from correctly estimating the roll behavior of such a vehicle. Thus, the ability of vehicle drivers to sense progression towards rollover has been proven to be variable from one driver to another and even for the same person in various conditions. It was noticed that the drivers may have no physical sensation of rollover of the rear units of a combination vehicle, especially when there is no roll coupling between the components of the road train (Preston-Thomas and Woodrooffe 1990).

Therefore, as the above referred authors have also done, it appears reasonable to assume that the incidence of some rollover accidents might be reduced if early warnings were given to the driver in the incipient stage of the progression towards a rollover accident. The driver thus warned might be able to stop the progression and initiate a recovery to a safe travel. Even if a timely warning of an incipient rollover were not possible, a rollover warning device could educate the driver as he negotiated several corners, feeding back how close the truck came to rollover in each case.

## 1.3. Benefits and limitations of a rollover warning device

It must be mentioned that not all vehicle accidents involving rollover can be considered recoverable. It is obvious that accidents having rollover as a subsequent event (after a collision, for example) cannot be prevented by such a device. Furthermore, other accidents involving mainly rollover but associated with hitting road-side barriers or off-road excursions into ditches or accidents emerging from extreme maneuvers (such as jackknifing) could also not be prevented by such a device.

However, there still remains two important categories of accidents involving rollover as the first event which could be prevented by a rollover warning device:

- accidents resulting from excessive speeds on roads with small radii of curvature;
- accidents emerging from excessive lateral accelerations encountered in rapid lane changing on highways.

A study done by Sparks and Berthelot in 1989 and referred to by Preston-Thomas and Woodrooffe in 1990 made an effort to identify the cost-effectiveness of a rollover warning device. The authors investigated 661 accidents involving heavy trucks operating in western Canada and northwestern USA. Irrespective of the type of heavy truck, Sparks and Berthelot broke down the total number of accidents into the following mutually exclusive categories:

- rollover;
- hit the ditch;
- (caused by a) third party;

- jackknife;
- hit animal;
- driver error;
- equipment malfunction.

From Table 1.1 which shows the results by accident type it appears that 22.8% of the 661 accidents were classified as rollover accidents.

**Table 1.1.** Accident distribution by accident type (from Preston-Thomas and Woodrooffe 1990)

| **Category of Accidents** | **Number of Accidents** | **Percent of Accidents** |
|---|---|---|
| Rollover | 151 | 22.84 |
| Hit Ditch | 65 | 9.98 |
| Third Party | 263 | 39.79 |
| Jackknife | 51 | 7.72 |
| Hit Animal | 102 | 15.43 |
| Driver Error | 19 | 2.87 |
| Equipment Malfunction | 9 | 1.36 |
| Total | 661 | 100 |

Furthermore, the authors of the above study divided the rollover accidents into the following categories:

- preventable;
- potentially-preventable;
- non-preventable;
- unknown if preventable.

The term "preventable" characterized an accident in which the vehicle driver would have been able to avoid rollover if an effective rollover warning device had been installed on the vehicle and have been properly operating at the time of the accident. A "potentially-preventable" accident was one that might have been prevented in the above conditions provided that the warning device had been accurate enough and the driver skillful enough. The "non-preventable" accidents (involving hitting and rolling over lateral physical barriers such as curbs, rollover following jackknifing and off-road excursions into ditches) were judged as unpreventable by such as a warning device which could not provide any useful information to the driver. The "unknown if preventable" accidents did not offer enough data in order to know in which of the other categories they might have fallen. It is to be underlined that the above classification is highly subjective and was made by Sparks and Berthelot based upon the description of each accident.

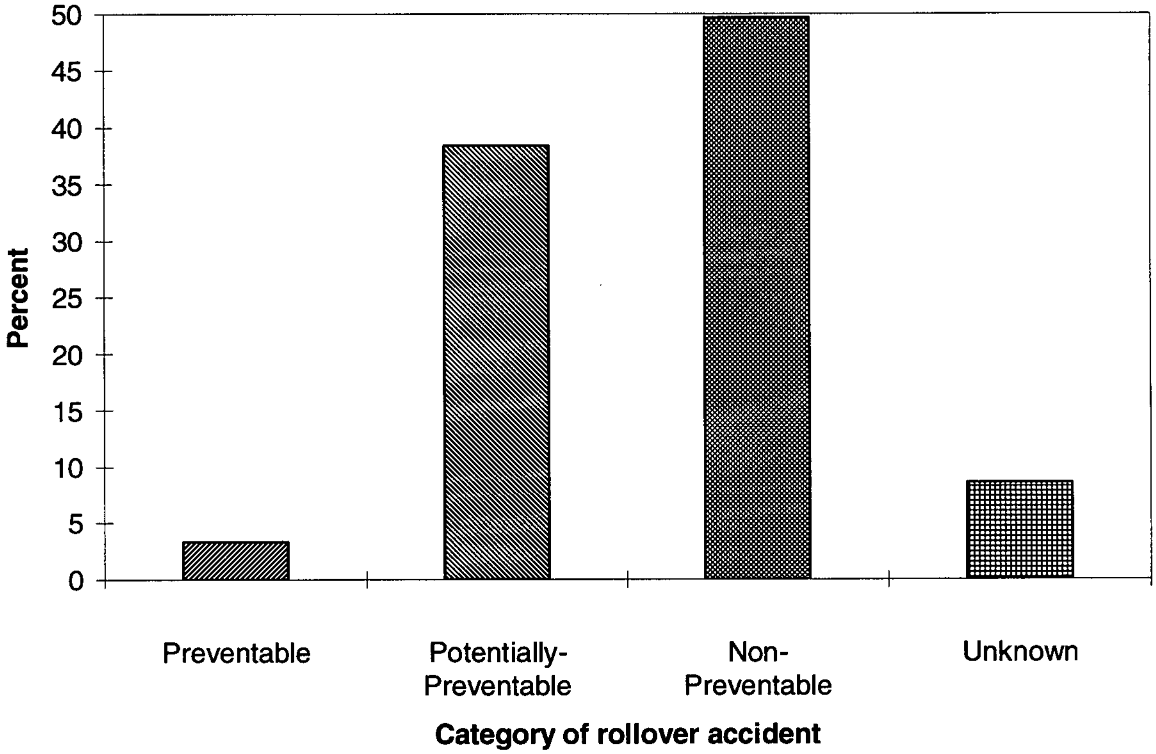



**Figure 1.3.** Percent of category of rollover accidents (from Preston-Thomas and Woodrooffe 1990)

From Table 1.2 and Figure 1.3 one can see that only 3.3% of the total rollover accidents were judged as preventable, 38.4% as potentially-preventable and 8.6% unknown if preventable. The remaining, almost half of them(49.7%), were judged as non-preventable.

**Table 1.2.** Accident distribution by rollover classification (from Preston-Thomas and Woodrooffe 1990)

| **Category of Accidents** | **Number of Accidents** | **Percent of Accidents** |
|---|---|---|
| Rollover | 151 | 100.00 |
| Preventable | 5 | 3.31 |
| Potentially-Preventable | 58 | 38.41 |
| Non-Preventable | 75 | 49.67 |
| Unknown | 13 | 8.61 |
| Total of Rollover Accidents | 151 | 22.84 |
| Potential Rollover Accidents | 116 | 17.55 |
| Other Accidents | 394 | 59.61 |
| Total | 661 | 100.00 |

According to the above statistics, a rollover warning device which could prevent only 3.3% of the accidents would seem of very little use. However, the referred report identifies also the accident cost (the monetary part only), the number of injuries and the number of fatalities for each general category of accident described. Thus, although rollover accounted for only 22.8% of the accidents, it was responsible for 61.8% of the total damages, 29% of the injuries and 28.6% of the fatalities.

**Table 1.3.** Rollover severity by rollover classification (from Preston-Thomas and Woodrooffe 1990)

| Category of Rollover Accidents | Percent of Damages | Percent of Injuries | Percent of Fatalities |
|---|---|---|---|
| Preventable | 3 | 3.7 | 0 |
| Potentially-Preventable | 37.8 | 18.5 | 30 |
| Non-Preventable | 48.8 | 77.8 | 60 |
| Unknown | 10.4 | 0 | 10 |
| Total | 100.0 | 100.0 | 100.0 |

Narrowing this analysis for the rollover accidents only, it was found (as shown in Table 1.3 and Figure 1.4) that the preventable rollover accidents are responsible for 3% of the total monetary costs, 3.7% of the injuries and 0% of the fatalities. As Preston-Thomas and Woodrooffe (1990) showed, however, care should be taken in extrapolating these results to larger numbers of accidents since the samples utilized were relatively small.

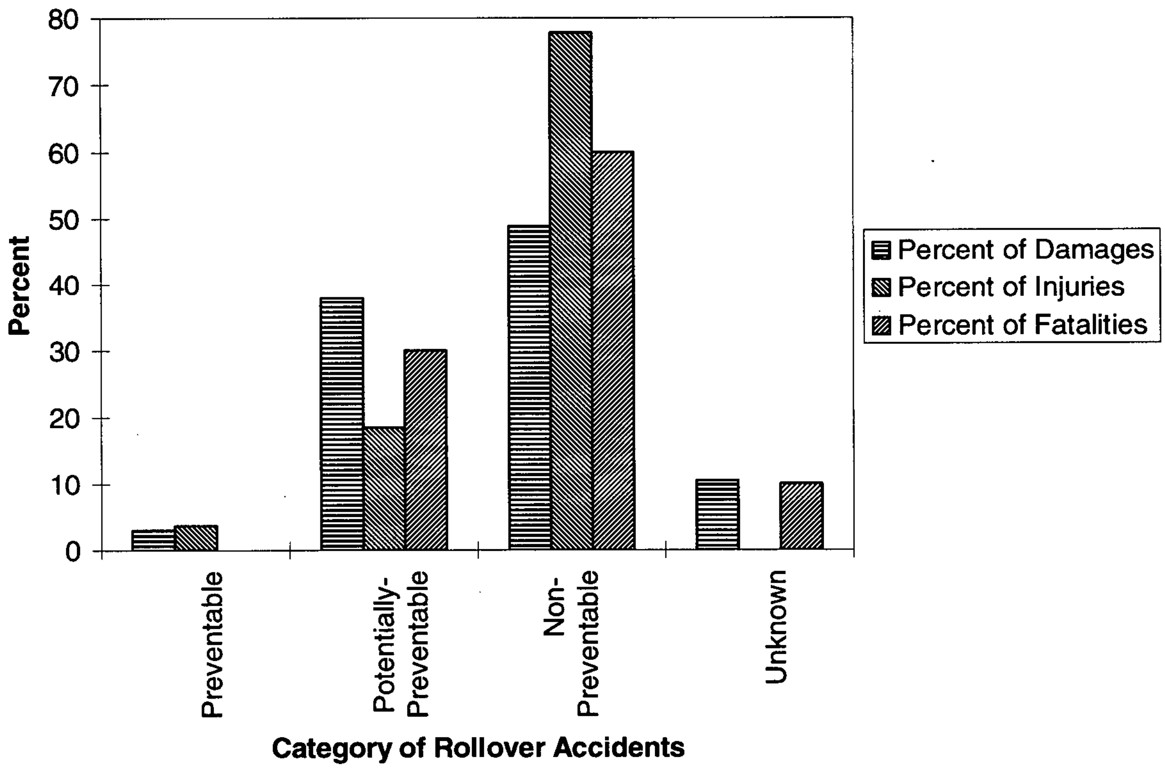



**Figure 1.4.** Percent of damages, injuries and fatalities by rollover classification (from Preston-Thomas and Woodrooffe 1990)

Sparks and Berthelot estimated also the cost-effectiveness of a rollover warning device assuming that this one would eliminate only the preventable rollover accidents. It was shown that such a device would be expected to eliminate one injury only (3.7% of the total) and $106,069 in damages (3% of the total). After estimating the average damage per vehicle per year caused in preventable rollover accidents, Sparks and Berthelot estimated that at the purchase of a new truck the operator could afford to invest no more than $152.95 for a rollover warning device if the decision had only an economic support.

As in practice it might be expected that an accurate warning device would eliminate some of the potentially-preventable accidents as well, the amount the operator could afford to invest would increase substantially. For example, assuming that a rollover warning device would also eliminate 25% of the potentially-preventable accidents, the amount that the operator may, economically, justify spending would be 4 times greater-namely $633. What is more important, injuries and fatalities would decrease from 27 to 25 cases and from 10 to 9 cases, respectively.

It is also to be mentioned that the above report does not consider other expenses caused by accidents such as claims for disability pensions and workers' compensations, costs to eliminate the damages caused to the environment and public health due to the spillage of hazardous commodities. More than that, the study does not evaluate what is, in fact, invaluable- human life and health.

In conclusion, rollover accidents involving heavy trucks, while not the most common type of accident, is sufficiently common to bear closer study. Further, the damage caused by a rollover accident, both economically and in terms of human suffering, is greater than that caused by an average trucking accident. Of the available avenues of approaching to decreasing the incidence of rollover accidents, the most promising in the short term is the rollover warning device which would warn the driver of an impending rollover accident. The driver might be able to react to the warning and thus avert an accident. Irrespective of whether the device could warn the driver sufficiently early to avert an accident, the rollover warning device would serve as an educational tool, feeding back to the driver how close the truck came to rolling over with each turn.

The target of the present work is to propose a roll sensing method different from those found in the available literature. This roll sensing method, if validated by future research and experiments, might eventually lead to the design of a rollover warning device for heavy trucks.

# Chapter 2

# BACKGROUND AND LITERATURE

The aim of this chapter is to present the state of developments for the prevention of rollover accidents involving heavy trucks as found in the literature. Developments comprise both theoretical work modeling the roll behavior of heavy trucks and full-scale experiments with heavy trucks. The theoretical modeling can be further subdivided into three-dimensional and two-dimensional models, with the type of model chosen to emphasize one or more aspects of the problem. Experimental studies also fall into two general categories: static and dynamic tests. The static tests, using a tilt table, offer relatively quick, inexpensive and controllable testing at the expense of some loss of reality. Dynamic testing, while closely simulating actual conditions during rollover accidents, are very difficult and expensive to perform. Some of the required characteristics of a rollover warning device will be deduced from this review.

## 2.1. Modeling

### 2.1.1. Three-dimensional models

Some of the theoretical approaches consider a heavy truck as a three-dimensional system composed of several bodies. These bodies are linked by connections having various damping and stiffness characteristics. By means of Newton's law or d'Alembert principle, a system of equations of motion is obtained. As this system of equations tends to have a great complexity when the number of degrees of freedom taken into account increases, some possible movements (such as torsional stiffness of the tractor and trailer frames) are usually neglected. Some authors, as for example Rakheja and Piche (1990), include torsional stiffness in the model but in a simple way by considering the torsion connection as an ideal spring.

In order to diminish the complexity of the system of equations, researchers make some other assumptions as well. Their models neglect, in most cases, the nonlinearities of the suspension and, especially, tires as well as the backlash of the suspension leaf springs and fifth wheel connection. Other assumptions are to neglect the movements of particular loads (such as liquids or meat) during cornering and their influence on load shifting.

In spite of the above simplifications, the resulting systems of equations are still complicated due to the complexity of the vehicle combination, as one can notice in studies done by Rakheja and Piche (1990) and Hasegawa et al. (1990) and as is also stated by several researchers in *Mechanics of Heavy Duty Trucks and Truck Combinations* (1986), etc. These systems are then solved for dynamic simulations having as inputs, in most cases, the vehicle speed and the steering angle of the front wheels and, only in some cases, the speed of steering of the vehicle combination. The usual target is to illustrate the time variation of the roll angle of the vehicle combination.

The most common conclusion, existing in all quoted references, shows the dependency of the roll angle on the lateral acceleration of the tractor-trailer combination. However, depending on the assumptions made and on the features taken into account, some additional conclusions are also provided. For example Rakheja and Piche (1990) state that the most important indication of the roll tendency of the vehicle would be the measurement of the tire loads. As measuring tire loads is a difficult task for practical cases, Rakheja and Piche suggested that an acceptable alternative would be to monitor the lateral acceleration of the trailer during low speed cornering and the roll angle of the semi-trailer axle during high speed directional maneuvers, respectively. The authors do not elaborate about how the above data could actually be acquired.

Another idea found in the available literature is to measure the dynamic suspension loads at the rear-most axle of the trailer. These loads become negative at tire lift-off (springs on one side change from compression to tension). Since this change of state would be detectable for leaf spring suspensions only, for air spring suspensions an alternate method is proposed- namely detecting the axle roll angle as this angle was found to be related to the tires and suspension loads.

Larocque and Croce (1985) state that a relation exists between the lateral acceleration and the suspension springs deflection. As a consequence, they suggest that the monitoring of roll tendency might be achieved by measuring the suspension spring deflections.

Most studies, such as that done by Nowak et al. (1990), provide additional conclusions that roll motion also depends on other influence parameters such as uneven roads, side slope, payload magnitude and position of the center of gravity. They recommend taking into account further improvements for a more accurate investigation: additional degrees of freedom, suspension lash and coulombic friction, nonlinearities in tire models, asymmetries in springs and tire stiffnesses, etc. Studies such as that done by Hasegawa et al.

(1990) which took into account the torsional frame rigidity, but usually made other simplifications, suggest that particular attention should be given to the matching of this stiffness and that of the suspension springs.

### 2.1.2. Two-dimensional models

Other theoretical approaches, such as those used by Preston-Thomas and Woodrooffe (1990), Ervin (1983) and MacAdam (1982), consist of regarding the vehicle combination in cross section as being lumped to a single sprung mass supported on a single axle. This approach allows more accurate investigations of the movements of the two vehicle masses in transversal section, during roll motion. Therefore the more complex dependencies between roll angles of both the vehicle body and the axle, the changes in wheel loads and the lateral acceleration can be emphasized.

Thanks to the reduced number of bodies and degrees of freedom of the heavy vehicle combination, such models may also consider some refinements such as the suspension nonlinearities as in *Mechanics of Heavy Duty Trucks and Truck Combinations* (1986). However, the tire behavior or suspension and fifth wheel backlash are still neglected. Only qualitative judgments about these realistic features could be encountered in some cases such as Ervin (1983) and in *Mechanics of Heavy Duty Trucks and Truck Combinations* (1986).

Obviously, two-dimensional models cannot properly take into account the load transfer between axles and cannot model realistically dangerous regimes leading to rollover such as braking on slopes. However these models reach the same conclusion as the three-dimensional models- namely that the lateral acceleration may be the best indication for the roll behavior of the motor-vehicles. Further, the rollover threshold is usually defined in terms of lateral acceleration (or fractions of the gravitational acceleration, more precisely).

However studies such as that done by Preston-Thomas and Woodrooffe (1990) show that the lateral acceleration is altered by several factors among which the most important is the specific load situation. They state the same conclusions as researchers who utilized three-dimensional heavy truck models (such as Rakheja and Piche 1990)- knowing the tire loads would provide a good indication about the roll situation of the vehicle. Preston-Thomas and Woodrooffe (1990) suggest replacing tire load measurements with suspension load sensing in an effort to detect the progression towards rollover. This should be done at least for the trailer axle of a heavy truck combination. They believe (as have other authors such as Ervin et al., Fancher et al.) that a nondimensional coefficient expressing the load transfer between the wheels of the same axle during roll motion would show clearly enough the roll state of a heavy truck. Therefore the authors suggest the utilization of an indicator characterizing the difference between the tire loads (or the suspension spring loads, actually) of the same axle and denoted as the lateral load transfer ratio.

As a conclusion it is obvious that three-dimensional models are closer to the reality. However, in order to limit the complexity of the mathematical part and of the computer software based on it, the spatial models found in the literature neglect many of the influence factors (such as load type and dynamic position, suspension and tires nonlinearities, backlash, torsional compliance of frames, etc.). Therefore these models might not be able to give a sufficiently accurate description of the roll behavior of heavy trucks. On the other hand, two-dimensional models, thanks to the reduced degrees of freedom, allow easier investigation but only in-plane. The fact that both modeling approaches can be found in the literature indicates that both have advantages and disadvantages and can be used depending on the aspects of the problem which are emphasized.

## 2.2. Experimental tests

### 2.2.1. Static tests

Static tests use a tilt table and consist of identifying the dependencies between tilt angle, roll angle and wheel loads. Obviously this kind of testing cannot take into account any dynamic effects but it allows some investigation of the suspension and tires nonlinearities as well as suspension backlash. These features are evaluated in most cases in a qualitative manner illustrating their contribution to the roll behavior of the heavy truck but not allowing the accurate estimation of rollover thresholds as stated by Ervin (1983).

Static tests usually deal with three static rollover metrics which are believed to provide fairly good information about a vehicle's propensity to rollover:

- Static Stability Factor (*SSF*) defined as the ratio of half track and the height of the center of gravity of the vehicle body:

$$SSF = \frac{T}{h_S} \tag{2.1}$$

- Tilt Table Ratio (*TTR*) defined as the tangent of the angle of the tilt table at which the vehicle rolls over:

$$TTR = \tan\varphi \tag{2.2}$$

- Side Pull Ratio (*SPR*) defined as the ratio of the lateral force required to destabilize a standing vehicle and the weight of the vehicle. The vehicle is standing and the lateral force is applied through its center of gravity.

The studies found in the literature show convergent opinions of the authors about these tests. Thus Winkler et al. (1992 and 1993) have generally optimistic conclusions about the accuracy and repeatability of the static tests. They stress, however, that accuracy refers to a comparison between static tests and not to the on-road (dynamic) roll behavior of the vehicle which is rather different. Although the tilt table tests are necessary and more accurate than the simple calculation of the *SSF*, the physical analogy is not perfect and the results may be in error with respect to the true stability limit. Winkler et al. (1993) show that both the "lateral" (parallel to the tilt table) acceleration and the "vertical" (perpendicular to the tilt table) acceleration are scaled down by a factor of $\cos\varphi$. This means that at a $45^{\circ}$ tilt table angle, for instance, both loads represent about 70% of the real ones. Further, on the tilted table, due to the suspension and tires compliance, both the effect of the lateral load shift and the effect of the rise of the center of gravity of the vehicle during roll tend to be overestimated. The global effect is that tilt table tests provide an unrealistically high estimate of static stability limit. Therefore *TTR* is an estimate of the static roll stability threshold and not the roll stability threshold itself. The same conclusion is suitable for *SPR* as well because during the tests the vehicle is subjected to a lateral force lumped in its center of gravity and not distributed over the entire vehicle as in reality. No accurate evaluation of the error could be found in the available literature.

The idea that tilt table testing is only an estimation of the reality can be found from other authors as well. Nowak et. al (1990) show that Australian studies have proven the angle $\varphi$ to be about 20...23$^{0}$ for smooth tilting of heavy trucks. These authors show that the limit angle is obviously smaller for real situations due to some external factors (wind gusts, uneven soil, load shifts, etc.). Tamny (1993) shows that when performing tilt table static tests, special care must be exercised to avoid inhibiting tire deflection on the tilt table as tire compliance has a greater effect on the results than suspension compliance.

### 2.2.2. Dynamic tests

This kind of testing takes into account the vehicle behavior during on-road motion and the influence of the lateral acceleration on the roll movement. However, these tests provide less information compared with those obtained on the tilt table, and most of them try to identify the rollover threshold expressed in terms of acceleration values (Ervin 1983).

The scarcity of information on dynamic tests can be explained obviously by their difficulty (regarding safety aspects and vehicle instrumenting) and cost. Das et al. (1993)

proposed a method for performing some dynamic tests at low speeds and extrapolating the results to high speeds. They showed that road tests done close to the rollover threshold situation require, for safety reasons, some outriggers that are costly and time consuming for installing. Further, as some modifications are necessary to the vehicle, its inertial parameters and roll response might be affected and the tests have poor repeatability.

The solution proposed by the above authors is to perform low-speed tests with no special modifications to the vehicle, measure the accelerations relatively easily and then, through a computer simulation, extrapolate the results to threshold values. Although the authors do not give details about addressing the nonlinearity problems (for suspension and tires with large deflections as it should be in the threshold zone), the study shows that good correlation was obtained for lane-change maneuver tests.

Therefore one can conclude that, despite the incomplete reproduction of reality, tilt table tests seem a fairly good and financially acceptable method of investigation of rollover propensity of heavy vehicles. These tests have quite good reproducibility and, therefore, make possible investigations regarding roll behavior of similar vehicles or of the same vehicle in various loading conditions (Garrott 1992, Chrstos and Guenther 1992, Winkler and Bogard 1993). On the other hand, on-road dynamic testing, especially at regimes close to the rollover threshold, is not an easy task because of safety reasons. Some tests consist of rolling a reinforced vehicle until the outriggers touch the ground (Ervin 1983). Other tests are done with usual vehicles but are conducted far below the rollover threshold (Das et al. 1993). In these conditions the investigation is, again, incomplete and is reliable only in a qualitative sense in most cases.

## 2.3. Remarks about previous studies

The studies found in the literature try to model the tractor-trailer combinations roll behavior due to its importance for road safety.

According to some opinions such as at Rakheja and Piche (1990) and Preston-Thomas and Woodrooffe (1990), it appears to be necessary to monitor the roll behavior of heavy trucks as the next level towards safer road transport. However there are no widely accepted conclusions or concrete solutions regarding how this goal could be achieved in practice.

Preston-Thomas and Woodrooffe (1990) discuss the feasibility of a rollover warning device starting from using existing load cells for on-board weight measurement. Although the load cells already exist and are even mounted on some vehicles, the study proves that such a device is not appropriate for economic reasons. This happens because the accidents having rollover as the first event and which could be prevented by such a warning device represent only a small percentage (as presented in Chapter 1) of the total number of accidents involving heavy trucks. On the other hand, McLaughlin (1993) thinks that a complex warning (and control) system is, however, feasible and would pay for itself in 3 to 5 years but he does not elaborate on the cost aspects as well as on other issues related to heavy truck road transport (installation, maintenance, reliability, drivers' role, etc.).

In any case, as rollover accidents of heavy trucks are responsible for the majority of damage to truck payloads plus injuries and fatalities of truck occupants, there is a serious reason for pursuing development of a viable rollover warning device.

## 2.4. Characteristics and requirements of a rollover warning device

Several authors outlined some features that a rollover warning device should possess. According to the available literature, Preston-Thomas and Woodrooffe (1990), presented a quite complete design problem for such a device. They stated that a rollover warning device should not only have the ability to warn the driver accurately and early about the progression towards rollover of the vehicle, but it must also have some other characteristics in order to perform its functions and be attractive for the operator.

Preston-Thomas and Woodroffe divided the features they considered important for such a warning device into several categories:

*(1) General*:

• <u>inexpensive</u> -low cost is important as such a device is a pure expense; the order of magnitude of the cost was discussed in Chapter 1;

• <u>reliable</u> -the trouble-free service interval must be long enough to make it accepted by truck owners;

• <u>suitable for all vehicles</u> -its installation should require minimal design changes for all kind of existing or new vehicles;

• trailer-mounted devices easily removable -by the operator when the trailer is dropped off.

*(2) Installation and durability*:

• no separate cables between different vehicle units -in order to not affect the existing wiring;

• environment protection -all components and cables must withstand water, salts, impact with small foreign objects, external temperatures;

• no or minimal field calibration required -in order to make it independent of human operator skills.

*(3) Performances*:

• warning must be independent of payload and center of gravity height;

• warning must be early enough by a consistent margin of safety in order to provide sufficient time for the driver to make the necessary corrections based on the output of the warning device; "sufficient time" means that the driver would have a period of time long enough to allow him/her to perceive the warning signal, to take a decision and to act- the consequence being a useful recovery maneuver before the rollover becomes irreversible;

• not significantly affected by usual variations in the design characteristics (such as position of axles or torsion stiffness) of heavy vehicles;

• unaffected by road irregularities (small bumps, potholes encountered on some rough roads);

• unaffected by some conditions encountered during low-speed maneuvering (for example changes in road elevation, road slope, etc.). These conditions should not produce warnings about an impending rollover.

*(4) Other features*:

• warning of device failure (the device should be able to perform a self-test when it is turned on and notify the operator when it is not functional);

• adjustable audio and visual warnings -could be an analog scale showing the state of the watched indicator and some audible warning which would emit sounds with increasing amplitude and frequency as the rollover threshold approaches.

Therefore it appears that a rollover warning device should possess many features (some of them contradictory). However, in evaluating these features one should not neglect the principal mission of this device- namely to sense as accurately as possible, and under any conditions, the progression towards rollover of a heavy truck.

# Chapter 3

# MECHANICS OF ROLLOVER

## 3.1. General remarks

As indicated previously in Chapter 1, rollover can have a large number of causes. Not all of these causes could be prevented by a rollover warning device. Investigations of accidents have shown that a rollover warning device might be useful in avoiding two types of accidents for heavy trucks:

- rapid lane-changing on highways;
- excessive speed on highway ramps.

As a warning device is supposed to warn the driver about a imminent accident so the driver can react, accidents in which rollover is due to various causes may not be prevented by a warning device. A warning device could not help avoiding events such as "tripping accidents" (involving hitting and, then, tripping over a road-side barrier) as well as rollover following off-road excursions, jackknifing or collision. However, as most of the above accidents as well as "pure rollover" are adversely affected by excessive speed and loss of stability, a rollover warning device could be useful in a broader sense for both preventing accidents and educating the truck drivers to drive safely.

This chapter presents the mechanics of rollover for one major cause of accidents in this category, namely rapid lane-changing on highways. The situation for the other major cause of rollover accidents, namely fast driving on highway ramps, could be developed in a similar manner except that the speeds and accelerations are lower for the latter case as well as road angle should also be taken into account.

A rollover usually occurs when the lateral acceleration becomes high enough to tip the vehicle over. As shown by Preston-Thomas and Woodrooffe (1990), the most significant external factor leading to rollover is the ratio of the forces applied to the tires in the direction parallel to the road surface relative to the forces applied in the direction normal to the surface. Because these forces arise from accelerations (centrifugal and gravitational, respectively), the force ratio is identical to the corresponding acceleration ratio. Vehicle speed, road radius of curvature and road slope are primary determinants of the magnitudes of the above two accelerations. However, as Preston-Thomas and Woodroffe (1990) showed, the above factors do not significantly alter the maximum value of the acceleration ratio that a vehicle can withstand without rolling over. Therefore, one can assume, for simplicity, that the vehicle is operating on a horizontal surface. The only forces acting on the vehicle are those due to the lateral acceleration (which is strictly horizontal) and gravitational acceleration (which is strictly vertical).

## 3.2. A one-axle model

### 3.2.1. Presentation and assumptions

The first step in examining the roll behavior of a truck-trailer combination is to consider the behavior for a vehicle model with a single axle operating under quasi steady-state conditions. This model neglects differences which exist between axles (as loads, stiffnesses, tires, etc.) as well as the roll-coupling between various axles (by frame torsional stiffness, fifth wheel roll stiffness, etc.). The model also does not take into account other refinements encountered in the literature (such as Mallikarjunarao 1982 referred by Preston-Thomas and Woodrooffe in 1990) as it neglects nonlinearities (suspension and tires) or backlash (fifth wheel and suspension). Despite its simplicity, the model is sufficient to describe the general roll response of most heavy commercial vehicles (Preston-Thomas and Woodrooffe 1990). All the commercial vehicles have generally rigid axles and suspension elements connecting these axles (which constitute the unsprung mass) to the vehicle body (which is the sprung mass).

Irrespective of the suspension type (with leaf springs or air springs, or even, sometimes, with rubber elements and torsion bars) the lateral acceleration $a_y$ developed during cornering will cause the vehicle to tilt and roll outwards as shown in Figure 3.1.

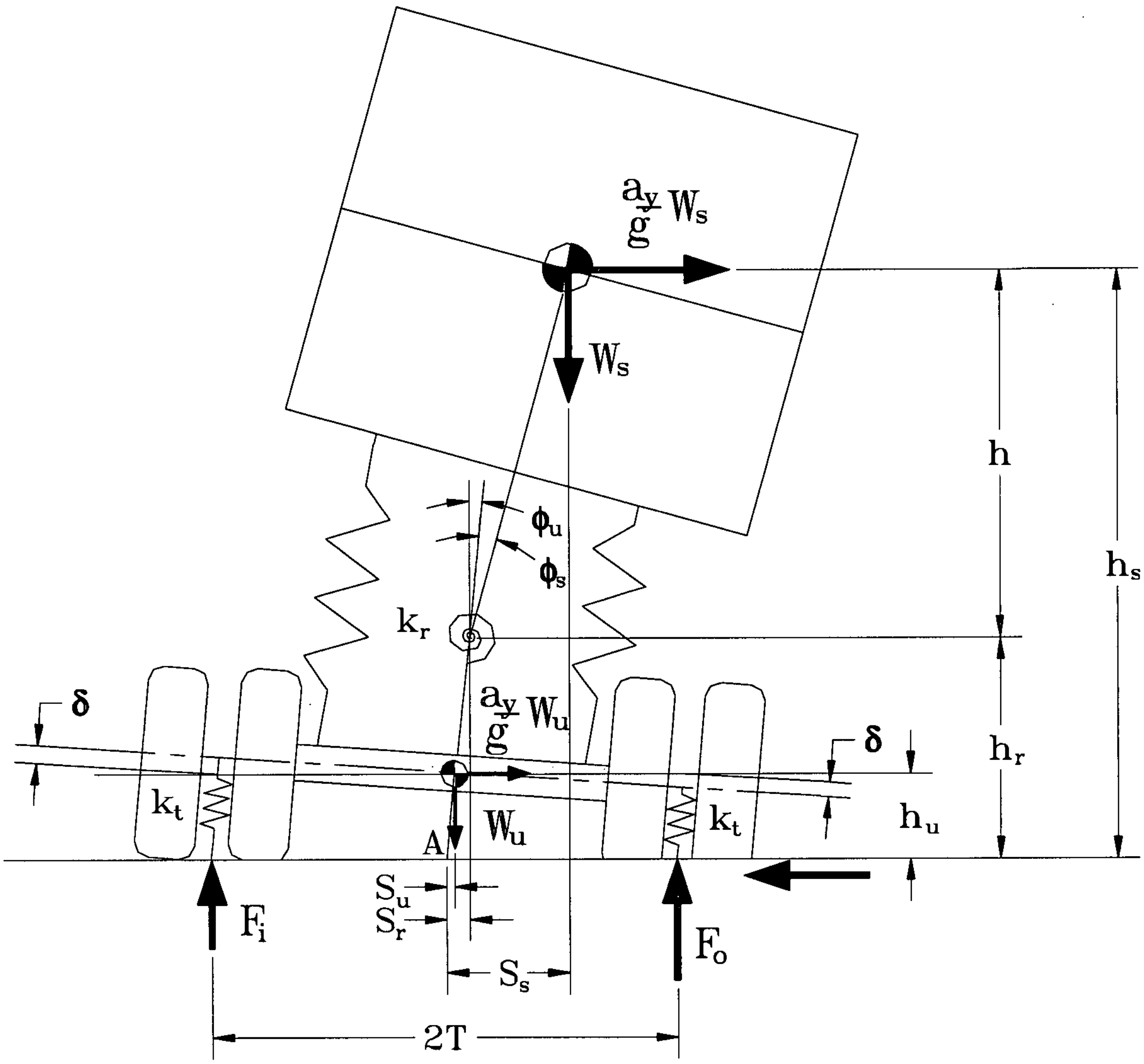



**Figure 3.1.** A one-axle roll model of a heavy truck subjected to roll

Because of the change in rolling radii of the tires, the axle rolls through an angle $\phi_u$ relative to the road surface. Further, the vehicle body rolls through an additional angle $\phi_s$ relative to the axle due to the compliance of the suspension springs.

### 3.2.2. Roll angles calculation

According to the above assumptions and simplifications, determination of roll angles for the vehicle body and axle is relatively easy.

The two suspension springs provide a roll resisting stiffness $k$ that can be approximately calculated with the formula:

$$k = 2k_S \left( \frac{T_S}{2} \right)^2 \tag{3.1}$$

The total roll resisting stiffness of the suspension is usually higher as additional anti-roll bars or simply the longitudinal twisting of leaf springs increase this stiffness.

Therefore the total roll stiffness becomes:

$$k_r = 2k_S \left( \frac{T_S}{2} \right)^2 + k_{aux} \tag{3.2}$$

When rolling, the vehicle body rolls with respect to the roll center. This is a point situated on the vehicle longitudinal middle plane (for symmetrical vehicles). Its height depends on the characteristics of the suspension. The general approach for a leaf spring suspension is to consider this roll center as being placed at the same height as the spring connection ends (Preston-Thomas Woodrooffe 1990, Untaru et al. 1982). For more accurate studies one can use practical measurements such as those done by Fancher et al. (1986)

proving that the roll center position for trucks and trailers has variations from 450 to 800 mm from the ground level.

In Figure 3.1 it is assumed that the lateral acceleration acts strictly horizontally through the centers of mass of both the axle (situated at the height $h_u$ above the ground) and of the sprung mass (situated at the height $h_s$ above the ground).

The opposing effect to the roll tendency is given by the moment of the vertical forces $F_i$ and $F_o$. These forces are applied from the ground to the tires and are assumed to remain constant in direction and position relative to the tires (in the mid position between the twin tires or in the center of the contact patch for single tires), regardless of the roll angle of the axle $\phi_u$.

The roll of the unsprung mass with angle $\phi_u$ during cornering causes a lateral shift outwards of the center of gravity of that mass, $S_u$. Similarly, the roll center shifts with $S_r$ and the sprung mass center of gravity with $S_s$ from the vertical passing through the same midpoint $A$, due to the combined roll effect of both masses.

For angles of roll assumed small, the above distances can be expressed as follows:

$$S_u = h_u \tan(\phi_u) \approx h_u \phi_u \tag{3.3}$$

$$S_r = h_r \tan(\phi_u) \approx h_r \phi_u \tag{3.4}$$

$$S_s = h_r \tan(\phi_u) + h\tan(\phi_s + \phi_u) \approx h\phi_s + h_s\phi_u \tag{3.5}$$

The forces acting on the vehicle and causing progression towards rollover are the lateral forces $W_s a_y/g$ and $W_u a_y/g$ acting through the centers of gravity of the two masses (sprung and unsprung, respectively) where $g$ is the gravitational acceleration and $W_s$ and $W_u$ are the weights of the same masses, respectively.

By considering the equilibrium of moments about the roll center, one can calculate the sprung mass roll angle $\phi_s$:

$$\sum M_{roll\_center} = 0 = -k_r\phi_s + W_s\left(S_s - S_r\right) + hW_s\frac{a_y}{g} \tag{3.6}$$

$$0 = -k_r\phi_s + W_s\left(h\phi_s + h\phi_u + h\frac{a_y}{g}\right) \tag{3.7}$$

$$\phi_s\left(k_r - hW_s\right) = W_s h\left(\phi_u + \frac{a_y}{g}\right) \tag{3.8}$$

$$\phi_s = \frac{W_s h\left(\phi_u + \frac{a_y}{g}\right)}{k_r - hW_s} = \frac{\phi_u + \frac{a_y}{g}}{\frac{k_r}{W_s h} - 1} \tag{3.9}$$

The resisting forces $F_i$ and $F_o$ can be expressed in terms of the vertical tire stiffnesses $k_t$, the tire deflection $\delta_0$ when $\phi_u$=0 and the change in deflection $\delta$ that occurs at a certain roll angle $\phi_u$:

$$F_i = k_t\left(\delta_0 - \delta\right)$$
$$F_o = k_t\left(\delta_0 + \delta\right) \tag{3.10}$$

On the other hand, the change in deflection $\delta$ can be expressed in terms of the roll angle:

$$\delta = T\phi_u \tag{3.11}$$

Considering moments about the point $A$ on the ground, midway between $F_i$ and $F_o$, a second relationship between the roll angles of the two masses can be found:

$$\sum M_A = 0 = \left(F_i - F_o\right)T + W_u S_u + W_u \frac{a_y}{g} h_u + W_s S_s + W_s \frac{a_y}{g} h_s \tag{3.12}$$

By substituting equations 3.3 , 3.4, 3.5, 3.10 and 3.11 into 3.12 one can obtain:

$$0 = -2k_t T^2 \phi_u + W_u h_u \phi_u + W_u \frac{a_y}{g} h_u + W_s h \phi_s + W_s h_s \phi_u + W_s \frac{a_y}{g} h_s$$

$$\phi_u \left(2k_t T^2 - W_u h_u - W_s h_s\right) = W_s h \phi_s + \frac{a_y}{g}\left(W_u h_u + W_s h_s\right) \tag{3.13}$$

By substituting equation 3.9 into 3.13 one can obtain the explicit expression for the axle roll angle:

$$\phi_u \left(2k_t T^2 - W_u h_u - W_s h_s\right) - \frac{a_y}{g}\left(W_u h_u + W_s h_s\right) = \frac{W_s h \left(\phi_u + \frac{a_y}{g}\right)}{\frac{k_r}{W_s h} - 1}$$

$$\phi_u = \frac{\frac{a_y}{g}\left(\left(W_u h_u + W_s h_s\right)\left(\frac{k_r}{W_s h} - 1\right) + W_s h\right)}{\left(2k_t T^2 - W_u h_u - W_s h_s\right)\left(\frac{k_r}{W_s h} - 1\right) - W_s h} \tag{3.14}$$

The most important conclusion of the above calculation is that roll angles of both vehicle masses (that of the body depends linearly on that of the axle) are proportional to the lateral acceleration $a_y$.

### 3.2.3. Roll behavior diagram for the one-axle model

This diagram shows the roll response of a single-axle vehicle model (Mallikarjunarao 1982 referred by Preston-Thomas and Woodrooffe 1990, *Mechanics of Heavy Duty Trucks* *and Truck Combinations* 1986). The diagram is based upon a moment balance equation assumed to exist all the time before the vehicle actually tips over:

$$M_a = M_r - M_g \tag{3.15}$$

where:

- $M_a$ is the primary overturning moment with respect to a point on the road surface and caused by the lateral acceleration of the mass center;
- $M_r$ is the roll resisting moment about a point on the road surface halfway between the tires and resulting from the roll-resisting forces applied by the road surface to the tires on the axle;
- $M_g$ is the secondary overturning moment about a point on the road surface halfway between the tires and caused by the lateral shift of the center of gravity.

The roll behavior diagram plots the above moments against the roll angle in the right hand side and against the lateral acceleration in the left hand side, respectively. The roll behavior of the one-axle model can be easily understood by analyzing the diagram of Figure 3.2.

The primary overturning moment, which is the main cause of vehicle roll, increases linearly and, theoretically, with no bound with lateral acceleration. At the same time the axle roll-resisting moment increases linearly with roll angle (which was proven previously to be proportional to the lateral acceleration). When the latter moment increases, the load of the axle is progressively transferred to the outer tires of the vehicle. Therefore the roll resisting moment increases only until it reaches a maximum value (point *A* on the diagram) which corresponds to the situation when all the load of the axle has been transferred to the outer tires. Any further increase of acceleration (and, therefore, roll angle) can only cause the inside wheels to lift-off with no effect on the roll-resisting moment (therefore there is a flat zone in the diagram).

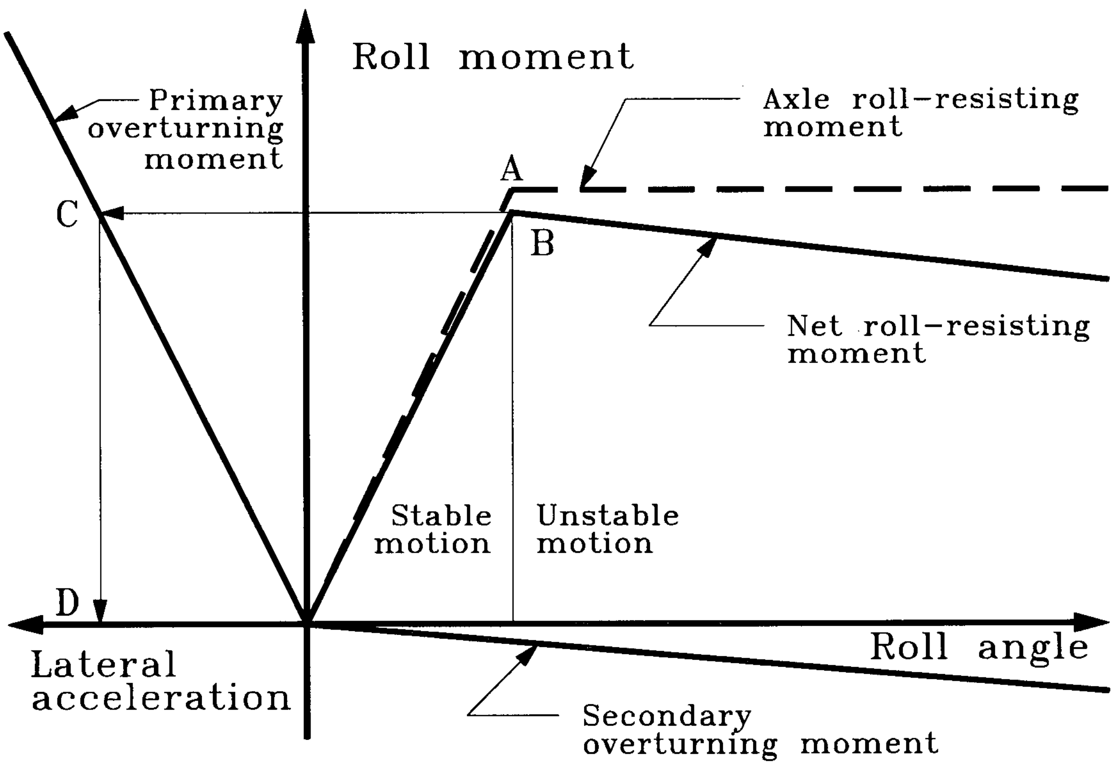



**Figure 3.2.** Roll behavior diagram for an one-axle model of a heavy truck

The secondary overturning moment diminishes the roll resisting moment and, unlike the latter, continuously increases with roll-angle even beyond the lift-off point. The difference between the roll-resisting moment and the secondary overturning moment gives the net roll-resisting moment of the one-axle vehicle. The slope of the net roll-resisting moment provides information about the roll stiffness of the vehicle (the greater the slope the greater the stiffness). It can be seen that this net moment increases linearly from zero to a point *B* (lower than *A* on the moments scale) after which it decreases continuously as the roll angle continues to increase.

On the same horizontal line with point *B* one can find point *C* in the left hand side of the diagram. This is the maximum overturning moment that can be applied to the vehicle without causing roll. The corresponding lateral acceleration (which in most of the studies is defined as the "rollover threshold") can be found by drawing a vertical line from point *C* to point *D* in the left part of the diagram. This acceleration is usually expressed in terms of gravitational acceleration units and it separates two domains:

- stable motion (before the rollover threshold- the internal wheel has not lifted-off and the vehicle, although tilted, is not tipping over in the assumed steady-state conditions);
- unstable motion (beyond the rollover threshold- the internal wheel has lifted and all conditions for the vehicle tipping over are met).

A more refined analysis should take into account that all moments of the roll behavior diagram feature nonlinearities as follows:

• The primary overturning moment has a decreasing slope because the height of the mass center decreases as the roll angle increases. A rough estimation shows that for a large roll angle ($30^{\circ}$) the actual moment would be about 14% less than the value of the diagram.

• The roll resisting moment has a nonlinear shape (until the wheel lift-off) due to both nonlinearities and backlash of the suspension and fifth wheel as well as to tire behavior. As tire and suspension characteristics are different for various vehicles, it is difficult to estimate the type of the error: conservative or nonconservative. In any case, the effect of backlash (featured by the fifth wheel and by the trailer and tractor drive axle if equipped with leaf springs) is similar to a "softening" of the suspension. The consequence is an increase of the limiting roll angle and a decrease of rollover threshold, as demonstrated by Ervin (1983).

• The secondary overturning moment appears to have, in reality, an increasing slope because the lateral offset of the center of mass depends on the tangent of the angle of roll and not on the angle itself, as assumed in formula (3.5). A rough calculation shows that for a roll angle of $30^{\circ}$ the nonconservative error in expressing the secondary overturning moment would be about 4%.

Therefore a diagram which would consider the nonlinearities of the three moments would look as in the Figure 3.3. As a conclusion, neglecting nonlinearities in the roll behavior diagram generates some errors. In general, the direction and magnitude (and, therefore, importance) of these errors are difficult to be estimated without knowing the geometric and elastic characteristics of a specific vehicle. Despite its simplicity, the linear diagram of Figure 3.2. is a very useful instrument for better perceiving the roll behavior of a road vehicle as well as for understanding the components of the rollover process and the significance of the rollover threshold.

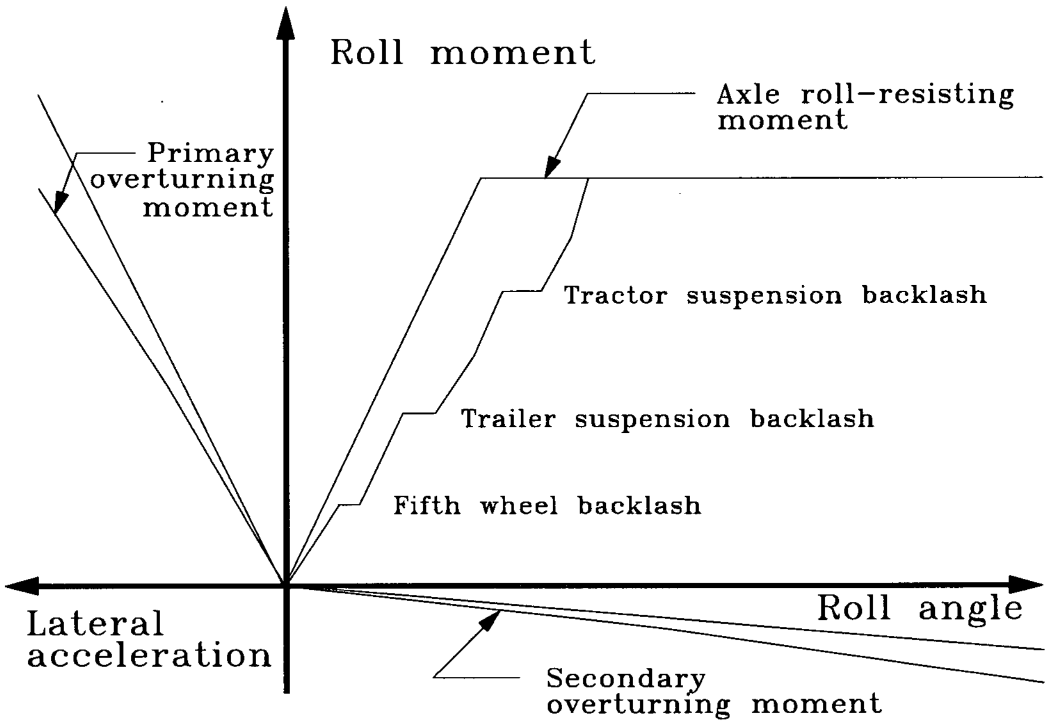



**Figure 3.3.** Roll behavior diagram featuring nonlinearities for an one-axle model of a heavy truck

## 3.3. A three-axle model

### 3.3.1. Presentation and assumptions

The one-axle model is not sufficient for describing the roll behavior of real multi-axle vehicles because it does not take into account differences regarding axle loads, stiffnesses, etc. Therefore a model used for analyzing the roll behavior of a tractor-trailer combination should consider the vehicle as having at least three axles (two for the tractor and one for the trailer) with different stiffnesses and loads, namely the axle of the trailer is the stiffest and carries the highest load and the front axle of the tractor is the softest and carries the lowest load. Although variations due to particular design and load conditions from these assumptions might exist, for real heavy commercial vehicles the above described configuration is the most common.

As the proposed model is meant to describe only the effects on the roll behavior of the differences between the axles of such a vehicle, other refinements such as frame torsional compliance, fifth wheel backlash and compliance, etc., are neglected. The assumption that their influence does not affect the model significantly is supported by other authors' opinions (Preston-Thomas and Woodrooffe 1990). Obviously these assumptions imply an approximation of reality but this is a common approach found in the available literature (as shown in Chapter 2). Taking into account the torsional compliance of a heavy truck (tractor and trailer) frame would be a challenging problem by itself, beyond the scope of the present work. Therefore the tractor-trailer combination is considered as torsionally rigid which implies that the roll angle is constant along the vehicle.

### 3.3.2. Roll behavior diagram for the three-axle model

As in the lumped-masses model, a roll response diagram is utilized to analyze the qualitative roll response of the vehicle. As the vehicle now has several bodies, equation (3.15) expressing the moment balance will now be in terms of summations over all the axles of the vehicle:

$$\sum M_a = \sum M_r - \sum M_g \tag{3.16}$$

The diagram for the whole vehicle is presented in Figure 3.4. As the lateral acceleration and the roll angle are the same, respectively, for the whole vehicle, both primary and secondary overturning moments can be represented, each of them, by single lines, as in the case of an one-axle vehicle. However, because of the difference in loading conditions and stiffnesses between axles, the roll-resisting moments are different. Consequently the net roll-resisting moment, obtained by summing all the roll-resisting moments and subtracting the secondary overturning moment, has several inflection points where changes in its slope occur.

The first inflection point is *A3* and it corresponds to the interior wheel of the trailer axle lifting-off. After this event, the net roll-resisting moment continues to increase with roll angle (but at a lower rate since only the tractor axles resist roll) therefore the vehicle combination is still stable despite the trailer wheel lift-off. The next inflection point, *A2*, corresponds to the tractor rear internal wheel lift-off. Usually, from this point the slope of the net roll-resisting moment becomes negative as the secondary overturning moment grows more rapidly with roll angle than the only roll-resisting moment left- that of the tractor front axle. The last inflection point, *A1*, corresponds to the roll angle at which the tractor front axle, usually much softer than the other two, lifts off. The slope of the net roll resisting moment becomes even lower and remains constant beyond this point.

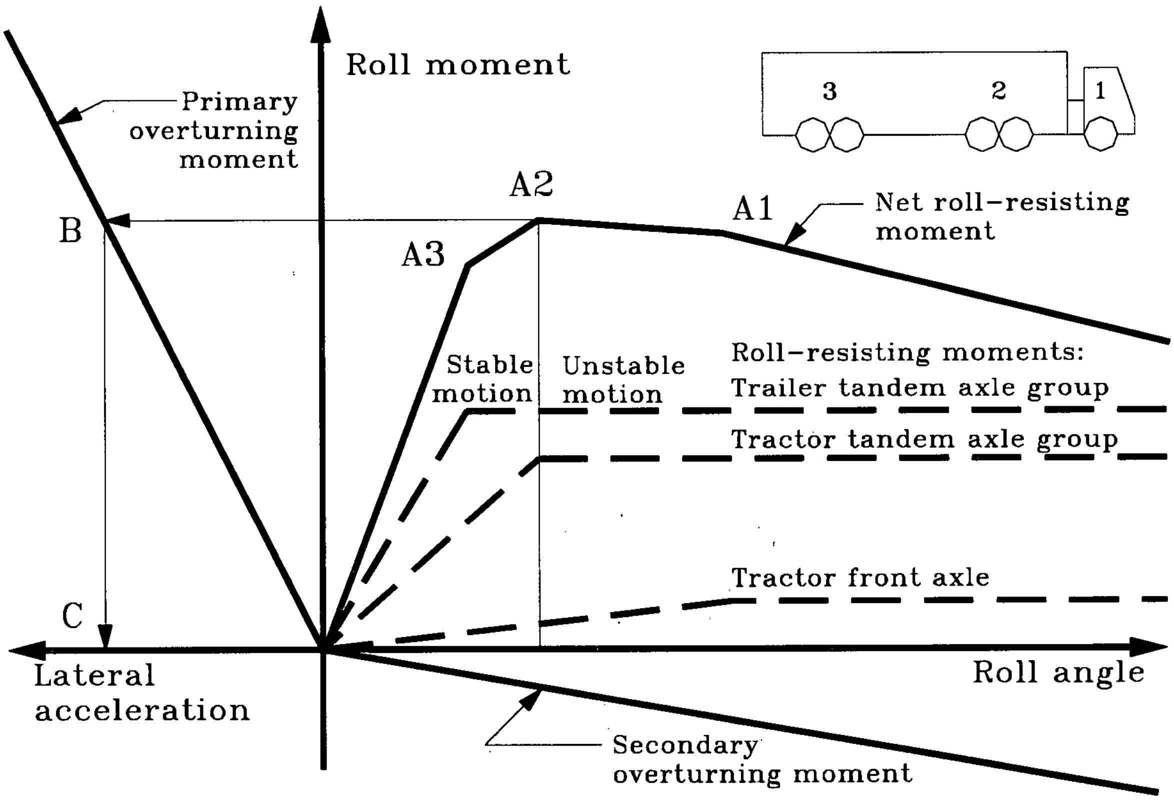



**Figure 3.4.** Roll behavior diagram for a three-axle model of a heavy truck

At this angle all the wheels near the interior of the curve no longer touch the ground and rollover is unavoidable. However, as the diagram shows, rollover is likely to happen earlier. The maximum net roll-resisting moment is produced at point *A2* which marks the boundary between the stable region (characterized by positive slope of the net roll-resisting moment) and unstable region (with negative slope because the destabilizing moment increases faster than the stabilizing moment).

Therefore in the case of a real vehicle, all conditions for rolling over are assured at point *A2*. Although the spacing of the points *A1*, *A2* and *A3* might differ form one vehicle to another depending on their design and loading conditions, the situation described above is very common for most heavy commercial vehicles. The inclusion of suspension backlash would change the shape of the roll-resisting moment but would not change the final observation.

Therefore one can conclude that the rollover threshold for these vehicles can be defined as the value of the lateral acceleration at which the interior wheel of the tractor rear axle lifts off.

## 3.4. Observations and conclusions

Although simple, the above model and diagrams provide qualitative information about the roll behavior of a real commercial vehicle. According to the above discussion, one can state that the rollover threshold is the lateral acceleration at which the tractor rear internal wheel lifts off. However, it must be stressed here that the rollover threshold is not a precise value but varies with the values of the parameters which influence the process:

- vehicle (tire sizes, wear, pressure; suspension characteristics modifications during service time);
- payload (magnitude, density, position);

• environment (road slope and curvature, winds).

Tamny (1993), for instance, insists on including the effect of the tires in the vehicle tilt process as the tire compliance has greater effect than the suspension compliance.

However it is thought that the variation of the rollover threshold (with any of the above parameters) is not too large as the accident statistics available in the literature did not prove a strong influence. Furthermore, any rollover warning device must warn the driver well before the rollover threshold in order to give the human operator enough time to react. Therefore the relative lack of accuracy in assessing the rollover threshold of a real heavy truck does not affect the philosophy of a rollover warning device.

# Chapter 4

# ROLL SENSING METHODS

## 4.1. Basic philosophy

According to several authors, as seen in Chapter 2, sensing the roll behavior of heavy trucks appears to be a logical step towards reducing the incidence of rollover accidents. Sensing the roll behavior might be done by an appropriate rollover warning device. This device should not only have the ability to warn the driver reliably and early enough about the progression towards rollover of the vehicle but it must also have some other characteristics in order to perform its functions, as shown in Chapter 2.

A number of approaches are possible for sensing an imminent rollover. Sensing approaches based on measured lateral acceleration and evaluation of lateral load transfer ratio will be discussed. The reasons behind the approach being taken within this work will then be presented.

## 4.2. Principles of work

### 4.2.1. Measurement of lateral acceleration

As lateral acceleration is the essential cause of rollover, and as the rollover threshold is usually defined in terms of lateral acceleration (or fractions of gravitational acceleration), it appears that measuring the lateral acceleration would be sufficient to determine how close a vehicle is to rolling over. As relative inexpensive accelerometers are very common, a rollover warning device based on lateral acceleration would seem an easy task.

However, as shown in the literature (Preston-Thomas and Woodrooffe 1990, *Mechanics of Heavy Duty Trucks and Truck Combinations* 1985) the rollover threshold is a lateral acceleration for some specific loading condition (weight and position of the center of gravity). It is also influenced by other parameters such as axle roll stiffness of springs and tires. Therefore a rollover warning device based on lateral acceleration measurement should also take into account these various parameters which are not easily measured. Accurate determination of the position of the center of gravity, for example, is very difficult even in static conditions (Winkler et al. 1992) and the center of gravity height has large variations in heavy trucks (Fancher et al. 1986).

However, a rollover warning device based on lateral acceleration measurement would be an attractive solution for the small category of vehicles which usually have only a few loading conditions. For example, with petroleum tank trucks it would be possible to determine previously the position of the center of gravity for different levels of liquid in the tanks. Then, provided a system for on-road liquid level measurement exists on the vehicle, calculation of the position of the center of gravity as a correction factor for the accelerometer output would be relatively easy and the rollover warning device based on the above information might be quite accurate. Special care should, however, be given to liquid slosh at part-load which could significantly contribute to rollover even though its influence could not be detected by the accelerometer.

Another disadvantage is that, for maximum accuracy of the rollover warning device, the accelerometer should be mounted on the trailer. This makes necessary the signal transmission between trailer and truck. Furthermore, if the accelerometer is mounted on the trailer chassis, its reading could be seriously affected by the roll angle since a component of gravity vector acts along the accelerometer axis at higher roll angles (i.e. when approaching the rollover threshold and where accuracy should be the highest). Preston-Thomas and Woodrooffe (1990) estimated that the reading error with the accelerometer mounted on the trailer frame could be around 25-30% at higher roll angles. The solution would be to mount the accelerometer on the rearmost axle which gives three times higher accuracy but exposes the accelerometer to vibrations, salt and moisture. In either case, as the accelerometer readings would be higher than the actual magnitudes of the acceleration, the error would be conservative such as the warning would be too early. This is not necessarily an advantage as the truck operator might finally neglect false warnings, which are not actually related to the state of the vehicle, and that might result in rejection of the device.

### 4.2.2. Evaluation of lateral load transfer ratio

Another approach for a rollover warning device is to consider the lateral load transfer ratio (*LTR*) defined by Erivn and Guy (1986) referred by Preston-Thomas and Woodrooffe (1990) as:

$$LTR = \frac{F_o - F_i}{F_o + F_i} \tag{4.1}$$

This formula comes from the observation that the progression towards rollover means the lateral transfer of vertical load from the tires on the inside of the turn to those on the outside (as shown in Chapter 3). Thus the one axle model in Figure 3.1. becomes unstable and rolls over when the inside tire load $F_i$ becomes zero and the outside tire load becomes $W_s + W_u$. The *LTR* ratio can be expressed in terms of the roll angle by utilizing the expressions of the tire loads and body roll angle of Chapter 3 as well as the obvious relationship between loads:

$$F_o + F_i = W_s + W_u \tag{4.2}$$

$$LTR = \frac{2k_t T\phi_u}{W_s + W_u} \tag{4.3}$$

As seen in Chapter 3, the roll angle $\phi_u$ is proportional to the lateral acceleration $a_y$ and, therefore, *LTR* itself is proportional to the lateral acceleration in the quasi-static case. The value of *LTR* increases from 0 when the lateral acceleration is zero and the loads on the two tires are equal, to 1 in the situation when all the load of the internal tire has been transferred on the external tire ($F_i$=0, $F_s$=$W_s$+$W_u$). This actually corresponds to the maximum value of the lateral acceleration which can be withstood by the vehicle without rolling over- i.e. the rollover threshold. For higher values of lateral acceleration the vehicle crosses the border between the stable and unstable domains (Figure 3.2) and rollover is imminent. Thus *LTR* is an useful non-dimensional parameter able to provide a continuous indication about the roll status of any vehicle, irrespective of vehicle design parameters or specific loading conditions.

For real vehicles (which are multi-axle) *LTR* becomes an addition of the corresponding coefficients of the axles:

$$LTR_{vehicle} = \frac{1}{N}\sum \frac{F_o - F_i}{F_o + F_i} \tag{4.4}$$

where *N* represents the number of axles and has been introduced in order to maintain the coefficient within the same range of values: 0 to 1. As seen in Chapter 3 and Figure 3.4, the *LTR* for the trailer tandem-axle group becomes 1 at point *A3*, then that of the tractor tandem-axle group at point *A2* and, finally, that of the tractor front axle at point *A1*. The *LTR* for the whole vehicle starts from zero when no lateral acceleration is present but, unlike the case of the one-axle model, has a value smaller than 1 at the rollover threshold. This happens because *LTR* is 1 at point *A1* when all the tires from the interior of the vehicle lift off whereas rollover is likely to occur around point *A2* (tractor rear axle lift-off). The specific *LTR* threshold value for a certain vehicle will therefore depend on its design characteristics and loading particulars.

Although there is no consensus in the literature (Ervin and Guy 1986, Fancher et al. 1986, Preston-Thomas and Woodrooffe 1990) concerning how many axles of a heavy truck should be monitored and how the warning value should be chosen, the *LTR* warning value for detecting the incipient rollover (in order to give the driver enough time to react), is an attractive method. However its practical implementation seems neither easy nor cheap.

The calculation of *LTR* for one axle requires evaluation of the tire loading while the vehicle is in motion which is a difficult task. Alternate methods are proposed by Preston-Thomas and Woodrooffe (1990) by estimating these loads from measurements made on the vehicle. There are several approaches along this direction. One of them consists of inserting a device with a small strain-gauged sensing disk into a hole machined in a vertical portion of the frame above the suspension (Barnett and West 1983). Changes in tire loading cause changes in frame stresses, which deforms the disk and provides an electrical output. Another approach is to measure the pressure of the fluid for a pneumatic or hydraulic suspension of the payload (Preston-Thomas and Woodrooffe 1990).

Both the above solutions are not quite satisfactory due to accuracy problems (especially when auxiliary roll stiffness devices such as anti-roll bars are also present) as well as serious reliability problems. Despite providing good estimation for tire loads, the same durability problems plus very high costs are characteristics of another approach consisting of sandwiching transducers between the wheel hubs and wheels to measure wheel loads while the vehicle is in motion.

Preston-Thomas and Woodrooffe (1990) considered, as the most attractive practical solution for estimating the *LTR*, the utilization of commercial on-board weigh scale systems which consist of various types of force transducers developed by Philips and other companies. The most successful systems of on-board weighing have used electronic load cells mounted on each side of the trailer axle and at the fifth wheel. Preston-Thomas and Woodrooffe (1990) considered that the rear-most axle is mandatory to be monitored for rollover sensing because it is the most affected by rearward amplification and it is the least affected by the tractor stabilizing action. Sending signals to the cab unit could be achieved relatively easy by telemetry, as they stated.

As the current costs of commercial weigh scale systems have been rather high - between \$4000 and \$6000 (according to Philips 1989)- and as it is expected that the cost of the load measurement system will be in the same range, this seems to preclude the development of an independent rollover warning device based on *LTR*. A solution for reducing the cost proposed by Preston-Thomas and Woodrooffe (1990) is the modification of current on-board weigh scale systems to perform both functions: weighing and rollover warning. However, the above study is not specific either about the technical problems or about the extra-cost to implement the proposed modification.

Although it seems appropriate for the target of monitoring the roll behavior of heavy trucks, the solution of utilizing electronic load cells currently used for on-board weight measurement has, however, several disadvantages. First of all it involves instrumentation of the trailer what is not very desirable. All trailers to be used with a roll sensor-equipped tractor would need to have weigh scales installed. Then instrumenting the trailer might cause some tuning problems as the same truck might combine with different trailers (longer or shorter for instance) and thus the roll behavior of the combinations might be different from one situation to another. A better solution would be to also instrument at least the rear axle of the tractor but this would dramatically increase cost and complexity.

Another important disadvantage is the cost. This matter has two aspects- cost per device which is expected to be high or relatively high (as shown by Preston-Thomas and Woodroffe 1990) and cost per fleet of vehicles which was neglected in the above study. As is well known, the number of trailers is much larger than the number of tractors and therefore it would be considerably more useful to instrument tractors that are almost always on road than trailers which are immobilized for loading and unloading. Furthermore, instrumenting tractors seems a reasonable approach because, as showed in Chapter 3, rollover of heavy tractor-trailer combinations is closely associated with the tractor rear axle lift-off.

### 4.2.3. Principles of existing commercial devices

As far as it is known, some commercial devices able to perform rollover warning or to be the base for such a system already exist.

One such a device was the "STABE ALERT Stability Monitoring and Alarm System" developed by Roadway Safety Systems Inc., of Lakeland, Florida. This was meant to be used for tractor-trailers, tandem combinations and straight trucks of more than 26 feet. The system used inductive front-wheel speed sensors similar to those used for anti-lock braking systems as well as strain gauged-disk load sensors fitted precisely into holes machined in the frame of the trailer. The system was intended to warn about several types of events such as defective front-wheel bearings, flat tires and broken leaf springs in addition to roll instability. Roll motion was believed to be sensed by detecting the uneven distribution of metal strain (and, therefore, loads). As apparently the company is no longer operational, there is no information available about the performance of the system.

Another system, not available yet but believed by its author to be feasible and acceptable by the trucking industry, is the McLaughlin truck/trailer control system (1993). This seems to be a complex system able to perform several important functions: automatic steering and braking control of the trailer wheels, near object detection, overhead clearance detection, pulse width braking control, battery recharging, etc. Last but not least is the trailer tilt monitoring in four directions. The tilt sensing would use a transformer located at the rear top of the trailer and the result would be to not only monitor roll but also trailer braking and steering assistance. The author affirms that the system might be accepted by the trucking industry and it might pay for itself in about 3 to 5 years. No details are, however, provided in support of the above judgments.

Other systems are on-board weigh scale systems, that Preston-Thomas and Woodrooffe (1990) think to be appropriate for roll sensing. One system based on estimating wheel loads has been developed by International Road Dynamics of Saskatoon, Saskatchewan. The system is able to measure individual wheel loads (even in a tandem) by measuring the shear stress in the portion of each axle between the suspension attachment point and the brake. This system appeared to ensure a better indication of wheel and axle loads provided its accuracy and, especially, reliability are good enough.

More conventional weigh scale systems having transducers placed between the frame and the suspension are currently being produced by several manufacturers. Preston-Thomas and Woodrooffe (1990) believe that these systems, although less accurate than the above one, have a potential for integration with a rollover warning device based on the lateral transfer ratio (*LTR*).

### 4.2.4. Principle of a tilt sensor system

Rollover of a tractor-trailer combination is assured when the inside wheels of all the axles of the combination have lifted-off the road. A rollover warning device must therefore predict whether all of the inside wheels will lift-off during the current maneuver, or at least whether the inside wheels of the drive axle will lift-off. The wheels of the drive axle are critical because that axle's wheels generally lift-off after the trailer axle's inside wheels lift-off and because the steering axle plays an insignificant role in preventing rollover due to its limited roll stiffness. Drive axle lift-off depends on the load on the drive axle, its roll stiffness and the roll angle of the frame of the tractor. Of these, only the roll angle of the tractor frame is the real question. Future values of this angle depend on the current value of the roll angle, the current value of the roll velocity and the current and future values of the roll acceleration, at least if the combination is assumed to have a single degree of freedom in roll. This assumption eliminates from further consideration trucks hauling hanging meat or partially-full tanks.

We will return to the measurement of the current roll angle and velocity presently. Current and future values of lateral acceleration depend on the rotary inertia of the vehicle and on the roll moment applied to the vehicle. The roll moment, as was stated in the previous chapter, can be considered as the sum of the primary and secondary overturning moments and the roll resisting moment. The roll resisting moment depends on the specifics of the tractor and trailer suspensions, which can be determined, at least approximately, in advance. The secondary overturning moment depends on the height and weight of the vehicle, with payload, as well as on the characteristics of the suspension. The most difficult quantity to evaluate is the future value of the primary overturning moment. That quantity depends not only on the weight and height of the vehicle but also on the instantaneous lateral acceleration induced in the vehicle.

To try to develop a suite of sensors and an algorithm to predict the instantaneous lateral acceleration even a few seconds ahead would be a formidable task. Rather, the thrust of the current developments will be to retain the driver as the heart of the rollover prediction system and to provide additional information to the driver to aid in an accurate prediction.

Reviewing the steps leading to the prediction of rollover, it would appear that the most basic information which the driver requires are the current lateral acceleration and roll angle and roll velocity of the vehicle. That information is not easily sensed directly by the driver as the driver has become better isolated from vibrations in the truck. Additionally, an indication of the height of the center of gravity of the truck would be useful as the rollover threshold is directly linked to that height and the height of the center of gravity is difficult to sense.

The roll angle sprung mass of the vehicle is, if we neglect torsional flexibility and backlash, a constant throughout the length of the truck. Therefore, the roll angle can be measured anywhere along the truck's length. From operational considerations, sensors which are mounted solely within the tractor are preferable to those which are split between the tractor and the trailer. A second issue is to consider in the measurement of roll angle is whether to measure the absolute roll angle of the truck or the roll angle relative to the road. Again, it is much easier to measure a roll angle relative to the road than to measure an absolute roll angle. The disadvantage of measuring the relative roll angle, that road roughness will add noise into the measurement, can be alleviated by filtering of the signal.

The principle of the system would be to sense the lateral tilt of the tractor body subjected to both its roll movement and that transmitted through the fifth wheel from the trailer. A single tilt sensor mounted transversely onto the tractor frame would not be capable of measuring the tilt of the body because of the apparent change in the direction of the gravity vector due to centripetal acceleration. Instead, two tilt sensors would be situated on the longitudinal trailing arms of the truck drive axle(s). As the tractor tilts, one of the tilt sensors would have a reduced angle with respect to the ground, while the other trailing arm would have an increased angle. The difference in tilt angles between the two trailing arms is proportional to the tilt of the tractor body with respect to the road. A third tilt sensor mounted transversely on the frame of the tractor would measure the lateral acceleration experienced by the tractor.

The above arrangement is believed to be appropriate for most heavy trucks which feature rigid axles and air spring suspension with longitudinal trailing arms for transmitting longitudinal loads (due to driving and braking) between the vehicle body and the drive axle. The arrangement would be completed by an electronic circuitry able to compare the signals received from the three tilt sensors to some limit pre-set values and, thus, detect the roll state of the whole heavy truck and warn, when appropriate, the driver. Furthermore, the tilt of the tractor frame can be presented to the driver as a lateral load transfer ratio (*LTR*) for the drive axle(s). The *LTR* is proportional to the frame tilt. A value of *LTR* of 1 implies rollover.

As the tilt sensors suitable for the above mission are already available and are not very costly, as the wiring is of the tractor only and the mounting is not complicated, it seems that this system would be less costly than other proposed systems. Therefore roll sensing based on tilt sensors is believed to be a solution which deserves further attention and investigations.

# Chapter 5

# PRESENTATION AND TESTING OF A TILT SENSOR

## 5.1. General presentation. Principle of work

As shown in the previous chapter, one possible solution for estimating the roll position of a heavy truck could be a combination of tilt sensors. One commercially-available tilt sensor is basically a small lightweight box (5 x 6.5 x 3.5 cms) which can detect quite accurately its angular position with respect to gravity along a direction denoted as the tilt direction. The tilt sensor outputs a pulse-width modulated signal, with pulse "on" time proportional to the tilt of the sensor. This tilt sensor is manufactured by TVI Corporation of San Jose, California.

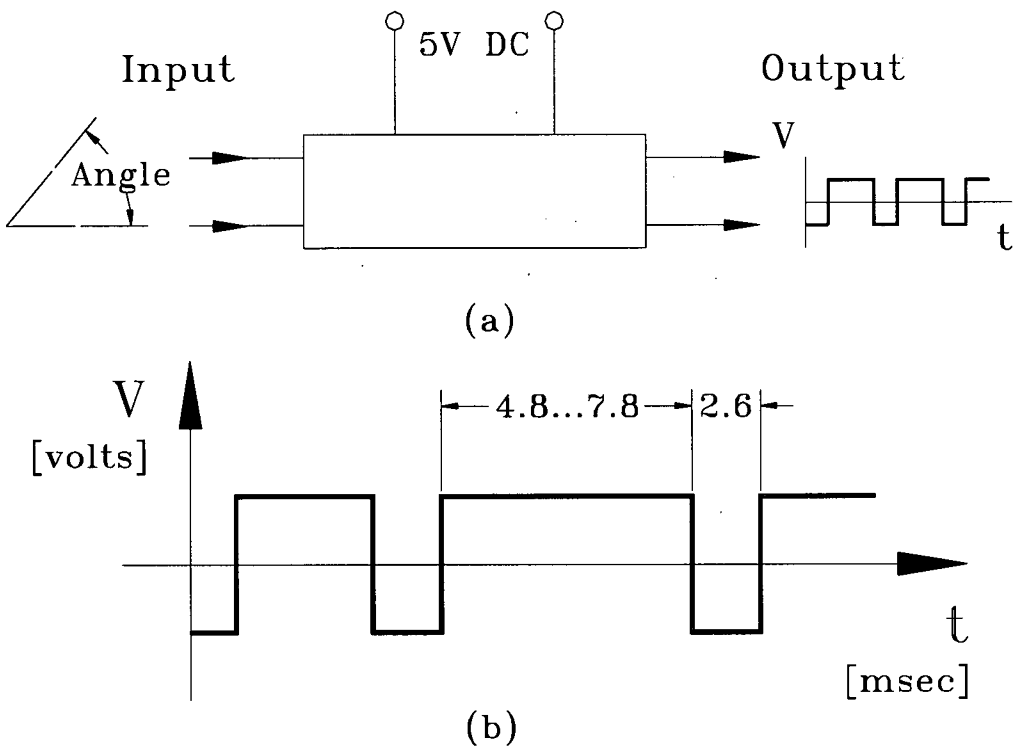



**Figure 5.1.** Tilt sensor:

**(a)** input and output;

**(b)** output signal range

More specifically, the "negative" (lower) part of the signal has the constant length of 2.6 msec while the "positive" (upper) part has a length between 4.8 and 7.8 msec for tilt angles between -90$^{0}$ and +90$^{0}$ with a value of 6.3 msec corresponding to the flat table (horizontal) position. The device is powered by 5V DC and it works on an optical principle- the light beam emitted by a source towards two light receivers is scattered by a small bubble of air whose position changes when the box is tilted in the active direction.

According to all the above characteristics, it was thought that the tilt sensor might be an appropriate solution for detecting and monitoring the roll position of a tractor-trailer combination under on-road conditions. In order to assess the suitability of the tilt sensor for the purpose three preliminary kinds of tests were performed: static tests, dynamic tests and on-road tests. Static and dynamic tests were performed in the period October-November 1995 at the Department of Mechanical Engineering of the University of British Columbia. On-road tests utilized a Toyota Previa minivan which features a rear axle configuration similar to a truck (rigid axle and trailing arms). The tests were performed in January 1996 on roads in the UBC area.

## 5.2. Measurement chain

The measurement chain utilized for tilt sensing of vehicles consisted of the following instruments:

- *Tilt sensor* -Is denoted as "AutoSense sensor" by its manufacturer- "TV Interactive Corporation" of USA. This sensor is able to detect the angle of its base plate with respect to gravity. The output is a square wave with the frequency dependent on the tilt angle.

- *Data acquisition system* - Manufactured by "Strawberry Tree Incorporated" and used for converting the analog data into samples of digital data.

- *Toshiba T3200 laptop computer* -The final link of the data acquisition chain.
- *9V battery and converter 9V to 5V* -Used for powering the tilt sensor with 5V DC.

During the tests, additional equipment was also used:

- *Dividing head* -Used for accurately measuring the actual tilt angle and cross tilt angle of the tilt sensor during static tests.
- *Bruel&Kjoer shaker* -Used for shaking the tilt sensor both horizontally and vertically during lab dynamic tests.
- *Bruel&Kjoer sine-random generator* -Used for generating the exciting signal for the above shaker.
- *Bruel&Kjoer power amplifier* -Used for amplifying the signal of the above generator.
- *Nicolet dual channel FFT analyzer* -Used for measuring the frequency of the signal fed into the shaker.
- *Bruel&Kjoer accelerometer* -Used for measuring the acceleration of the tilt sensor.

| | |
|---|---|
| •*Bruel&Kjoer charge amplifier* | -Used for amplifying the signal of the above accelerometer. |
| •*Oscilloscope* | -Used for verifying the measurements against the data acquisition system in the early stages of the experiments. Also used to display the acceleration signal from the accelerometer during lab dynamic tests. |
| •*RP Electronic Components power inverter* | -Used for powering the Toshiba computer from the car battery. |
| •*Toyota Previa minivan* | -Used for on-road tests. |

## 5.3. Static tests

### 5.3.1. General description

All static tests were performed by using a dividing head in order to provide the necessary accuracy for the actual angle. The tilt sensor was clamped in the dividing head and this latter was put on a horizontal flat table. Various angles in both the tilt plane and the cross tilt plane were generated according to the aim of the respective tests.

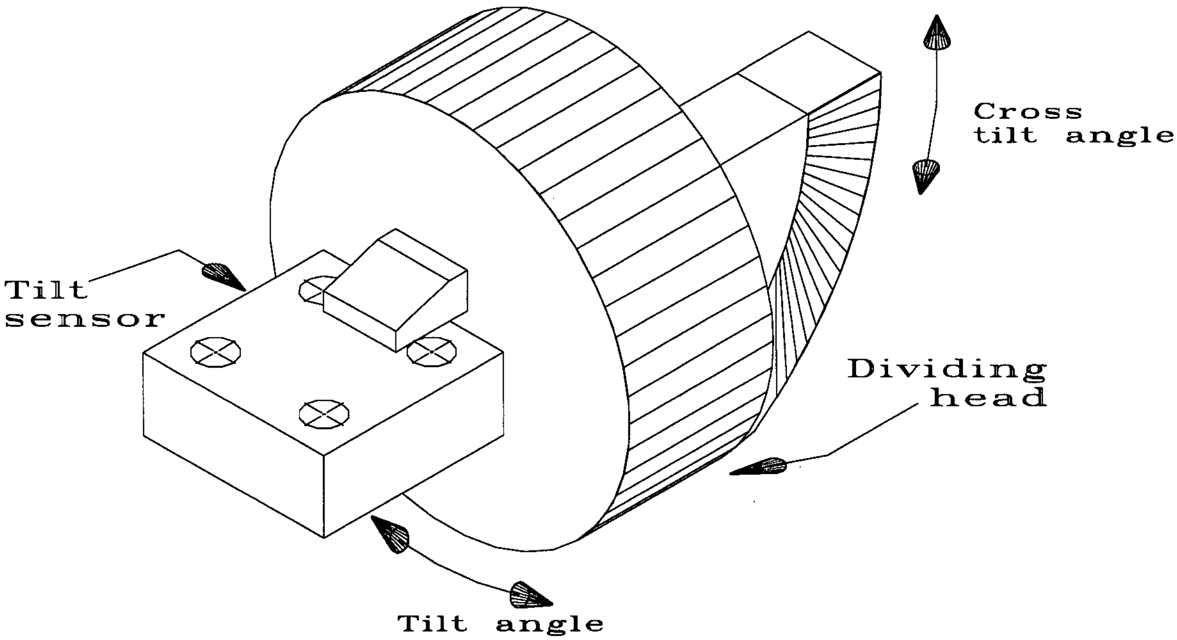



**Figure 5.2.** Static testing of the tilt sensor with a dividing head

## 5.3.2. Tests and results

Each measurement was done by recording a file of 500 samples containing directly measured angles. Then the data was processed with statistical tools available in Mathcad 4.0 and Excel 5.0.

The static tests had as their main targets:

**(a)** investigation of the behavior of the tilt sensor over its entire measuring domain ($-90^{0}$ to $+90^{0}$) as well as of possible nonlinearities;

**(b)** investigation of possible hysteresis properties;

**(c)** check of the influence of various cross tilt angles on the measured tilt angles.

**(a) Behavior of the tilt sensor within the measuring domain ($-90^0$ to $+90^0$)**

The measurement strategy consisted of tilting the sensor in steps of $1^0$ in the measuring domain between $-30^0$ and $+ 30^0$ and with progressively increasing steps in the rest of the domain. The results of the test are shown in Appendix B-1 and Figure 5.3.

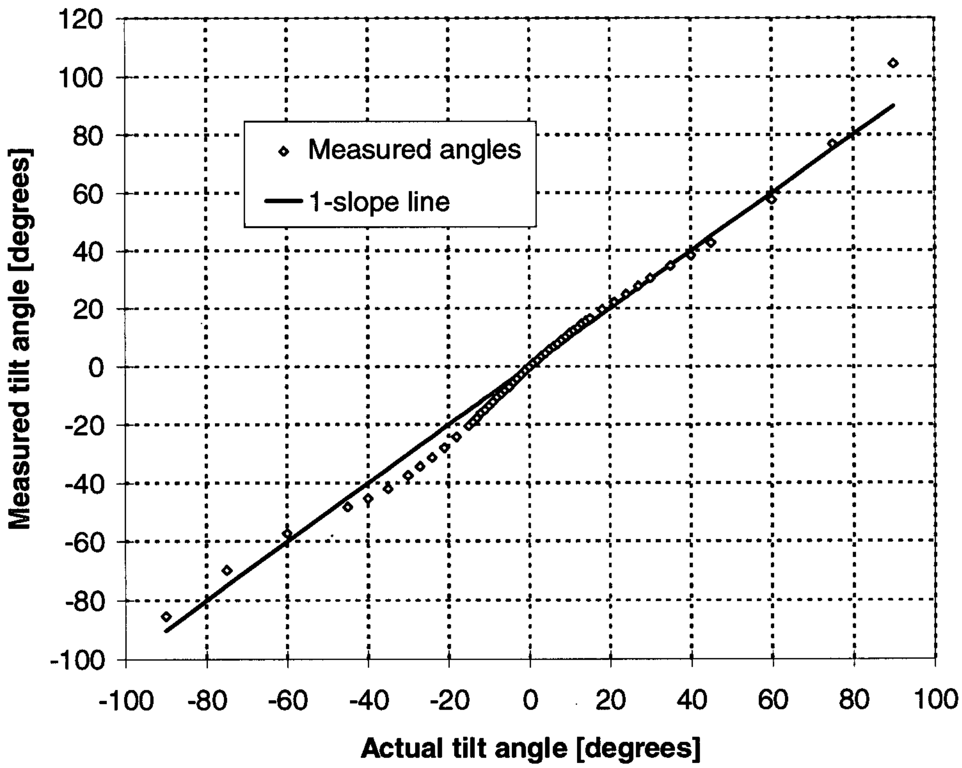



**Figure 5.3.** Measured angle and 1-slope line versus actual tilt angle [degrees]

One can notice that in the usual range of measurements ($-30^0$ to $+30^0$) the tested tilt sensor displayed quite good linearity and fairly low spread of the output. Advancing towards the limits of the domain ($-90^{o}$ or $+90^{o}$) showed some nonlinearity as well as the increasing spread of the results.

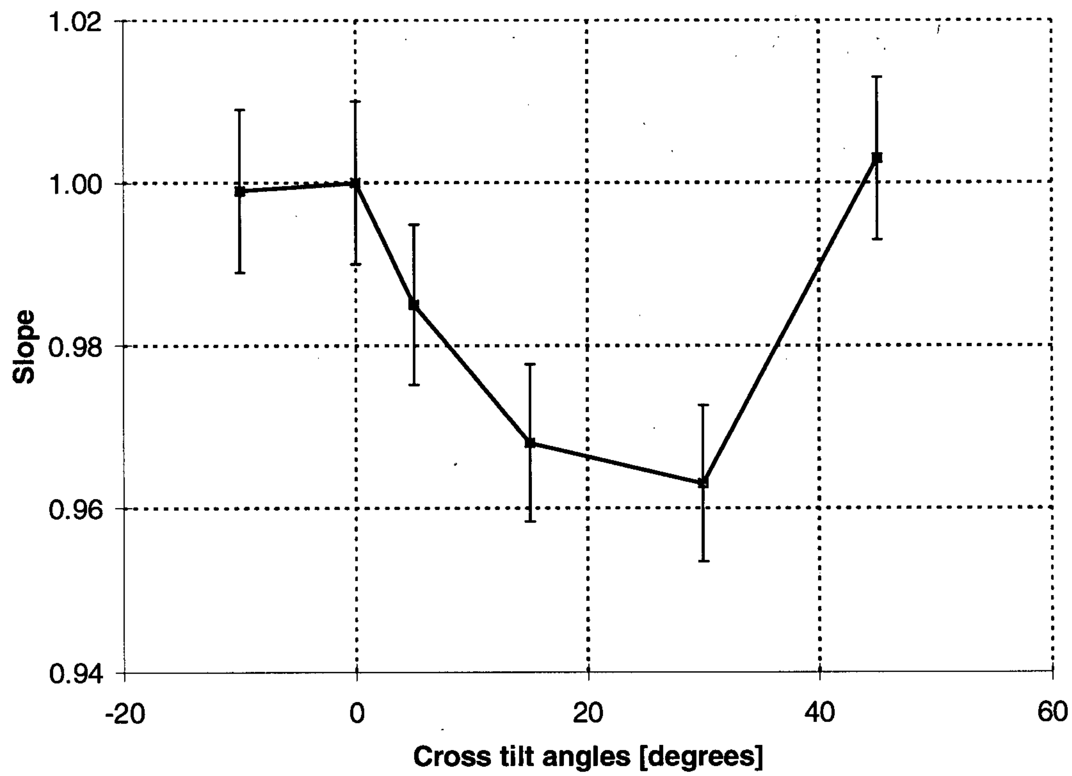
**(b) Investigation of hysteresis properties**

The measurement strategy consisted of performing measurements in two directions: increasing tilt angle ("loading") and decreasing tilt angle ("unloading") respectively. These measurements were done with both zero cross tilt angle and at various cross tilt angles ($-10^{o}$ to $+45^{o}$). Results are shown in Appendices B-2 to B-7. The conclusion was that, for the experiments performed, the hysteresis effect has not been significant as absolute values measured differed, in most cases, by an angle of less than $0.5^{o}$ for the usual range of measurements ($-30^{o}$ to $+30^{o}$).

**(c) Influence of cross tilt angles**

In order to see the nature and magnitude of the influence generated by cross tilt angles, the dividing head was tilted at several cross angles: $-10^{o}$, $5^{o}$, $15^{o}$, $30^{o}$ and $45^{o}$ respectively. For each of the above cross angles, the sensor was then tilted in the active direction in a range of angles between $-30^{o}$ and $+30^{o}$ as this situation is the most likely to be encountered in practical applications. Results are shown in Appendices B-3 to B-7 as well.

For each set of recorded data for a certain cross tilt angle, a line which best fits the data was determined by its slope and intercept. Then all these slopes and intercepts, together with those corresponding to zero cross tilt angle, were plotted on the same graph, also showing the error bars corresponding to 1% error for slope and to 20% error for intercept, as shown below:



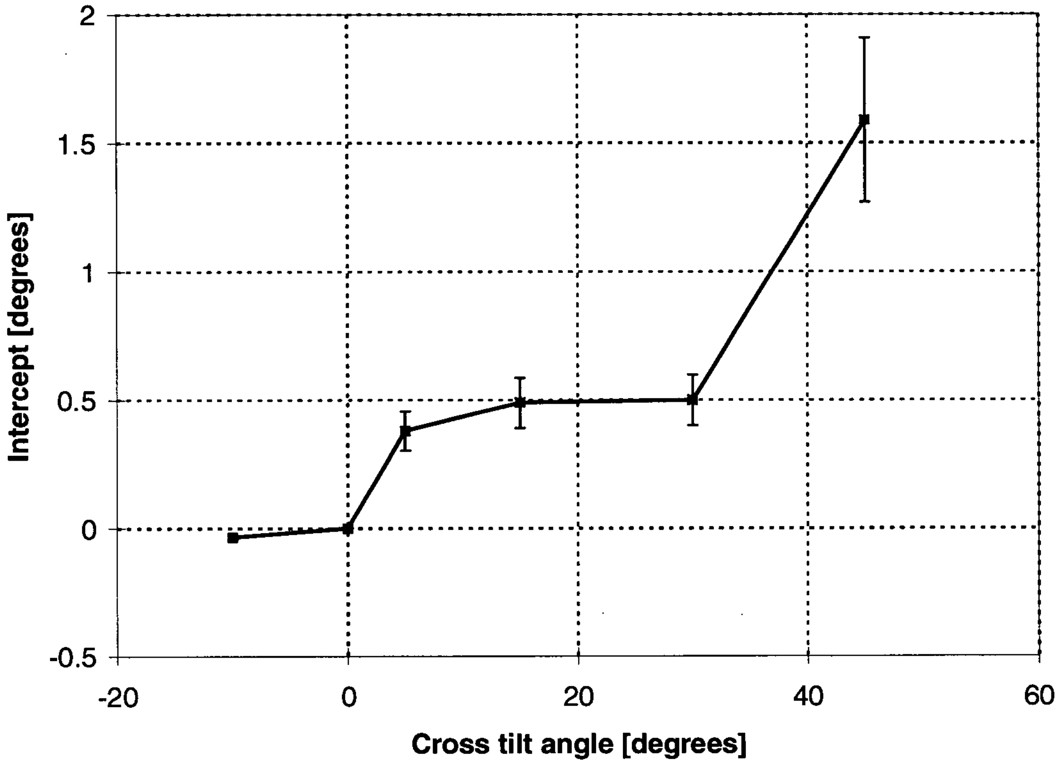



**Figure 5.4.** Slopes and intercepts of lines best fitting the data versus cross tilt angles [degrees]

From the above plots one can see that, although an influence on the measured tilt angle caused by the cross tilt angle does exist, it is quite small for a reasonable range of measurements. For the maximum slope deviation which appeared to occur, for the tested tilt sensor, at a $30^{0}$ cross tilt angle, the error due to this angle was below 2.5%.

## 5.4. Lab dynamic tests

### 5.4.1. General description

All dynamic tests were performed using a shaker controlled by a wave generator which could provide a range of frequencies and amplitudes. A measurement chain for the acceleration measurements was also utilized. The tilt sensor was clamped on the head of the shaker in the horizontal position (no tilt or cross tilt angles).

Two types of measurements were done:

**(a)** horizontal shaking;

**(b)** vertical shaking.

Each measurement was taken by recording a file of 1000 samples containing directly measured angles. This data was processed with statistical tools available in Mathcad 4.0 and Matlab. The dynamic tests over the frequency domain 10-50 hertz were the main target for the investigation of the behavior of the tilt sensor.

### 5.4.2. Tests and results

**(a) Horizontal shaking**

The tilt sensor was subjected to horizontal shaking in an approximate frequency range from 2 to 50 Hz.

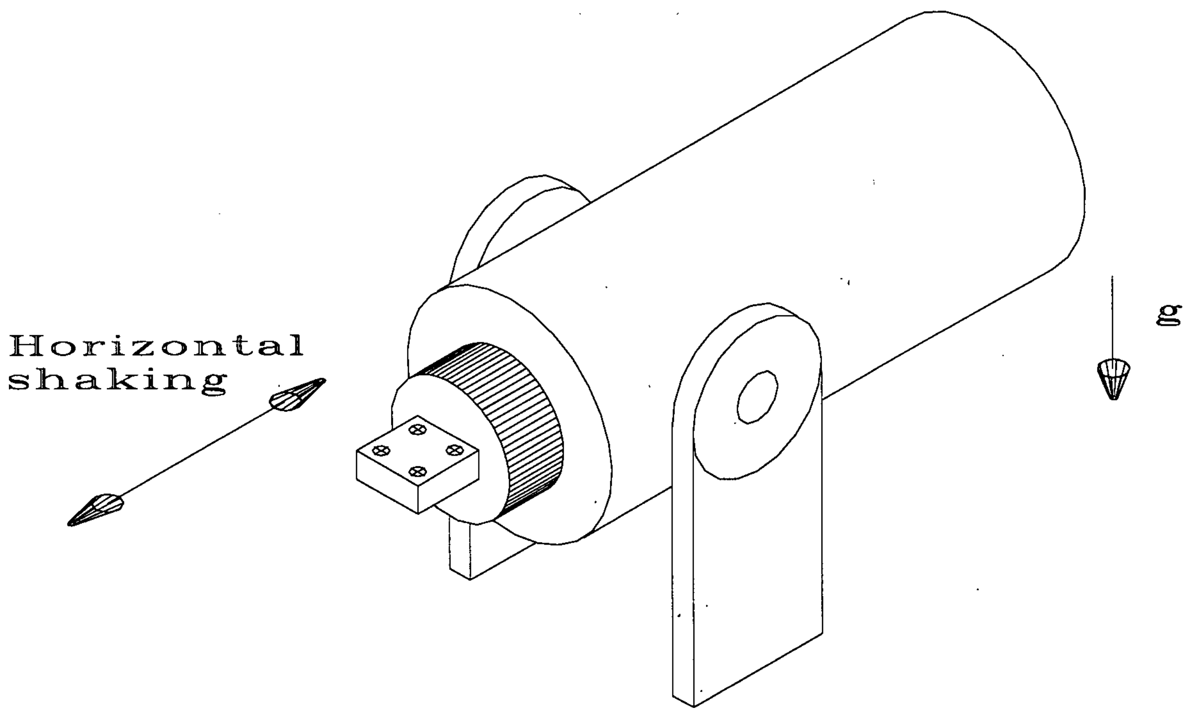



**Figure 5.5.** Lab dynamic tests-horizontal shaking

At each frequency step the measured angle as well as the horizontal acceleration was recorded. Then the standard deviation of the measured angle as well as the standard deviation of the actual angle (formed by the horizontal acceleration and the gravity acceleration) were calculated. Finally the ratio of "*standard deviation measured angle / standard deviation actual angle*" was plotted versus the driving frequency. Standard deviations ratio was thought to be a better measure of tilt sensor accuracy than the ratio of the average values of the above angles as the former was taking into account the spread of the results. As a resonance peak was noticed in the approximate range of 22-26 Hz, in subsequent tests the data was filtered with a lowpass filter acting either at 5 or at 10 Hz. The results are displayed in the plot below. A table containing the numerical results can be also found in Appendix C-1.

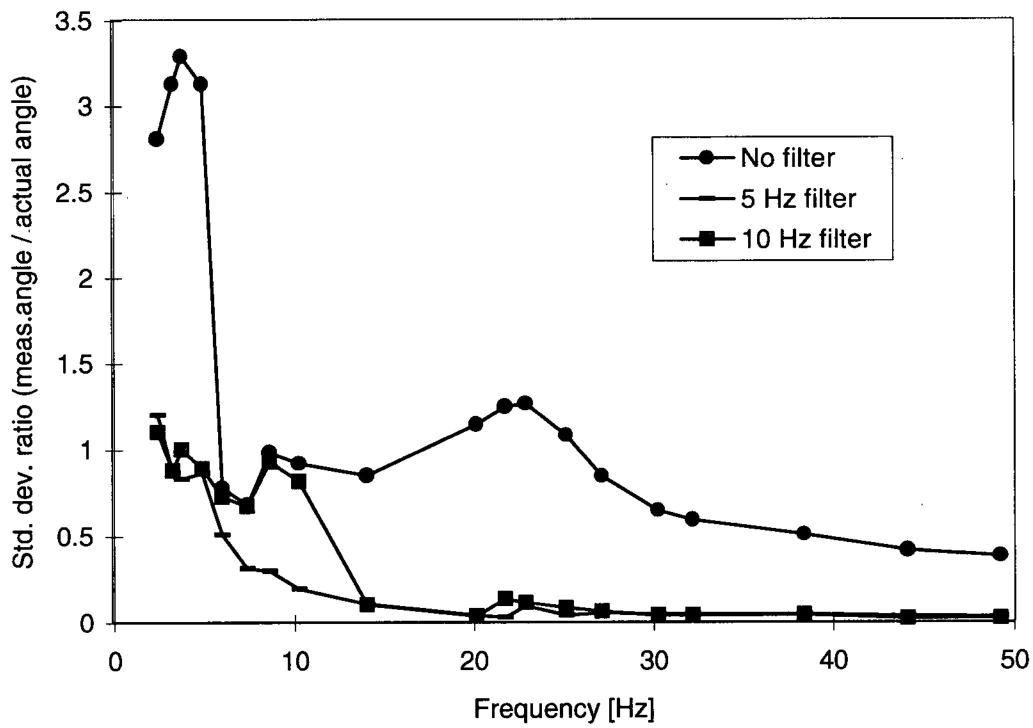



**Figure 5.6.** Ratio of standard deviation of measured angle and actual angle versus frequency of horizontal shaking [Hz]

As one can see, filtering has eliminated the resonant peak at 23 Hz. More significantly, the ratios of standard deviations below 10 Hz has been reduced from up to 3, down to 1. This indicates that there was some excitation of the resonant mode of the sensor even during low frequency excitation.

**(b) Vertical shaking**

The tilt sensor was subjected to vertical shaking in an approximate frequency range from 2 to 80 Hz.

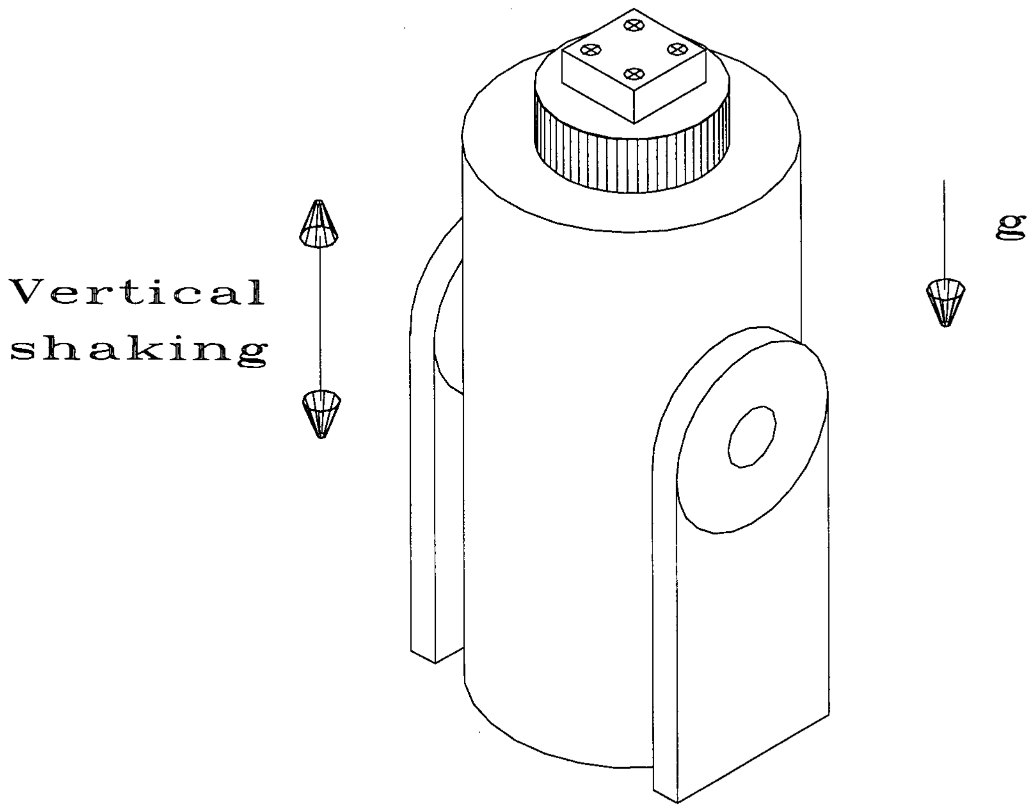



**Figure 5.7.** Lab dynamic tests-vertical shaking

At each frequency step the measured angle as well as the vertical acceleration were recorded. The ratio *standard deviation measured angle / standard deviation measured angle with no motion* has been plotted versus the driving frequency. This measure was meant to express the dynamic behavior of the tilt sensor when moving strictly vertically, with various frequencies but at constant (approximately $0^{O}$) tilt angle.

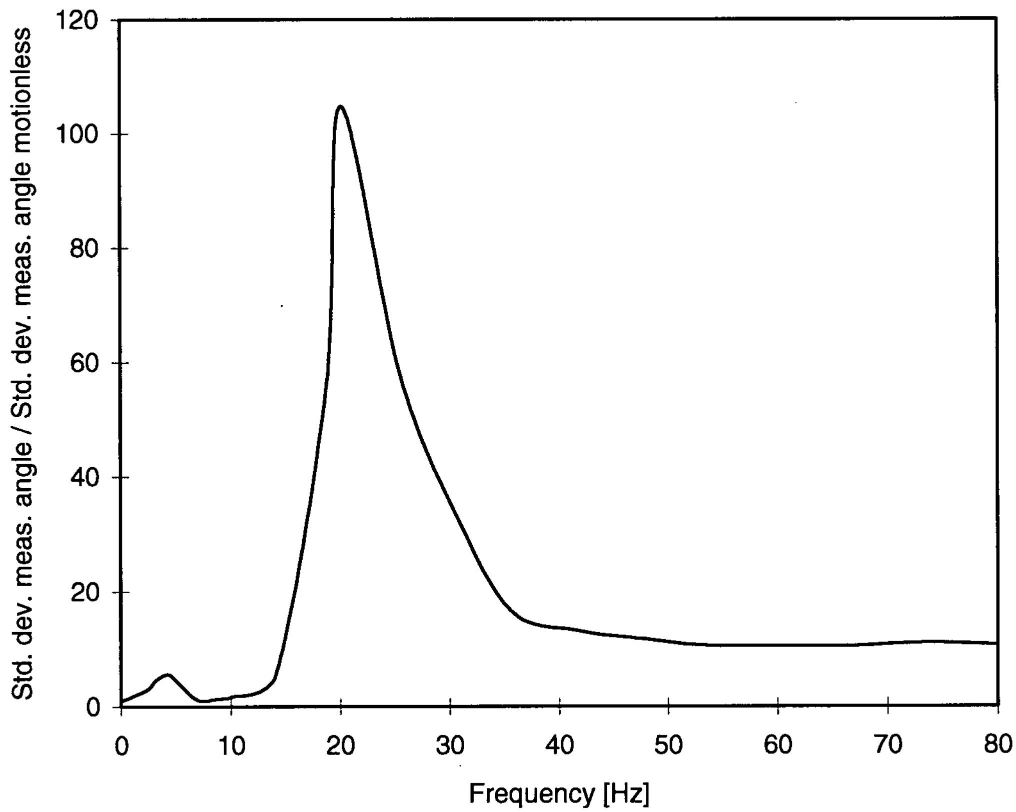


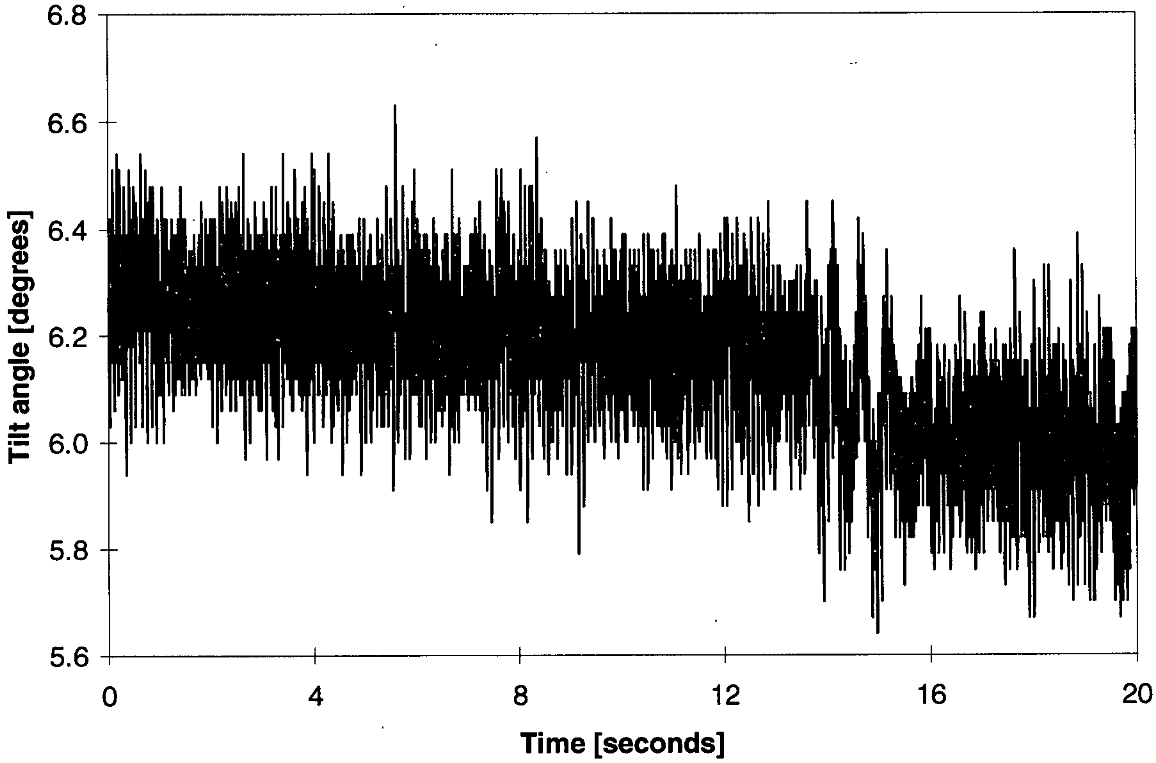

**Figure 5.8.** Standard deviation measured angle/ standard deviation measured angle motionless versus frequency of vertical shaking [Hz]

As can be seen, the tilt sensor features two resonance peaks- the smaller one at about 4-5 Hz and the larger one at about 20-22 Hz. The latter frequency corresponds to the resonant peak of the sensor, while the former may indicate some non-linear behavior, as was noted for the horizontal shaking test. According to the tests done and the results obtained, the tilt sensor seems to be an appropriate instrument for measuring angles either in static or in dynamic conditions for a reasonable domain of practical applications. Filtering of the sensor output to eliminate the resonant peak at 23 Hz is essential.

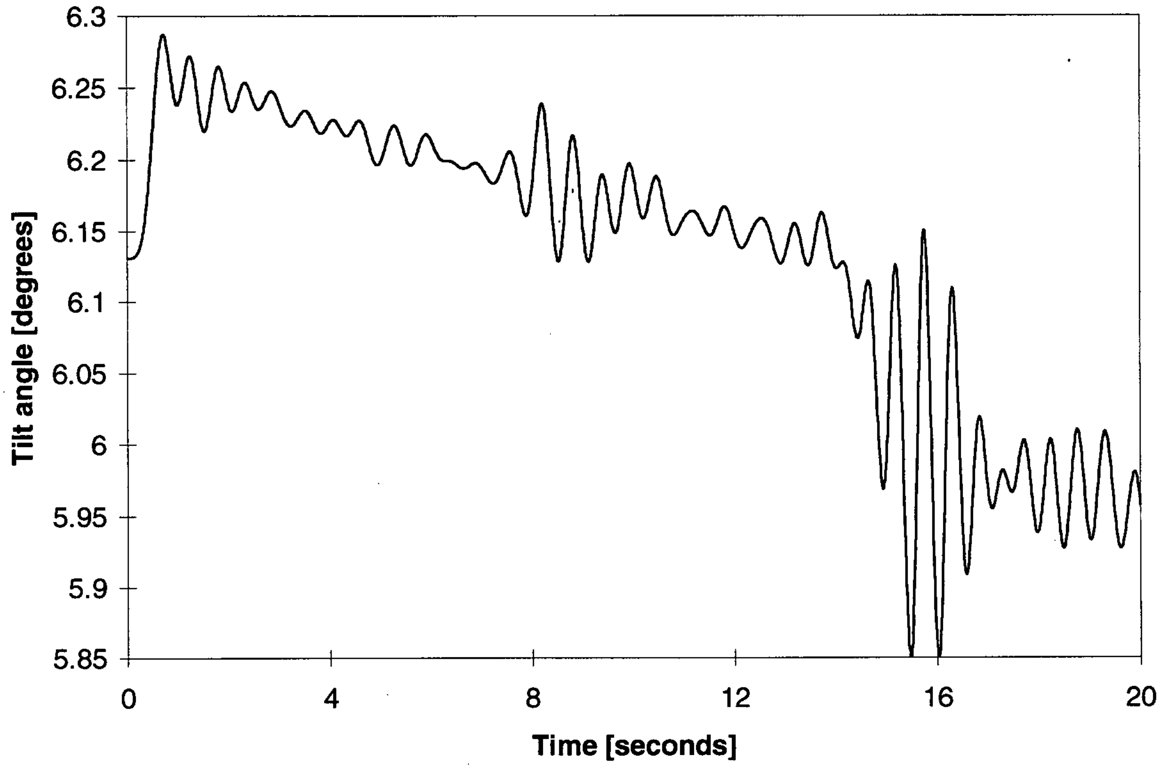
## 5.5. On-road tests

These tests were performed by installing the whole measurement chain on a vehicle. A new element was added to the measuring equipment on this occasion-namely a power inverter allowing powering of the computer from the car battery. The car utilized was a Toyota Previa minivan and the tests were performed in January 1996 on roads in the UBC area.

First, a single tilt sensor was placed on the car floor in order to investigate its sensitivity to car body motion and to car body vibrations due to the engine. These tests proved that the sensor had good sensitivity to lateral acceleration and fairly low sensitivity to body vibrations due to engine noise as the plots in Figure 5.9 show.


6.8
6.6
6.4
6.2
6.0
5.8
5.6
Tilt angle [degrees]
0
4
8
12
16
20
Time [seconds]

(a)




**(b)**

**Figure 5.9.** On-road noise test. Tilt sensor on car floor, engine idle, car motionless:

**(a)** unfiltered data;

**(b)** data filtered with 2 Hz low-pass filter

The next step consisted of mounting the tilt sensor on a trailing arm of the rear axle of the above specified car. The sensor was taped in such a manner that the tilt plane was parallel to the longitudinal plane of the car, thus being able to detect the motion of the trailing arm while the car was moving on a road.

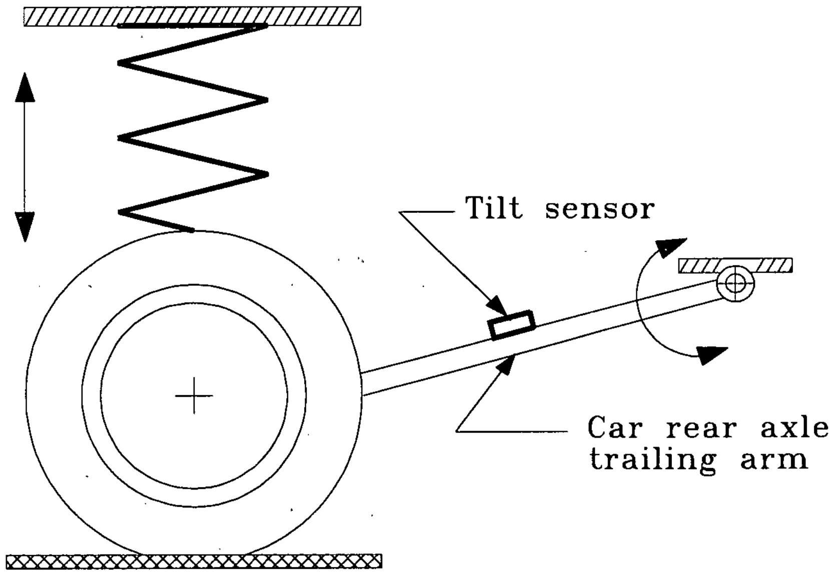



**Figure 5.10.** Tilt sensor placed on a trailing arm of a minivan

All tests consisted of recording files of 2000 samples at a sampling frequency of approximately 100 Hz.

Results, especially after being filtered at frequencies as low as 2 Hz in order to eliminate noise, have proven that the tilt sensor was able to detect changes in the trailing arm position due to the car motion. The values recorded were different between several measurements as the angle of the trailing arm varied with the specific movement of the car (i.e. cornering on a curve with a certain radius, with a certain lateral acceleration) and with the specific road cross-section.

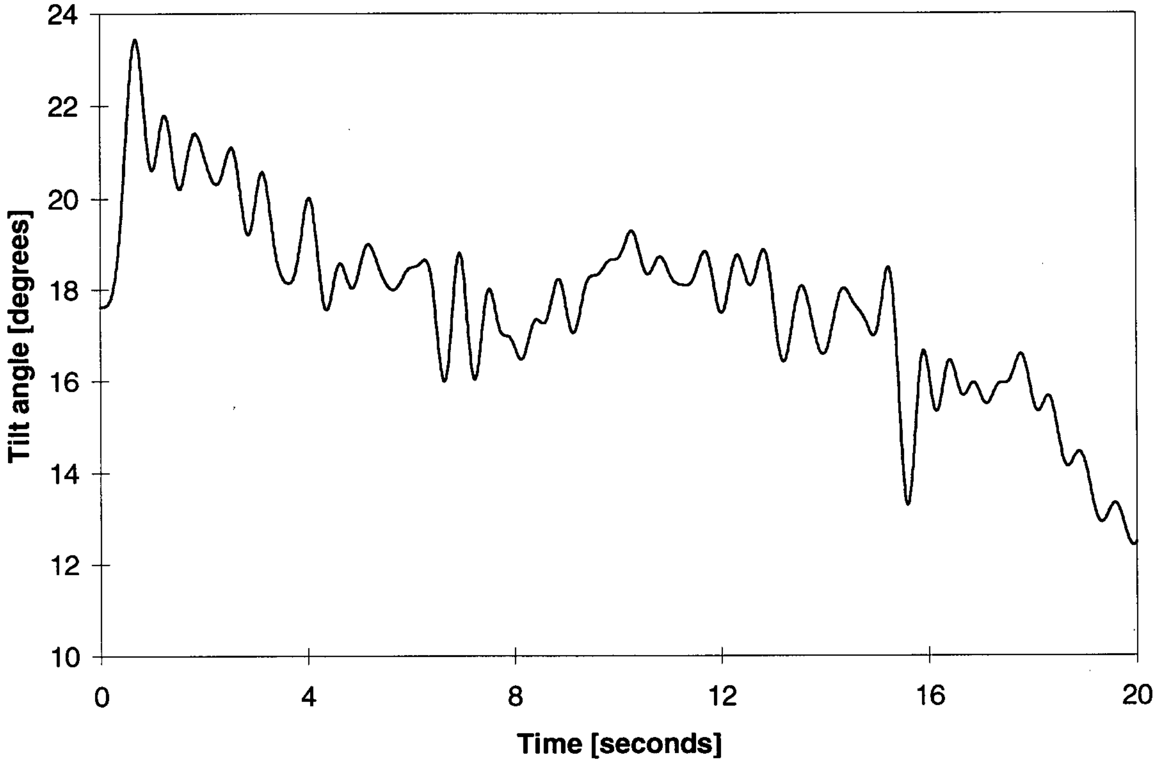



**Figure 5.11.** On-road test. Tilt angle measurement during a left turn. Data filtered with 2 Hz low-pass filter

One can notice that after filtering, the data recorded allows detection of fairly clear tendencies of the tilt angle- increasing or decreasing depending on the actual movement (upwards or downwards, respectively) of the trailing arm. Therefore the plots of Figures 5.11 and 5.12 show that clear signals were obtained after filtering the output of the tilt sensor; no further conclusions could be drawn from the on-road tests with a single tilt sensor.

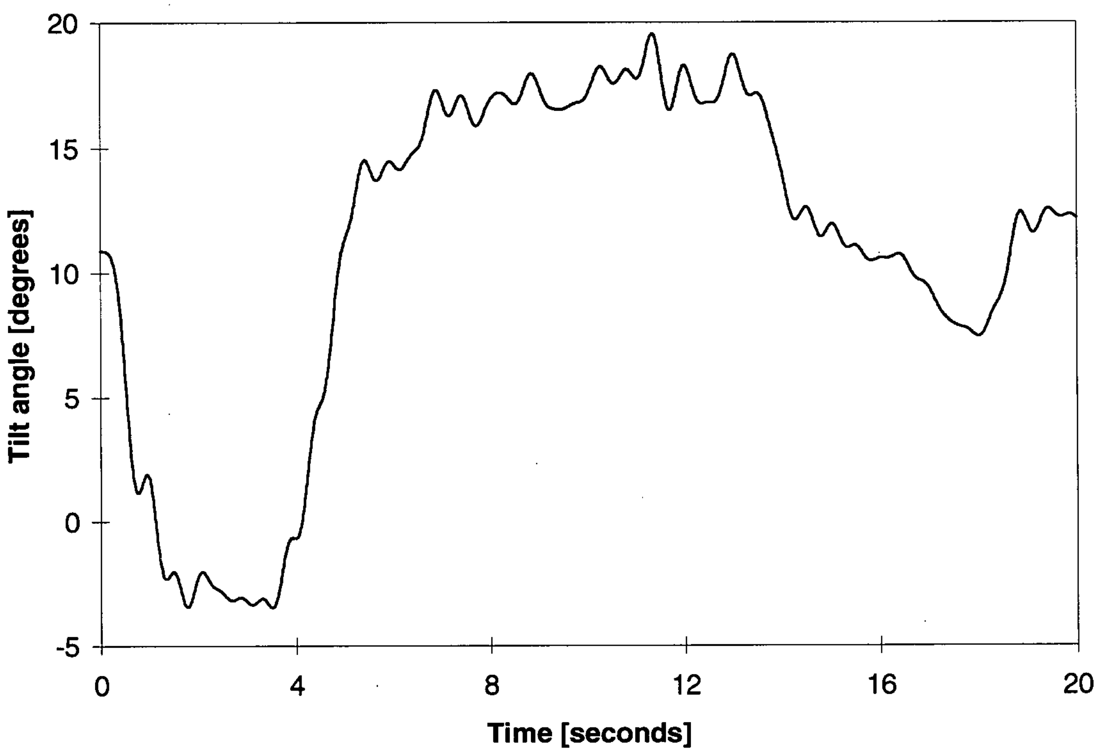



**Figure 5.12.** On-road test. Tilt angle measurement during a right turn. Data filtered with 2 Hz low-pass filter

It must be mentioned that the above plots are based on data recorded in usual traffic conditions. Therefore the "left" and "right" turns were not negotiated under controlled conditions. As a consequence, results can be examined appropriately only in a qualitative manner. As one single tilt sensor was not enough to completely monitor the roll movement of the car, the next logical step was to mount several tilt sensors, in a specific arrangement, on a vehicle in order to assess the viability of the proposed roll sensing approach.

# Chapter 6

# TESTING OF A ROLL SENSING SYSTEM

## 6.1. Principle of sensor arrangement

Following the assessment of different theoretical and practical solutions for roll sensing of heavy trucks, it was found that a system of three tilt sensors mounted in a specific arrangement on a vehicle could be an attractive approach. Such a system would have many advantages compared to other proposed or existing devices. An extensive discussion regarding the above aspects has been developed in Chapter 4 of this work.

For the reasons shown previously, the basic idea has been to monitor the roll state of the tractor only. As shown in Chapter 3 and Chapter 4 of the present work, the behavior of the driving (rear) axle of the tractor is of the highest interest for assessing the roll state of the whole multi-axle vehicle. When this specific axle (which could also be a tandem one) is close to the internal wheel lift-off rollover is very likely to occur.

Most heavy commercial tractors have tandem rigid rear axles supporting the vehicle body by air springs and transmitting the longitudinal forces due to traction and braking through longitudinal trailing arms. When the vehicle is rolling, the axles tilt with respect to the ground and the vehicle body rolls with a supplementary angle with respect to the axles as shown in Chapter 3. The supplementary tilt of the vehicle body causes a differential tilt between the two trailing arms- one upwards and the other one downwards, depending on the roll direction (towards right or left, respectively). It was thought that measuring the tilt angle of the two trailing arms and the lateral acceleration experienced by the vehicle body could provide a good indication of the dynamic roll state of the vehicle.

As a consequence, a system of three sensors was installed on a minivan featuring a truck-like rear axle with a rigid housing and two longitudinal trailing arms. As shown in Figure 6.1, a tilt sensor was mounted on each trailing arm in order to have its measurement direction along the longitudinal axis of the trailing arm. These two tilt sensors were able to measure the tilt angles of the trailing arms with respect to their initial position corresponding to a motionless vehicle. The third tilt sensor was placed on the vehicle body (assumed not to have any torsional compliance) in such a position as to measure the lateral acceleration.

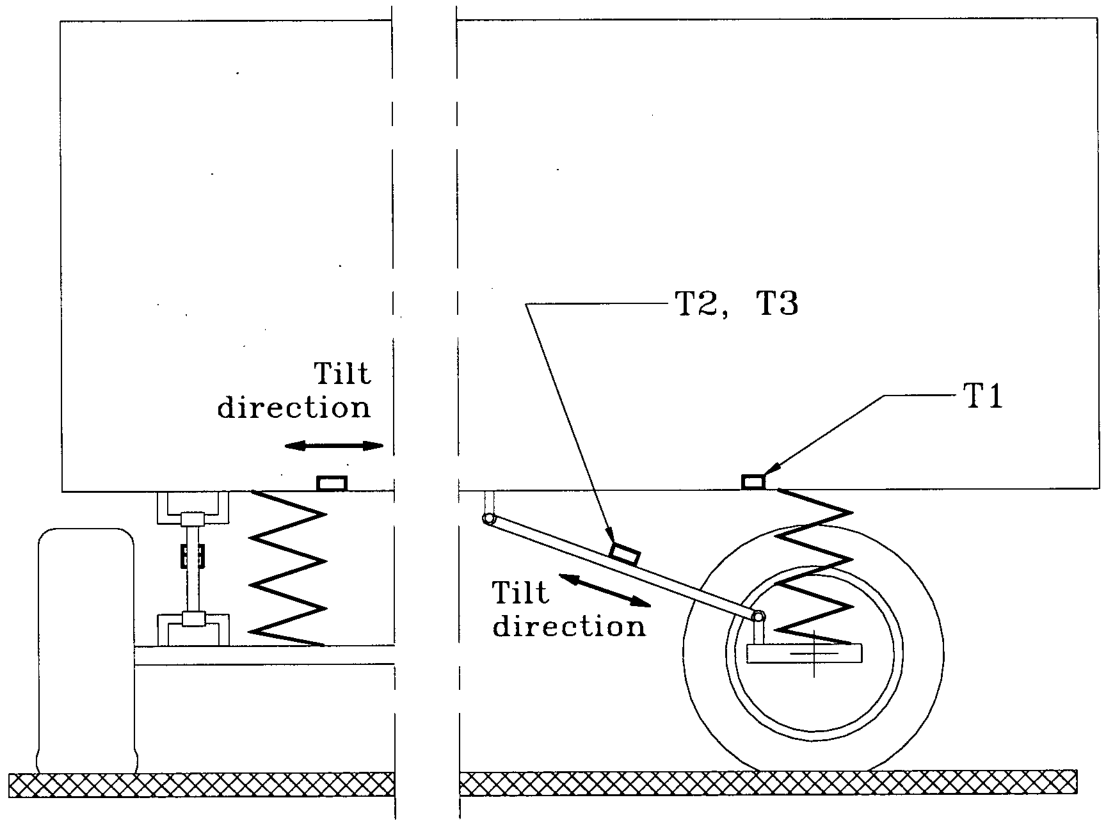



**Figure 6.1.** Tilt sensor arrangement and their measurement directions on a vehicle:

**T1** -tilt sensor mounted on the vehicle body;

**T2, T3** -tilt sensors mounted on the rear axle trailing arms.

The same data acquisition system was used but the software was modified in order to receive simultaneously signals from the three sensors.

## 6.2. Tests and results

Several types of tests were performed on various roads in the UBC area during the spring of 1996. The first type of tests were static ones. Data was recorded with the system mounted on the vehicle, engine running and small impulses being given to the vehicle while it was motionless. These tests were a general check of the system as well as a reference basis for the subsequent ones. The actual dynamic tests were performed on several types of roads in the UBC area: smooth roads while negotiating several $90^0$ curves, smooth sinuous roads and rough but relatively straight roads. These tests exercised the capability of the system to discriminate tilt angles induced by curves from noise caused by road irregularities.

Each dynamic test as well as all static tests involved the recording of 10,000 data samples in separate files. The data files were subsequently analyzed by computer using Matlab software. Data was filtered with a 2 Hz low-pass filter in order to eliminate noise caused by several sources: the tilt sensors themselves, the power supply and, especially, car vibrations induced by road irregularities.

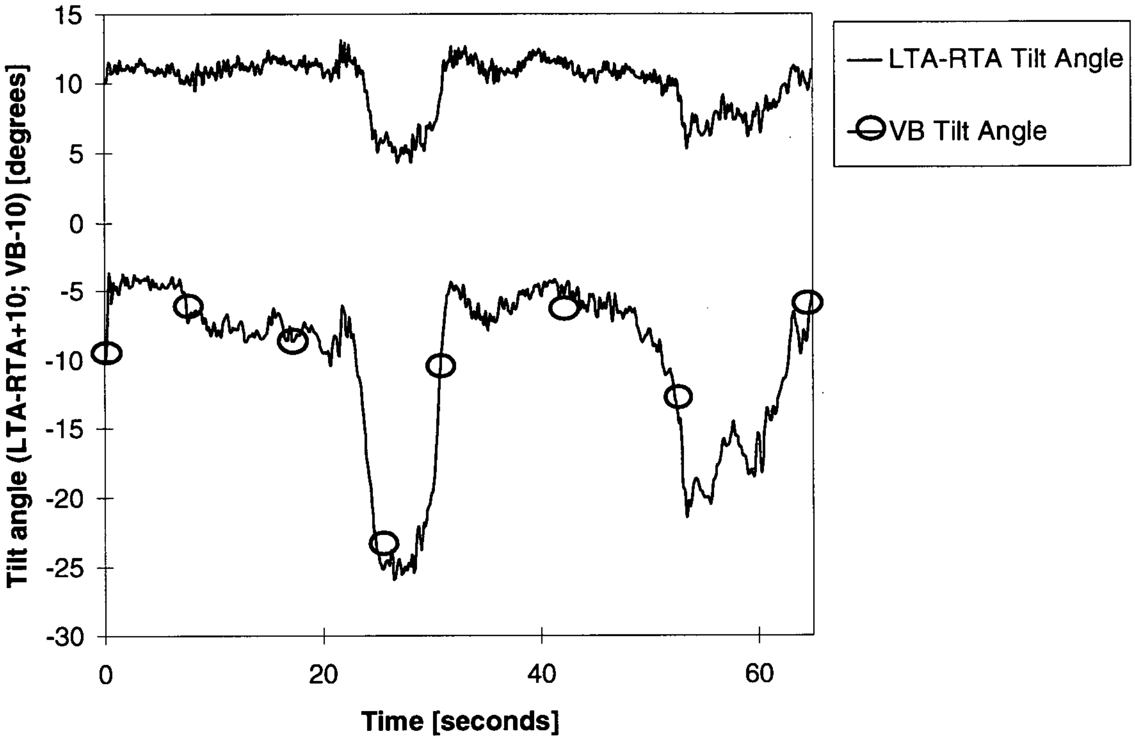
Data analysis was completed by a system calibration which consisted of tilting each tilt sensor, in turn, by means of an accurate dividing head, as described in Chapter 5. Therefore a calibration constant relating the numerical output of each tilt sensor to its actual tilt angle was calculated. The values of these calibration constants are shown in the following table.

**Table 6.1.** Tilt sensors calibration constants

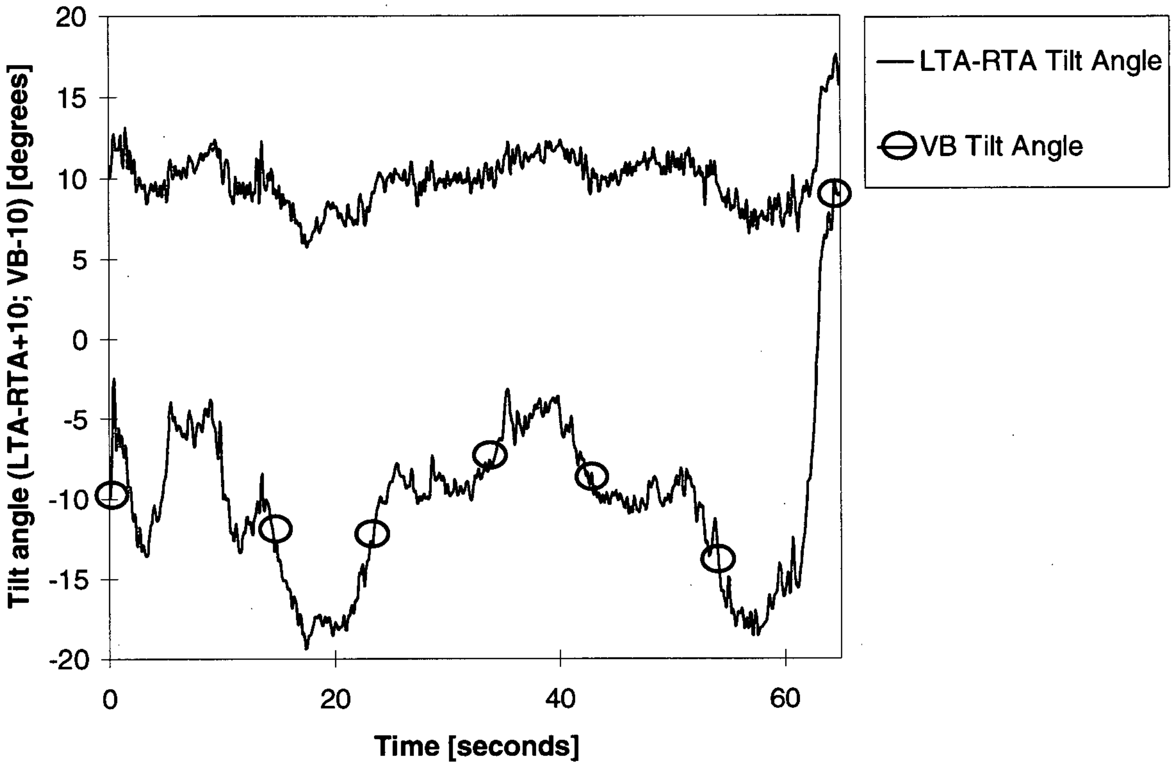
| **Tilt sensor mounted on:** | **Calibration constant [units / degree]** |
|---|---|
| Right trailing arm (RTA) | 19 |
| Left trailing arm (LTA) | 17 |
| Vehicle body (VB) | 24 |

As one can notice, the calibration constants of the three tilt sensors feature a quite significant variation. Since the calibration was done with maximum accuracy, each measurement being performed twice for reliability reasons (with differences of less than 2% between measurements), the explanation of the variation of the calibration constants is that the tilt sensors display some nonuniformities due to manufacturing.

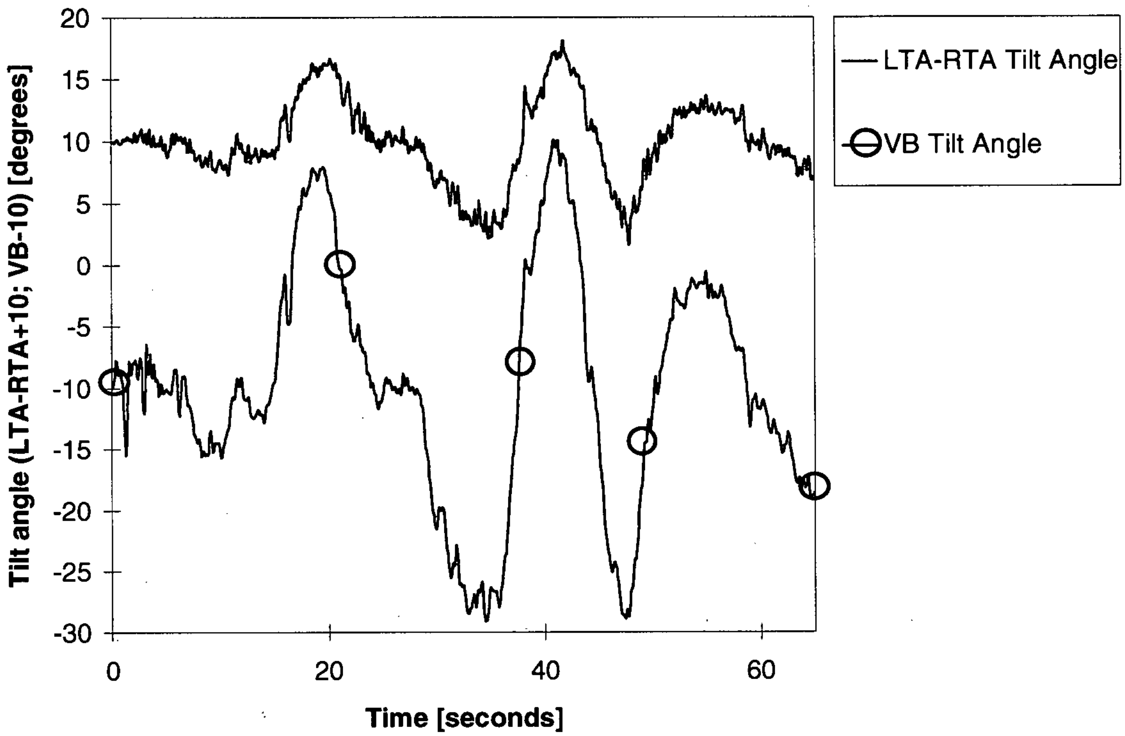
Knowing the calibration constants and having the filtered recorded data, the next step was to analyze whether and to what extent a correlation between the output of the tilt sensors exists and how this correlation could be linked to the roll state of the vehicle. Data from different recordings corresponding to various types of tests have been plotted against time as shown in the figure below.


15
10
5
0
-5
-10
-15
-20
-25
-30
Tilt angle (LTA-RTA+10; VB-10) [degrees]
0
20
40
60
Time [seconds]
LTA-RTA Tilt Angle
VB Tilt Angle

(a)

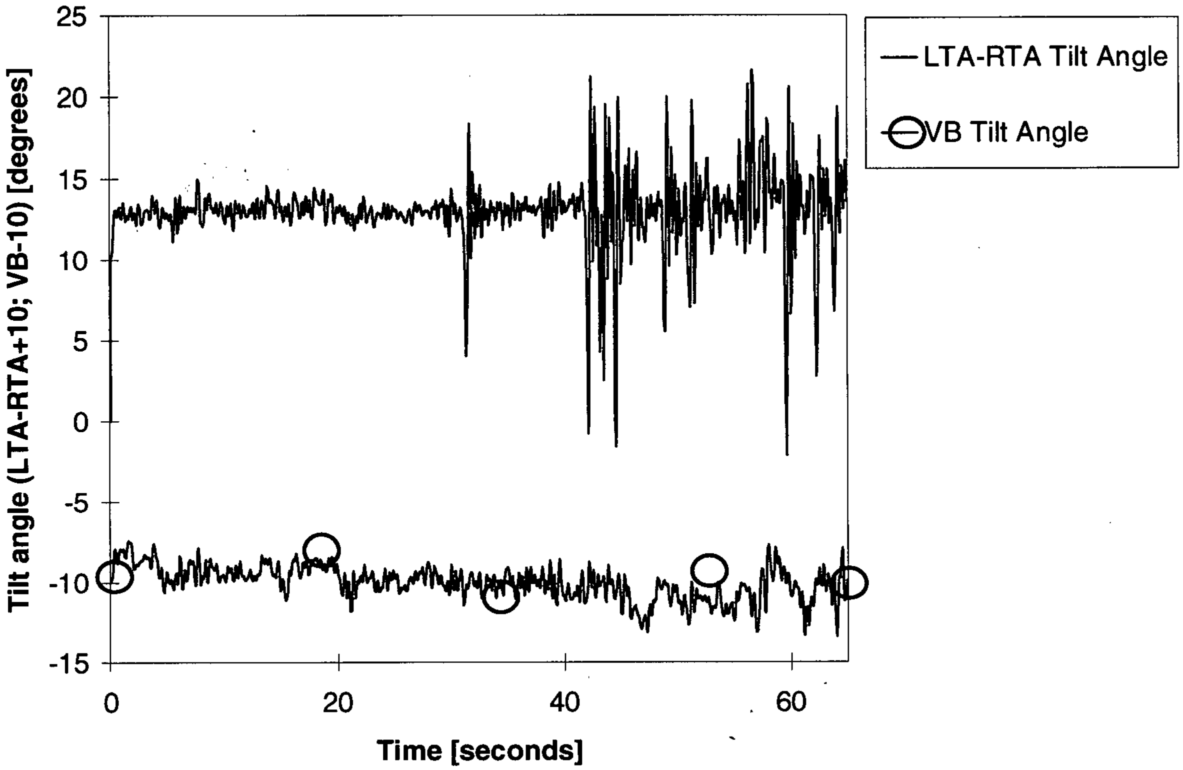

20
15
10
5
0
-5
-10
-15
-20
Tilt angle (LTA-RTA+10; VB-10) [degrees]
0
20
40
60
Time [seconds]
LTA-RTA Tilt Angle
VB Tilt Angle

(b)



Tilt angle (LTA-RTA+10; VB-10) [degrees]
20
15
10
5
0
-5
-10
-15
-20
-25
-30
0
20
40
60
Time [seconds]
LTA-RTA Tilt Angle
VB Tilt Angle


(c)




(d)

**Figure 6.2.** Vehicle roll sensing. Difference of outputs of tilt sensors mounted on rear axle trailing arms (*LTA-RTA*) compared to the output of the tilt sensor mounted on the vehicle body (*VB*) against time on various roads:

**(a)** negotiating two right turns on a smooth road;

**(b)** and **(c)** driving on a relatively smooth and sinuous road;

**(d)** driving on a relatively rough and straight road

When analyzing the above plots as well as those presented in Appendices D1 to D4 of this work, one can notice a fairly obvious correlation between the outputs of the tilt sensors mounted on the trailing arms and that of the tilt sensor mounted on the vehicle body. Namely it can be seen that when the vehicle body tilt sensor (*VB*) has a significant deviation from zero (which implies the presence of a significant lateral acceleration), the difference between the output of the left (*LTA*) and right (*RTA*) tilt sensors has exactly the same tendency (of increasing or decreasing, respectively). Of course the "plus" and "minus" signs of the above correlation depend on the particular mounting direction of the tilt sensors on the vehicle but the nature of the noticed correlation is essentially the same.

Another situation in which there is a noticeably differences in the outputs of the *LTA* and *RTA* tilt sensors might occur when the two wheels of the same axle encounter different road conditions (smooth and rough road, relatively). However, in this case the output of the *VB* tilt sensor is not significant which means that there is no significant lateral acceleration and, therefore, no noticeably roll motion but only a road irregularity. More information to support the above noticed situations is available from the plots of Appendices D1 to D4.

The utilization of two independent sensing ways is mandatory for accuracy. As shown in Chapters 1 to 4, the rollover threshold may be expressed in terms of lateral acceleration for a specific (known) loading condition only. On the other hand, the body roll may be induced by other factors than the lateral acceleration (such as road slope) and cannot properly describe, by itself, the roll state of the vehicle.

In order to better estimate whether there was a relation between the outputs of the three tilt sensors, a correlation coefficient between two sets of data was also calculated. The first set of data was the difference of the outputs of the two sensors mounted on the trailing arms (*LTA-RTA*) and the other set of data was the output of the sensor mounted on the vehicle body (*VB*). The coefficient of correlation utilized was the following:

$$\rho_{x,y} = \frac{Cov(X,Y)}{\sigma_x \sigma_y} \tag{6.1}$$

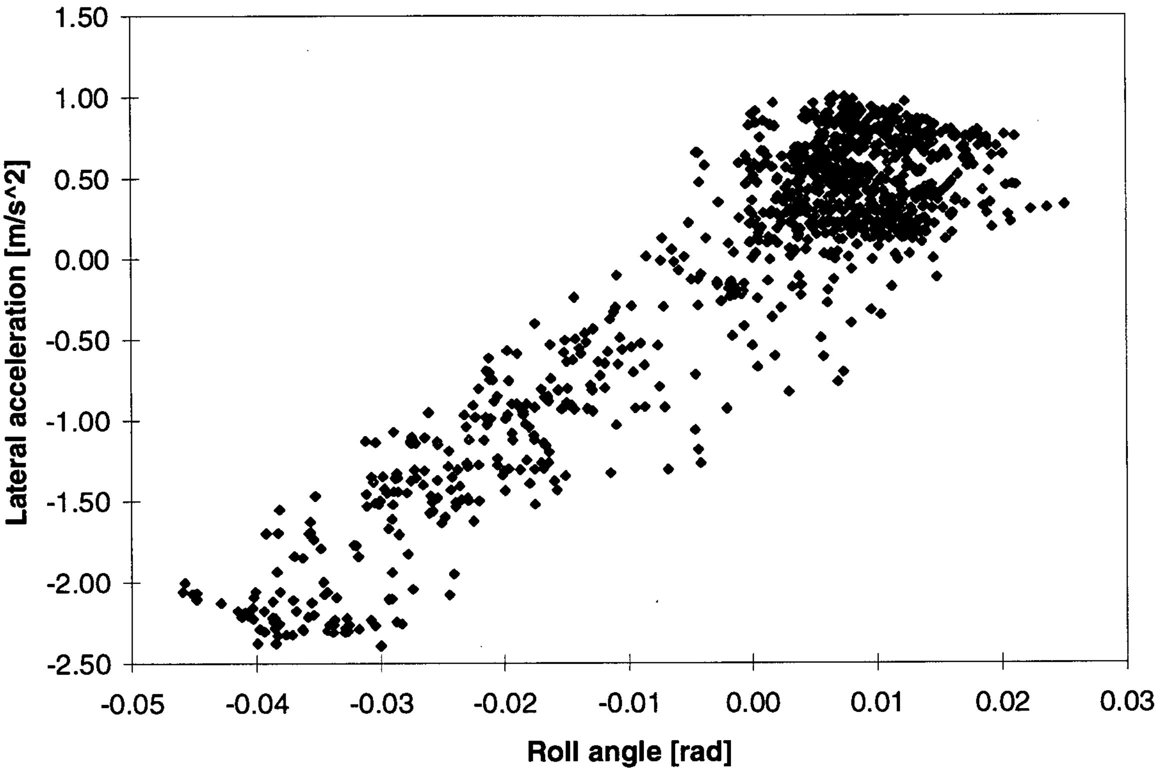
where *Cov(X,Y)* is the covariance of the *X* and *Y* sets of data and $\sigma_x$ and $\sigma_y$ are the standard deviations of the same sets of data, respectively. In this case the "*X*" set of data is the difference of outputs between *LTA* and *RTA* tilt sensors whereas the "*Y*" set of data is the output of the *VB* tilt sensor.

**Table 6.2.** Coefficients of correlation of the outputs *LTA-RTA* and *VB* sensors

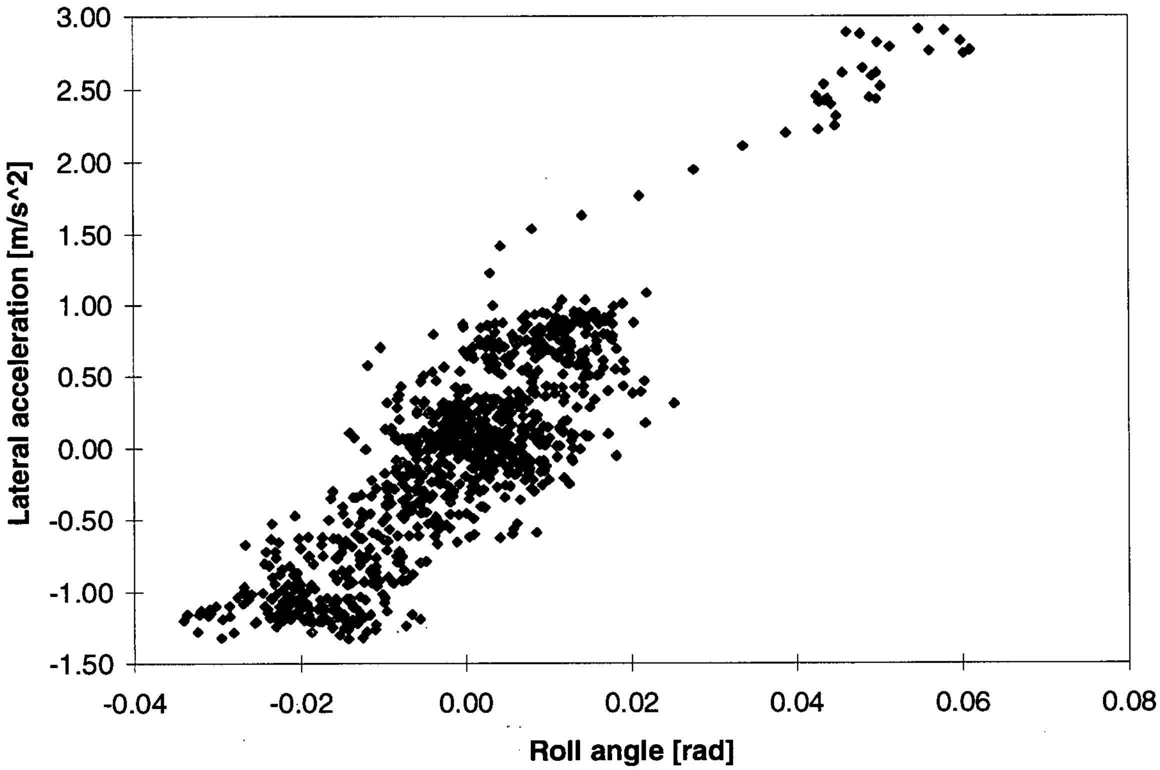
| **Test type (Plot)** | **Coefficient of correlation** |
|---|---|
| Negotiating two right turns on a smooth road (Figure 6.2.a) | 0.939 |
| Driving on a relatively smooth and sinuous road (Figure 6.2.b) | 0.901 |
| Driving on a relatively smooth and sinuous road (Figure 6.2.c) | 0.965 |
| Driving on a relatively rough and straight road (Figure 6.2.d) | 0.060 |

The results of Table 6.2 prove the existence of a quite high correlation between the two sets of data and, therefore a correlation between the outputs of the three tilt sensors does exist. This correlation appears to exist only on smooth roads and not on rough roads where the trailing arms may experience high amplitude oscillations, as plots of Figure 6.2.d and

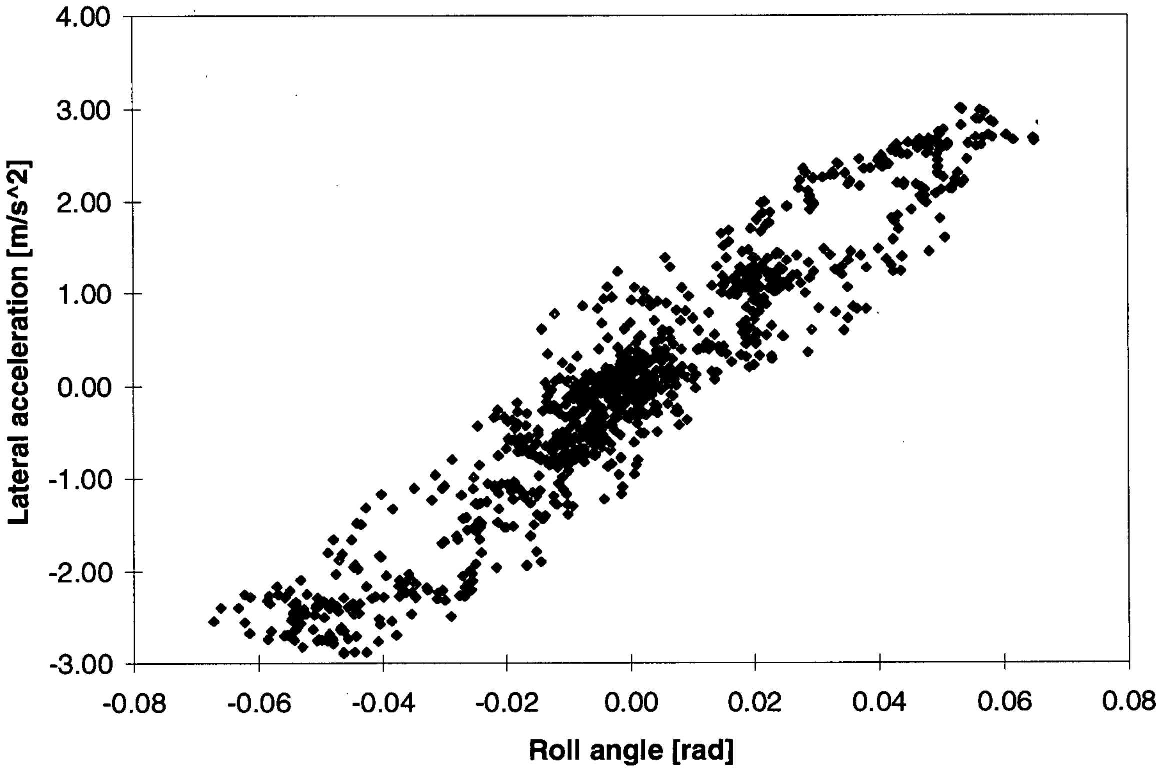
Appendix D4 show. This correlation appears fairly clearly when examining the spread of the plots of Figure 6.3 which show the dependency *lateral acceleration versus body roll angle* on smooth roads. One can notice the distribution of the 1000 plotted points along approximately the same line. The slope of the line is shown in Table 6.3.




(a)


3.00
2.50
2.00
1.50
1.00
0.50
0.00
-0.50
-1.00
-1.50
-0.04
-0.02
0.00
0.02
0.04
0.06
0.08
Lateral acceleration [m/s^2]
Roll angle [rad]

**(b)**




**(c)**

**Figure 6.3.** Lateral acceleration versus body roll angle during on road-tests:

**(a)** negotiating two right turns on a smooth road (as in Figure 6.2.a);

**(b)** driving on a relatively smooth and sinuous road (as in Figure 6.2.b);

**(c)** driving on a relatively smooth and sinuous road (as in Figure 6.2.c)

Static tests were also performed with the three tilt sensors mounted on the vehicle. The static tests contained two short time (about 5 seconds each) load shifts to the vehicle. The target was to see if the sensor system was able to sense the roll caused by the shift of a relatively small load (a human body). The plot of Appendix D1 shows the sensor placed on the vehicle body and those on the trailing arms indicated the same thing; small tilts towards left and then right respectively. During each tilt, the trailing arm tilt sensor on the side of the vehicle which was loaded decreased its output due to the combination of extra weight and applied moment, while the two effects counteracted each other on the opposite side tilt sensor.

## 6.3. Conclusions and future work

The purpose of the on-road tests was to determine whether useful signals could be obtained from the tilt sensors when mounted on a vehicle with a similar suspension to that of a heavy truck. The conclusion from the tests performed is that smooth, apparently clean, signals could be obtained in this circumstance so long as the road was smooth and the sensor readings were filtered at approximately 2 Hz. The strong correlation between the sensed lateral acceleration and the tilt of the body of the vehicle, as measured through the difference in trailing arm tilts, indicates that useful measurements were being made.

Future work will need to focus on quantitative measurements of the sensor outputs and roll behavior of actual heavy trucks, particularly tractor-trailer combinations. The first set of tests should be done on a tilt table. These tests must investigate the behavior of a roll sensing system in situations close to rollover threshold. These tests should demonstrate whether the system can detect the progression towards rollover. The tilt table tests will need to be performed using a variety of tractor-trailer combinations. The combination of static and dynamic tests should further confirm the usefulness of the roll sensing system. The key question is whether sensors mounted on the tractor alone are capable of sensing the roll behavior of the entire combination. If the roll sensing system is consistently able to determine the roll state of a heavy truck over a range of different tractors, trailers and payloads, then the next step would be to mount a number of these sensor packages on operating trucks with professional drivers. The feedback from those drivers would finally help to answer the fundamental question of whether the information provided is useful for decreasing the incidence of rollover accidents and what form that information should take to be of most use. The specifics of this testing program will need to be addressed much more carefully once a decision is made to take that step.

One of the possible outputs from the roll sensing system might be an estimation of the height of the center of gravity of the truck. The estimation would be based on a simple formula which can be obtained from the model presented in Chapter 3. For small angles, starting from expression (3.9) one can find the height of the center of mass of the vehicle as:

$$h_s = h_r + \frac{k_r}{W_s\left(1 + \frac{1}{\phi_s}\frac{a_y}{g}\right)} \tag{6.2}$$

The roll angle of the vehicle body can then be expressed in terms of the rotations of the trailing arms:

$$\phi_s = \frac{l}{T_t}(\alpha_1 - \alpha_2) \tag{6.3}$$

where: $l$ is the length of a trailing arm, $T_t$ is the spacing of the trailing arms and $\alpha_1$ and $\alpha_2$ are the two angles of rotation of the trailing arms, respectively.

Therefore the expression of the center of gravity height becomes:

$$h_s = h_r + \frac{k_r}{W_s\left(1 + \frac{T_t}{l(\alpha_1 - \alpha_2)}\frac{a_y}{g}\right)} \tag{6.4}$$

Assuming that the vehicle load can be measured and the other parameters which depend entirely on the vehicle geometry are known, the height of the center of mass can be expressed in terms of the outputs of the three tilt sensors: *LTA* and *RTA* tilt angles and *VB* lateral acceleration. Therefore it appears possible to obtain on-road estimation of the center of mass height. This information might help the driver in estimating the propensity for rollover of a heavy truck. Obviously this information would be useful for drivers who frequently pick-up trailers with different loads.

**Application**

As an illustration of the above approach of estimating the center of gravity height of a vehicle using the output of the roll sensing system, a calculation for the vehicle utilized during the tests was also done.

The following parameters of the vehicle were measured:

$T_f$=1.3 m;

$l$=0.6m;

$T$=0.75 m.

The following data was estimated:

$h_r$=0.35 m;

$m_b$=85 kg (the mass applied during the static tests);

$d$=1.6 m (the off-set of the mass applied during the static tests);

$M_v$=1400 kg (mass of the vehicle).

From the results of the static tests (Appendix D1), estimating the difference in tilt of the trailing arms of about $3^{o}$ ($\phi_{so}=3^{o}$) when the off-set load was applied, the roll stiffness of the vehicle was calculated to be $k_r$= 50961 Nm/rad with the following formula:

$$k_r = \frac{m_b dg}{\phi_{so} \frac{\pi}{180}} \qquad (6.5)$$

From the output of the sensor system, lateral acceleration was expressed directly in terms of the apparent tilt angle $\beta$ (measured between the tilt sensor mounting surface and the perpendicular on the resultant of lateral acceleration and gravity vectors) and the body roll angle, both measured in radians:

$$a_y = g\tan(\beta - \phi_S) \tag{6.6}$$

Finally, the center of gravity height could be calculated with expression of formula (6.4) slightly modified according to the available data:

$$h_s = h_r + \frac{k_r}{M_v\left(g + \dfrac{a_y}{\phi_S}\right)} \tag{6.7}$$

The ratio $a_y/\phi_S$ was expressed as the slope of the regression line for the recorded data. Calculations for data recorded on smooth roads lead to the results of Table 6.3.

**Table 6.3.** Center of gravity height estimation

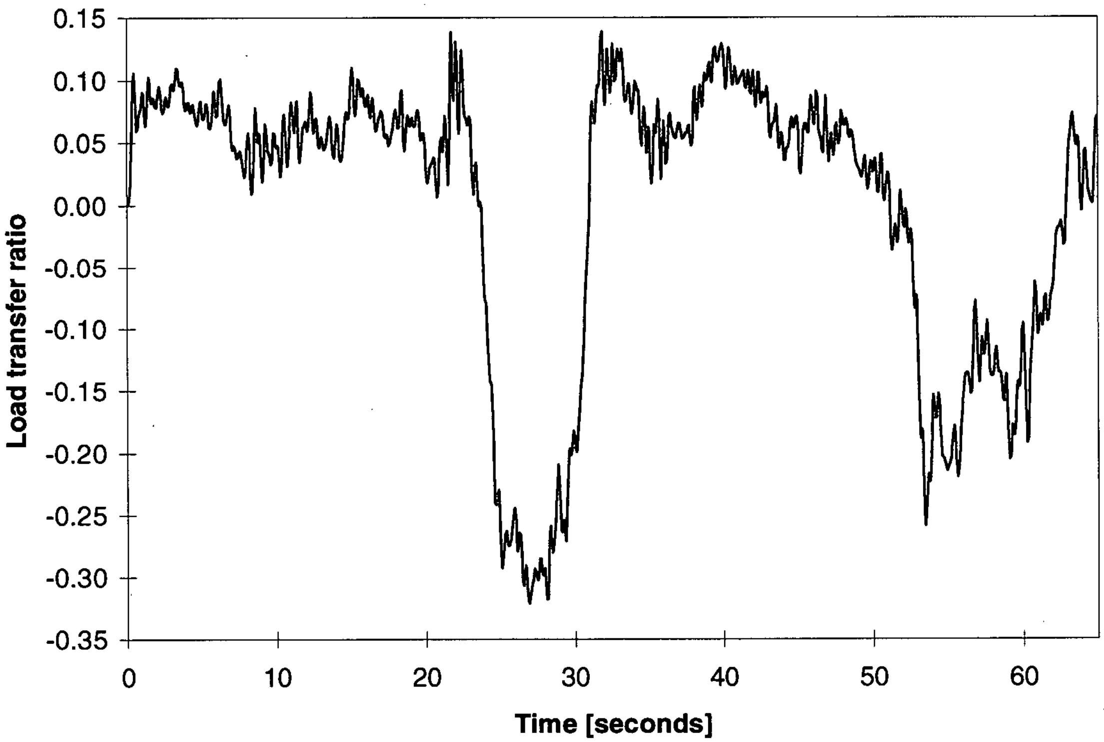
| **Test type (Plot)** | **Slope $a_y/\phi_S$ [m/s$^2$ / rad]** | **$h_S$ [m]** |
|---|---|---|
| Negotiating two right turns on a smooth road (Figure 6.2.a) | 51.95 | 0.94 |
| Driving on a relatively smooth and sinuous road (Figure 6.2.b) | 47.79 | 0.98 |
| Driving on a relatively smooth and sinuous road (Figure 6.2.c) | 49.09 | 0.97 |

As one can notice, the results are fairly consistent, although, due to numerous estimations made, the accuracy is not known. The relative consistency of the results shows that the estimation of the center of gravity height might be done on road, using the above method.

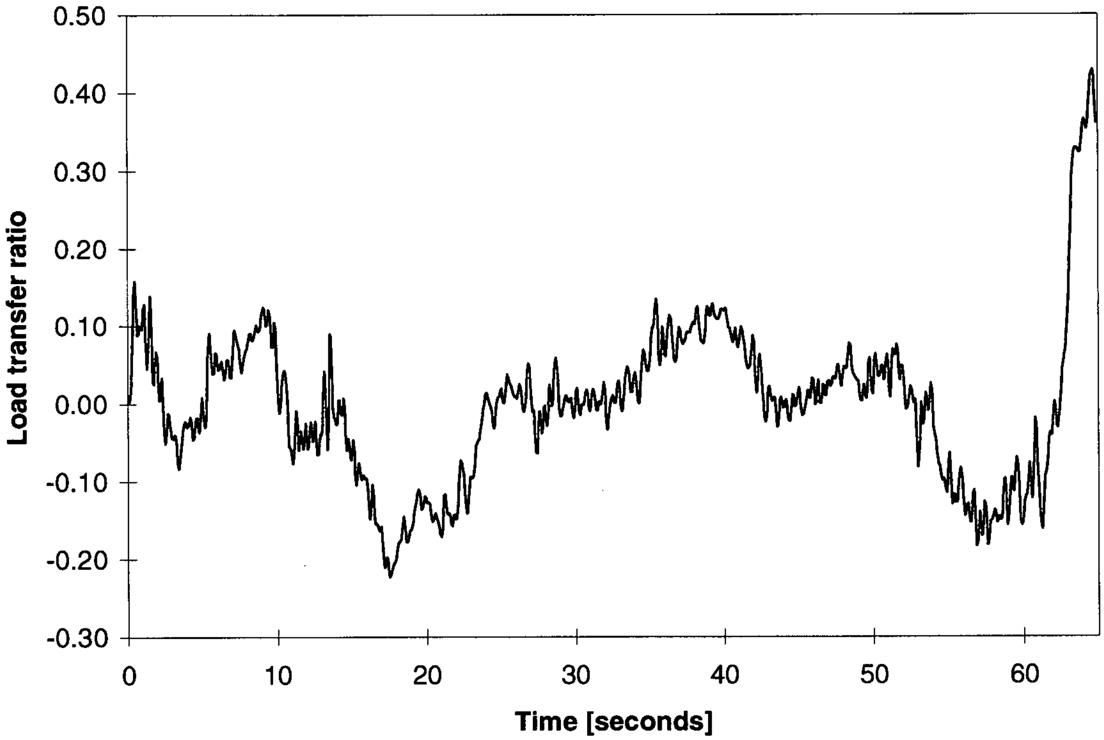
Another possible output of the tilt sensor system would be an estimation of the lateral load transfer ratio (*LTR*) which would show in a more suggestive manner the roll state of the vehicle. An approximate formula to express *LTR* would take into account the roll moment of the vehicle body with respect to its roll center and the roll moment at the roll center with respect to the ground. Therefore one can obtain:

$$LTR = \frac{\phi_s k_r + M_v g \tan(\beta - \phi_s) h_r}{M_v g T} \tag{6.8}$$

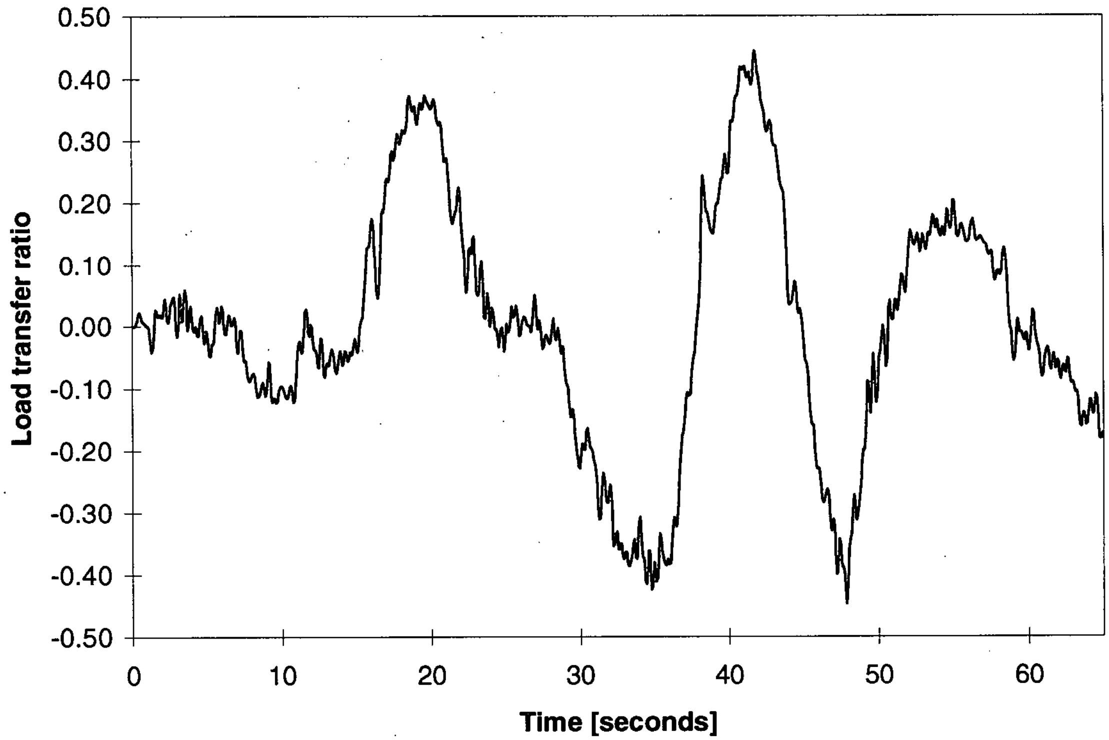
Based on the above formula, time dependencies of *LTR* were plotted for the data recorded on smooth road as well.


0.15
0.10
0.05
0.00
-0.05
-0.10
-0.15
-0.20
-0.25
-0.30
-0.35
Load transfer ratio
0
10
20
30
40
50
60
Time [seconds]

**(a)**


0.50
0.40
0.30
0.20
0.10
0.00
-0.10
-0.20
-0.30
Load transfer ratio
0
10
20
30
40
50
60
Time [seconds]

**(b)**




**(c)**

**Figure 6.4.** Vehicle roll sensing. Time variation of load transfer ratio (*LTR*):

**(a)** negotiating two right turns on a smooth road (as in Figure 6.2.a);

**(b)** driving on a relatively smooth and sinuous road (as in Figure 6.2.b);

**(c)** driving on a relatively smooth and sinuous road (as in Figure 6.2.c)

Plots of Figure 6.4 show that, as expected, the *LTR* coefficient has the same time tendency as the body roll angle and the lateral acceleration. Its variation from about -0.45 to +0.45 (as maximal amplitudes) shows the load shift towards right and left, respectively, on road.

# Chapter 7

# SUMMARY AND CONCLUSIONS

Statistics of road accidents in North America prove that rollover is the greatest cause of occupant fatalities and injuries in accidents involving heavy trucks as well as a major cause of environment damage by bulk spillage of hazardous commodities carried by these trucks. Therefore the study of rollover of heavy trucks with a view to reducing its incidence has been an important field of research for at least about 25 years. However no widely accepted solution of the problem has been found so far.

The objective of the present work has been to investigate whether it is possible to sense, in a simple and accurate manner, the roll behavior of heavy trucks with a view to decreasing the incidence of rollover accidents. The basic idea has been to obtain useful signals which could be used by a rollover warning device to warn the driver about the progression towards rollover. Having this information, the driver might be able to take the necessary restoring actions in order to prevent the approaching accident. If sufficient early warning were not possible, a rollover warning device might still be useful for educating the driver by feeding back to him how close to a rollover he had come.

Studies of the mechanism of the rollover of heavy trucks found in the available literature have proven that the rollover threshold is closely associated, in most cases, with the tractor internal rear wheel lift-off. Despite this relatively obvious observation, no simple and widely accepted way to detect the approach of the rollover threshold exists in the trucking industry or in the available literature.

The present study has proposed a method for sensing the roll motion of heavy trucks by utilizing an arrangement of tilt sensors placed in the rear part of the tractor of a heavy truck. In order to assess the suitability of a commercial tilt sensor for the proposed sensing system, the sensor was subjected to various static, dynamic and on-road tests. A system of three tilt sensors mounted in the rear-part of a minivan featuring a tractor-like rear axle was also tested on-road. As a result of the tests done, it appears that an arrangement of tilt sensors may be suitable for roll sensing.

However, it is obvious that before assessing the technical suitability of the proposed roll sensing approach, extensive tests using heavy trucks should be performed. If, after future tests and investigations, the roll sensing method presented in this work proves to be of some help for monitoring the roll state of a heavy truck, the next step of the research would be the design problem of a rollover warning device based on the above method. The design problem should be done according to both the existing background found in the available literature and the inputs from the trucking industry. The work will be complete once economic feasibility of a rollover warning device based on a tilt sensors arrangement, as described in this work, is assessed.

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**Appendix A.** Large Trucks Involved in Fatal Crashes by Most Harmful Event and Manner of Collision (from FARS 1991)

| First Harmful Event Collision With: | Vehicles Involved | | Occupant Fatalities | |
|---|---|---|---|---|
| | Number | Percent | Number | Percent |
| **Motor Vehicle in Transport** | **3,354** | **77.3** | **189** | **28.7** |
| Head-On | 1,024 | 23.6 | 42 | 6.4 |
| Rear-End | 766 | 17.6 | 69 | 10.5 |
| Angle | 1,374 | 31.7 | 61 | 9.3 |
| Side-Swipe | 179 | 4.1 | 14 | 2.1 |
| Other/Unknown | 11 | 0.3 | 3 | 0.5 |
| **Fixed Object** | **323** | **7.4** | **240** | **36.4** |
| Shrubbery/Tree | 27 | 0.6 | 23 | 3.5 |
| Pole or Post | 35 | 0.8 | 28 | 4.2 |
| Culvert/Curb/Ditch | 30 | 0.7 | 21 | 3.2 |
| Embankment | 28 | 0.6 | 26 | 3.9 |
| Guard Rail | 111 | 2.6 | 87 | 13.2 |
| Bridge | 33 | 0.8 | 26 | 3.9 |
| Other | 59 | 1.4 | 29 | 4.4 |
| **Object not Fixed** | **81** | **1.9** | **45** | **6.8** |
| Train | 24 | 0.6 | 24 | 3.6 |
| Parked Motor Vehicle | 44 | 1.0 | 15 | 2.3 |
| Animal | 6 | 0.1 | 3 | 0.5 |
| Other | 7 | 0.2 | 3 | 0.5 |
| **Nonmotorist** | **367** | **8.5** | **0** | **0.0** |
| Pedestrian | 292 | 6.7 | 0 | 0.0 |
| Pedalcyclist | 71 | 1.6 | 0 | 0.0 |
| Other Nonmotorist | 4 | 0.1 | 0 | 0.0 |
| **Noncollision** | **215** | **5.0** | **185** | **28.1** |
| Overturn | 177 | 4.1 | 157 | 23.8 |
| Other | 38 | 0.9 | 28 | 4.2 |
| **Total** | **4,340** | **100.0** | **659** | **100.0** |

**Appendix B-1.** Static tests. Actual tilt angle and measured tilt angle

| Actual angle [degrees] | Measured angle (data mean) [degrees] | Data standard deviation [degrees] |
|---|---|---|
| -90 | -85.18 | 0.18 |
| -75 | -69.84 | 0.16 |
| -60 | -57.27 | 0.12 |
| -45 | -48.25 | 0.12 |
| -40 | -45.36 | 0.11 |
| -35 | -41.96 | 0.09 |
| -30 | -37.36 | 0.10 |
| -27 | -34.36 | 0.08 |
| -24 | -31.38 | 0.11 |
| -21 | -27.97 | 0.07 |
| -18 | -24.24 | 0.10 |
| -15 | -20.33 | 0.07 |
| -14 | -19.06 | 0.05 |
| -13 | -17.63 | 0.10 |
| -12 | -16.01 | 0.09 |
| -11 | -14.77 | 0.09 |
| -10 | -13.42 | 0.09 |
| -9 | -12.14 | 0.09 |
| -8 | -10.58 | 0.06 |
| -7 | -9.41 | 0.10 |
| -6 | -8.16 | 0.08 |
| -5 | -6.94 | 0.07 |
| -4 | -5.50 | 0.08 |

| | | |
|---|---|---|
| -3 | -4.11 | 0.10 |
| -2 | -2.75 | 0.09 |
| -1 | -1.33 | 0.12 |
| 0 | 0 | 0.07 |
| 1 | 1.16 | 0.09 |
| 2 | 2.10 | 0.08 |
| 3 | 3.55 | 0.04 |
| 4 | 4.55 | 0.11 |
| 5 | 5.75 | 0.07 |
| 6 | 6.95 | 0.04 |
| 7 | 7.79 | 0.08 |
| 8 | 9.02 | 0.10 |
| 9 | 10.20 | 0.07 |
| 10 | 11.37 | 0.08 |
| 11 | 12.38 | 0.09 |
| 12 | 13.36 | 0.04 |
| 13 | 14.62 | 0.04 |
| 14 | 15.53 | 0.08 |
| 15 | 16.47 | 0.08 |
| 18 | 19.58 | 0.13 |
| 21 | 22.18 | 0.12 |
| 24 | 24.88 | 0.07 |
| 27 | 27.69 | 0.10 |
| 30 | 30.41 | 0.11 |
| 35 | 34.62 | 0.10 |
| 40 | 38.26 | 0.11 |
| 45 | 42.65 | 0.12 |

| | | |
|---|---|---|
| 60 | 57.51 | 0.12 |
| 75 | 76.76 | 0.17 |
| 90 | 104.38 | 0.23 |

**Appendix B-2.** Static tests. Loading (increasing tilt) and unloading (decreasing tilt) between the horizontal position and a $\pm 90^{0}$ position (sample size= 500 readings)

| **Actual angle [degrees]** | **Measured data** | | | |
|---|---|---|---|---|
| | **Loading (Increasing tilt)** | | **Unloading (Decreasing tilt)** | |
| | **Mean [degrees]** | **Standard deviation [degrees]** | **Mean [degrees]** | **Standard deviation [degrees]** |
| -90 | -85.18 | 0.11 | -85.18 | 0.18 |
| -75 | -69.84 | 0.16 | -69.94 | 0.15 |
| -60 | -57.27 | 0.12 | -57.21 | 0.15 |
| -45 | -48.25 | 0.12 | -48.26 | 0.12 |
| -30 | -37.36 | 0.10 | -37.32 | 0.10 |
| -15 | -20.33 | 0.07 | -20.43 | 0.08 |
| -10 | -13.42 | 0.09 | -13.52 | 0.08 |
| -5 | -6.94 | 0.07 | -7.08 | 0.06 |
| 0 | 0 | 0.07 | -0.43; -0.68 | 0.07; 0.07 |
| 5 | 5.75 | 0.07 | 5.35 | 0.07 |
| 10 | 11.37 | 0.08 | 11.21 | 0.08 |
| 15 | 16.47 | 0.08 | 16.50 | 0.13 |
| 30 | 30.41 | 0.11 | 30.24 | 0.10 |
| 45 | 42.65 | 0.12 | 42.93 | 0.13 |
| 60 | 57.51 | 0.12 | 57.93 | 0.14 |
| 75 | 76.76 | 0.17 | 76.59 | 0.18 |
| 90 | 104.38 | 0.23 | 104.38 | 0.23 |

**Appendix B-3.** Static tests. Tilting from horizontal position to a ± $30^{O}$ position with -$10^{O}$ cross tilt angle (sample size= 500 readings)

| Actual angle [degrees] | Measured data | | | |
|---|---|---|---|---|
| | Loading (Increasing tilt) | | Unloading (Decreasing tilt) | |
| | Mean [degrees] | Standard deviation [degrees] | Mean [degrees] | Standard deviation [degrees] |
| -30 | -37.58 | 0.11 | -37.58 | 0.11 |
| -25 | -32.81 | 0.07 | -32.72 | 0.09 |
| -20 | -26.80 | 0.08 | -26.80 | 0.06 |
| -15 | -20.37 | 0.10 | -21.25 | 0.07 |
| -10 | -13.26 | 0.05 | -13.38 | 0.09 |
| -5 | -6.46 | 0.04 | -6.86 | 0.07 |
| 0 | 0 | 0.09; 0.08 | -0.47; -0.14 | 0.08; 0.08 |
| 5 | 5.99 | 0.02 | 5.46 | 0.09 |
| 10 | 11.56 | 0.08 | 11.42 | 0.08 |
| 15 | 16.65 | 0.08 | 16.43 | 0.09 |
| 20 | 21.27 | 0.06 | 21.27 | 0.07 |
| 25 | 25.49 | 0.09 | 25.55 | 0.07 |
| 30 | 29.62 | 0.09 | 29.62 | 0.09 |

**Appendix B-4.** Static tests. Tilting from horizontal position to a ± $30^{O}$ position with $5^{O}$ cross tilt angle (sample size= 500 readings)

| Actual angle [degrees] | Measured data | | | |
|---|---|---|---|---|
| | Loading (Increasing tilt) | | Unloading (Decreasing tilt) | |
| | Mean [degrees] | Standard deviation [degrees] | Mean [degrees] | Standard deviation [degrees] |
| -30 | -36.64 | 0.13 | -36.64 | 0.13 |
| -25 | -31.75 | 0.07 | -31.72 | 0.07 |
| -20 | -26.00 | 0.10 | -26.00 | 0.10 |
| -15 | -19.58 | 0.11 | -19.63 | 0.11 |
| -10 | -12.65 | 0.09 | -13.10 | 0.07 |
| -5 | -6.53 | 0.12 | -6.73 | 0.09 |
| 0 | 0 | 0.09; 0.08 | -0.26; -0.11 | 0.08; 0.08 |
| 5 | 5.96 | 0.08 | 5.79 | 0.10 |
| 10 | 11.29 | 0.08 | 11.17 | 0.10 |
| 15 | 16.31 | 0.11 | 16.40 | 0.09 |
| 20 | 20.90 | 0.09 | 20.98 | 0.11 |
| 25 | 25.89 | 0.10 | 25.89 | 0.10 |
| 30 | 30.43 | 0.10 | 30.43 | 0.10 |

**Appendix B-5.** Static tests. Tilting from horizontal position to a $\pm 30^{0}$ position with $15^{0}$ cross tilt angle (sample size= 500 readings)

| Actual angle [degrees] | Measured data | | | |
|---|---|---|---|---|
| | Loading (Increasing tilt) | | Unloading (Decreasing tilt) | |
| | Mean [degrees] | Standard deviation [degrees] | Mean [degrees] | Standard deviation [degrees] |
| -30 | -35.93 | 0.11 | -35.93 | 0.11 |
| -25 | -31.03 | 0.09 | -31.52 | 0.09 |
| -20 | -24.98 | 0.10 | -25.95 | 0.09 |
| -15 | -18.94 | 0.08 | -19.58 | 0.05 |
| -10 | -12.22 | 0.08 | -12.99 | 0.10 |
| -5 | -6.22 | 0.03 | -6.20 | 0.03 |
| 0 | 0 | 0.10; 0.06 | 0.17; 0.01 | 0.06; 0.04 |
| 5 | 5.50 | 0.05 | 5.37 | 0.10 |
| 10 | 10.79 | 0.12 | 10.79 | 0.12 |
| 15 | 15.92 | 0.09 | 15.75 | 0.11 |
| 20 | 20.38 | 0.08 | 20.49 | 0.10 |
| 25 | 25.72 | 0.10 | 25.55 | 0.10 |
| 30 | 30.65 | 0.10 | 30.65 | 0.10 |

**Appendix B-6.** Static tests. Tilting from horizontal position to a $\pm 30^{0}$ position with $30^{0}$ cross tilt angle (sample size= 500 readings)

| **Actual angle [degrees]** | **Measured data** | | | |
|---|---|---|---|---|
| | **Loading (Increasing tilt)** | | **Unloading (Decreasing tilt)** | |
| | **Mean [degrees]** | **Standard deviation [degrees]** | **Mean [degrees]** | **Standard deviation [degrees]** |
| -30 | -35.53 | 0.13 | -35.53 | 0.13 |
| -25 | -30.52 | 0.12 | -30.48 | 0.12 |
| -20 | -24.68 | 0.05 | -24.67 | 0.05 |
| -15 | -18.53 | 0.09 | -18.52 | 0.09 |
| -10 | -12.29 | 0.10 | -12.31 | 0.08 |
| -5 | -6.39 | 0.08 | -6.26 | 0.09 |
| 0 | 0 | 0.06; 0.04 | 0.02; 0.01 | 0.04; 0.06 |
| 5 | 5.10 | 0.04 | 5.14 | 0.11 |
| 10 | 10.23 | 0.10 | 10.20 | 0.05 |
| 15 | 15.26 | 0.09 | 15.21 | 0.10 |
| 20 | 19.99 | 0.08 | 20.14 | 0.10 |
| 25 | 25.93 | 0.11 | 25.73 | 0.10 |
| 30 | 31.58 | 0.13 | 31.58 | 0.13 |

**Appendix B-7.** Static tests. Tilting from horizontal position to a $\pm 30^{\circ}$ position with $45^{\circ}$ cross tilt angle (sample size= 500 readings)

| **Actual angle [degrees]** | **Measured data** | | | |
|---|---|---|---|---|
| | **Loading (Increasing tilt)** | | **Unloading (Decreasing tilt)** | |
| | **Mean [degrees]** | **Standard deviation [degrees]** | **Mean [degrees]** | **Standard deviation [degrees]** |
| -30 | -35.68 | 0.07 | -35.68 | 0.07 |
| -25 | -30.51 | 0.08 | -30.28 | 0.10 |
| -20 | -24.15 | 0.08 | -24.39 | 0.09 |
| -15 | -17.08 | 0.04 | -17.86 | 0.04 |
| -10 | -12.26 | 0.07 | -12.42 | 0.09 |
| -5 | -5.72 | 0.12 | -5.88 | 0.07 |
| 0 | 0 | 0.02; 0.02 | 0.07; 0.87 | 0.02; 0.09 |
| 5 | 5.95 | 0.09 | 5.81 | 0.09 |
| 10 | 11.20 | 0.09 | 11.09 | 0.10 |
| 15 | 16.39 | 0.08 | 16.28 | 0.10 |
| 20 | 21.93 | 0.11 | 22.05 | 0.11 |
| 25 | 28.89 | 0.12 | 28.68 | 0.12 |
| 30 | 34.98 | 0.12 | 34.98 | 0.12 |

**Appendix C-1.** Lab dynamic tests. Horizontal shaking

| Freq. | Acceler. | Measured angle | | Data filtered at 5 Hz | | Data filtered at 10 Hz | | Actual angle | |
|---|---|---|---|---|---|---|---|---|---|
| | amplit. | Std. dev. | RMS | Std. dev. | RMS | Std. dev. | RMS | Std. dev. | RMS |
| [Hz] | [m/s$^2$] | [deg.] | [deg.] | [deg.] | [deg.] | [deg.] | [deg.] | [deg.] | [deg.] |
| 0 | 0 | 0.11 | 19.01 | 6.69 | 6.66 | 4.91 | 4.88 | 0 | 0 |
| 2.42 | 0.55 | 6.38 | 12.94 | 2.73 | 9.78 | 2.51 | 10.53 | 2.27 | 2.26 |
| 3.25 | 0.82 | 10.63 | 15.30 | 2.97 | 9.75 | 3.00 | 10.66 | 3.40 | 3.39 |
| 3.75 | 0.90 | 12.20 | 16.31 | 3.10 | 9.65 | 3.73 | 10.83 | 3.71 | 3.71 |
| 4.87 | 0.95 | 12.23 | 16.37 | 3.40 | 9.70 | 3.49 | 10.76 | 3.91 | 3.90 |
| 5.97 | 1.43 | 4.60 | 12.10 | 3.01 | 9.95 | 4.29 | 11.30 | 5.87 | 5.85 |
| 7.37 | 2.45 | 6.82 | 12.96 | 3.16 | 9.80 | 6.74 | 11.98 | 9.96 | 9.94 |
| 8.62 | 2.48 | 9.94 | 14.68 | 3.03 | 9.40 | 9.41 | 13.31 | 10.06 | 10.04 |
| 10.25 | 4.15 | 15.18 | 18.22 | 3.26 | 9.25 | 13.47 | 16.47 | 16.44 | 16.39 |
| 14.05 | 9.25 | 27.59 | 28.80 | 3.62 | 7.80 | 3.50 | 8.32 | 32.30 | 32.22 |
| 20.12 | 35.55 | 72.27 | 73.09 | 2.72 | 9.25 | 2.48 | 10.14 | 63.00 | 62.85 |
| 21.75 | 45.00 | 84.04 | 85.01 | 2.18 | 11.15 | 9.39 | 15.04 | 67.07 | 66.90 |
| 22.87 | 46.05 | 85.55 | 86.27 | 6.65 | 9.93 | 7.88 | 11.63 | 67.44 | 67.27 |
| 25.12 | 41.77 | 70.61 | 71.36 | 2.89 | 8.97 | 5.64 | 10.78 | 65.12 | 64.96 |
| 27.12 | 35.48 | 53.62 | 54.76 | 3.71 | 10.56 | 4.24 | 11.60 | 62.97 | 62.81 |
| 30.25 | 30.30 | 39.14 | 40.61 | 3.14 | 9.98 | 2.69 | 10.59 | 59.95 | 59.80 |
| 32.17 | 26.27 | 34.08 | 35.90 | 3.00 | 10.15 | 2.56 | 10.90 | 57.13 | 56.98 |
| 38.37 | 22.45 | 27.48 | 30.11 | 2.83 | 11.30 | 2.56 | 12.30 | 53.59 | 53.46 |
| 44.12 | 24.00 | 23.05 | 26.29 | 2.13 | 10.70 | 1.52 | 11.60 | 55.07 | 54.94 |
| 49.25 | 21.48 | 20.38 | 24.24 | 2.00 | 11.18 | 1.42 | 12.13 | 52.59 | 52.46 |

**Appendix C-2.** Lab dynamic tests. Vertical shaking

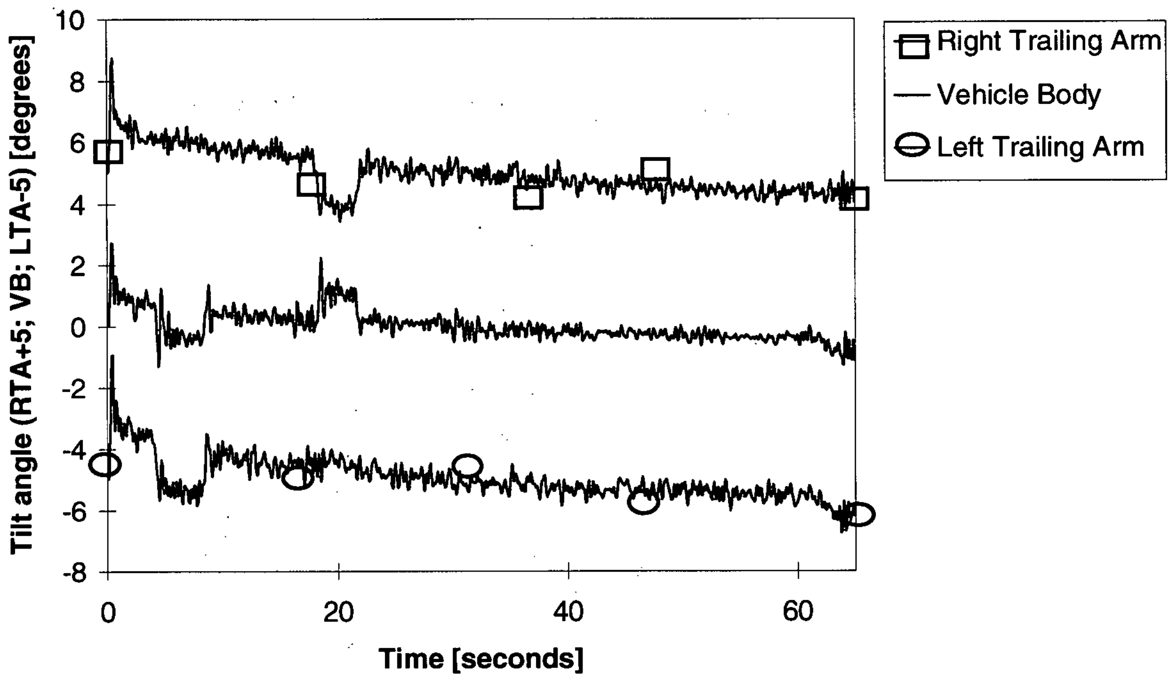
| Frequency [Hz] | Measured angle standard deviation [degrees] | Measured angle std. dev./ Measured angle motionless standard deviation [degrees] |
|---|---|---|
| 0 | 0.13 | 1 |
| 2.32 | 0.37 | 2.85 |
| 3.00 | 0.54 | 4.15 |
| 3.60 | 0.68 | 5.23 |
| 4.42 | 0.72 | 5.54 |
| 5.42 | 0.47 | 3.61 |
| 7.00 | 0.15 | 1.15 |
| 8.65 | 0.17 | 1.31 |
| 10.15 | 0.23 | 1.77 |
| 14.10 | 0.72 | 5.54 |
| 18.80 | 7.32 | 56.31 |
| 20.12 | 13.61 | 104.69 |
| 25.50 | 7.52 | 57.85 |
| 30.50 | 4.40 | 33.85 |
| 35.62 | 2.15 | 16.54 |
| 41.50 | 1.72 | 13.23 |
| 46.37 | 1.57 | 12.08 |
| 53.25 | 1.38 | 10.61 |
| 60.25 | 1.36 | 10.46 |
| 66.75 | 1.36 | 10.46 |
| 73.25 | 1.44 | 11.08 |
| 79.75 | 1.40 | 10.77 |

**Appendix D-1.** Vehicle roll sensing. Vehicle engine idle, car motionless

Small roll movements were artificially induced.

Plot based on 10,000 data samples.

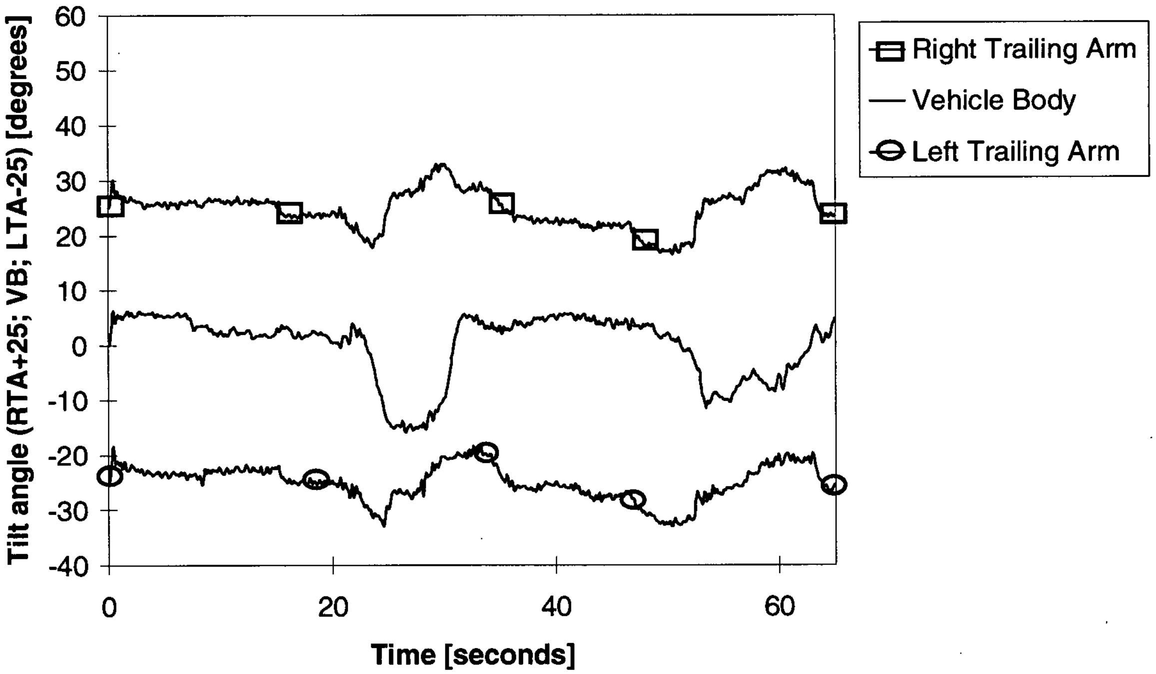
Data was filtered with a 2 Hz low-pass filter.



**Appendix D-2.** Vehicle roll sensing. Driving on smooth road in UBC area.
Negotiating two right turns at moderate speed.

Plot based on 10,000 data samples.

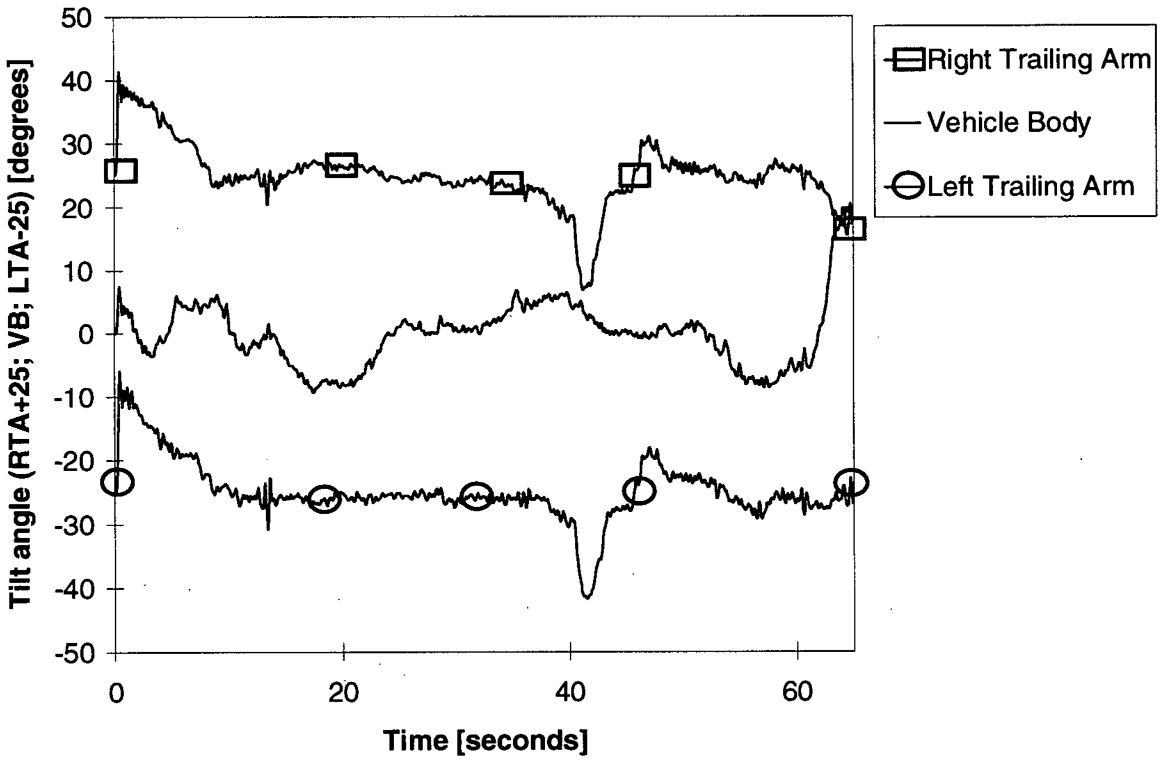
Data was filtered with a 2 Hz low-pass filter.

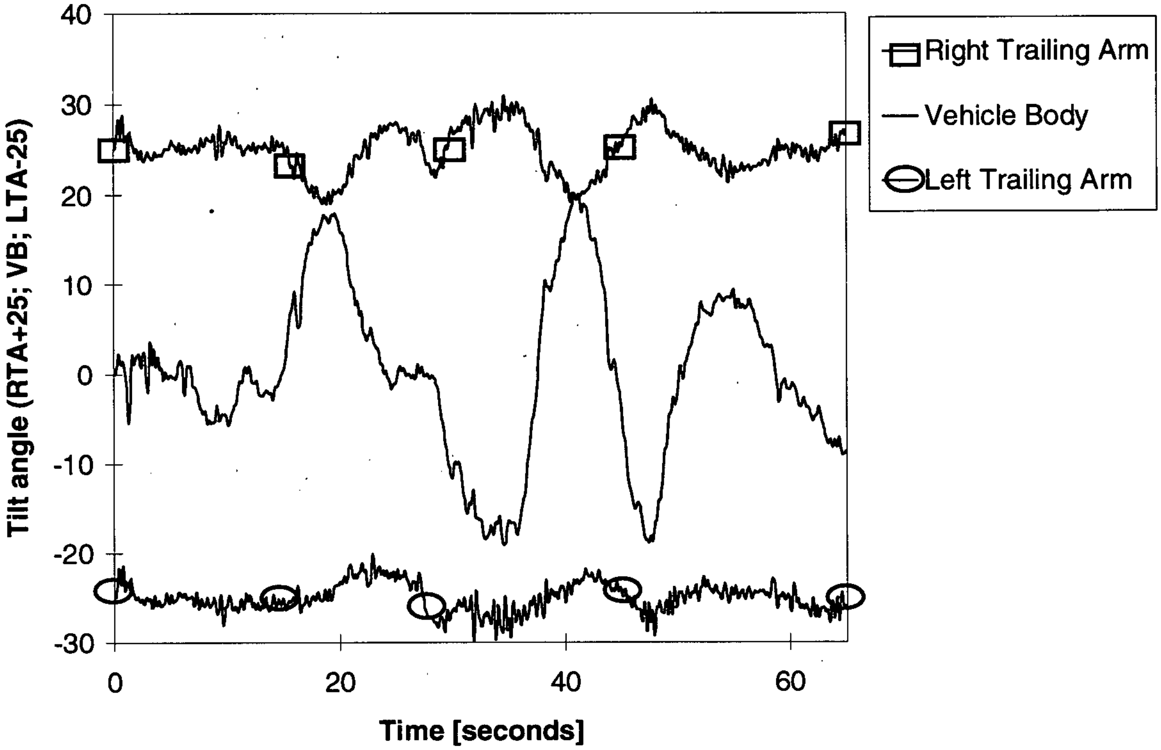


**Appendix D-3.** Vehicle roll sensing. Driving at moderate speed on smooth but sinuous road in UBC area

Plots based on 10,000 data samples recorded in separate files.

Data was filtered with a 2 Hz low-pass filter.

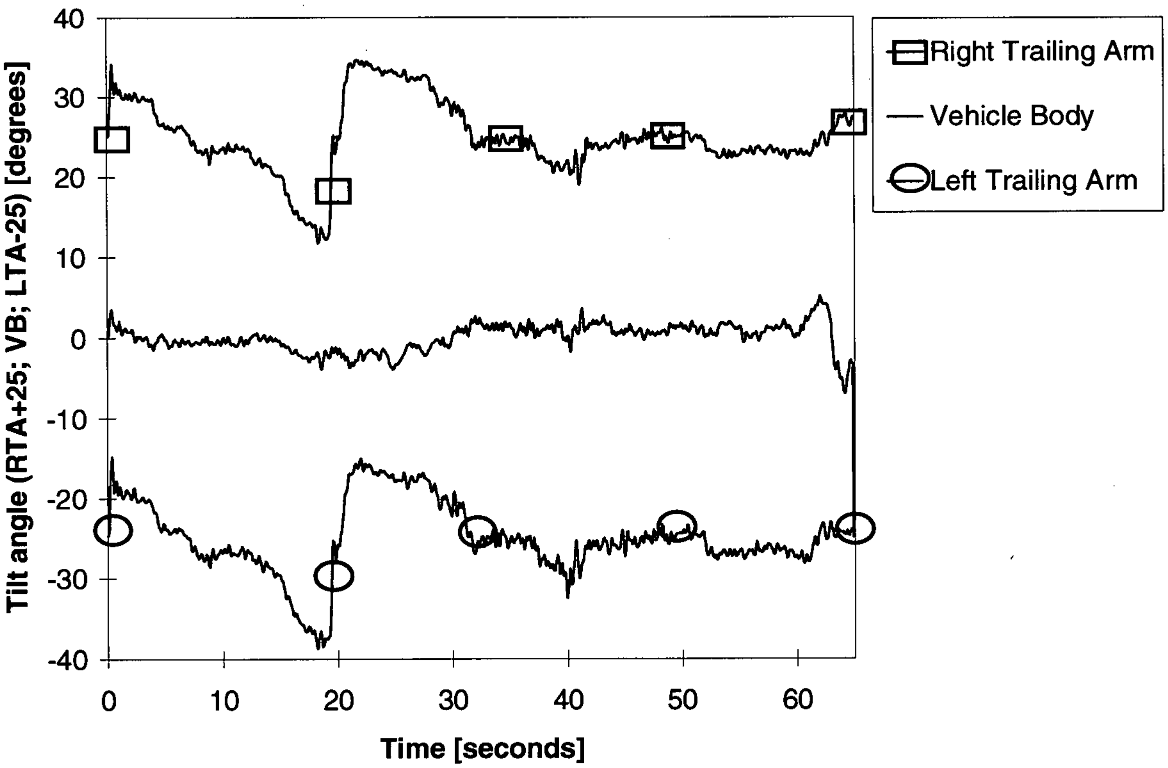


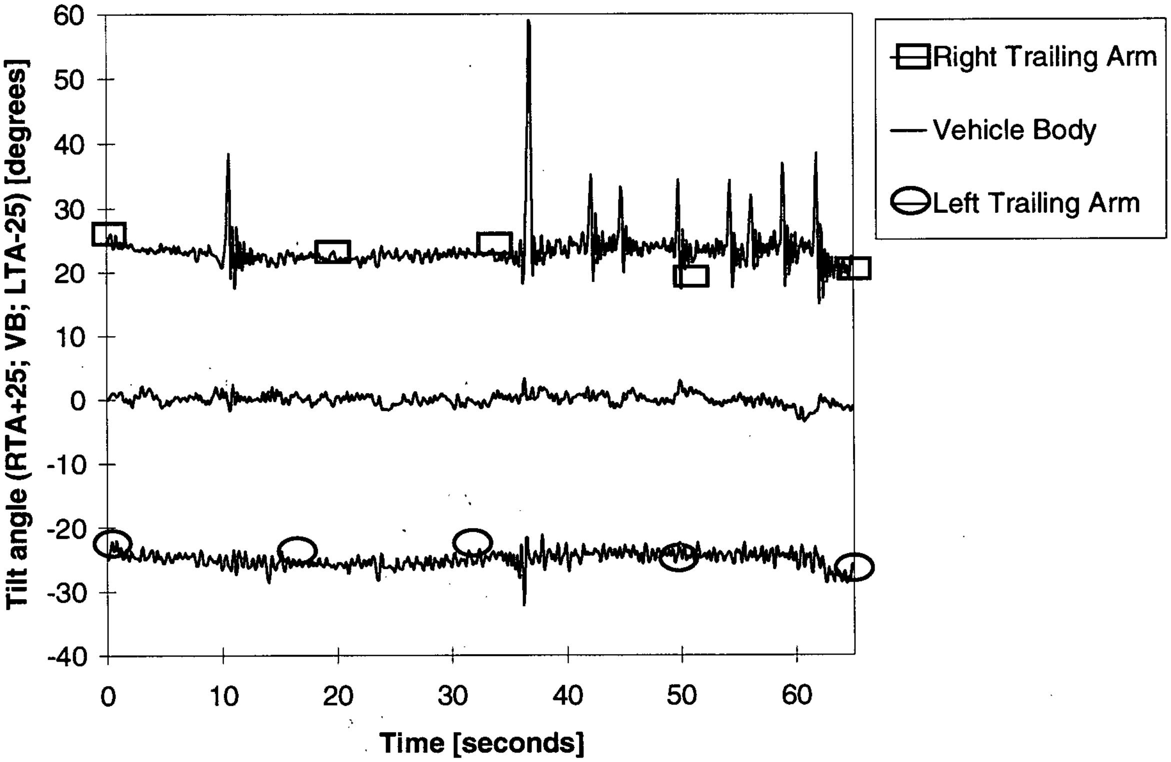

40
30
20
10
0
-10
-20
-30
Tilt angle (RTA+25; VB; LTA-25)
0
20
40
60
Time [seconds]
Right Trailing Arm
Vehicle Body
Left Trailing Arm

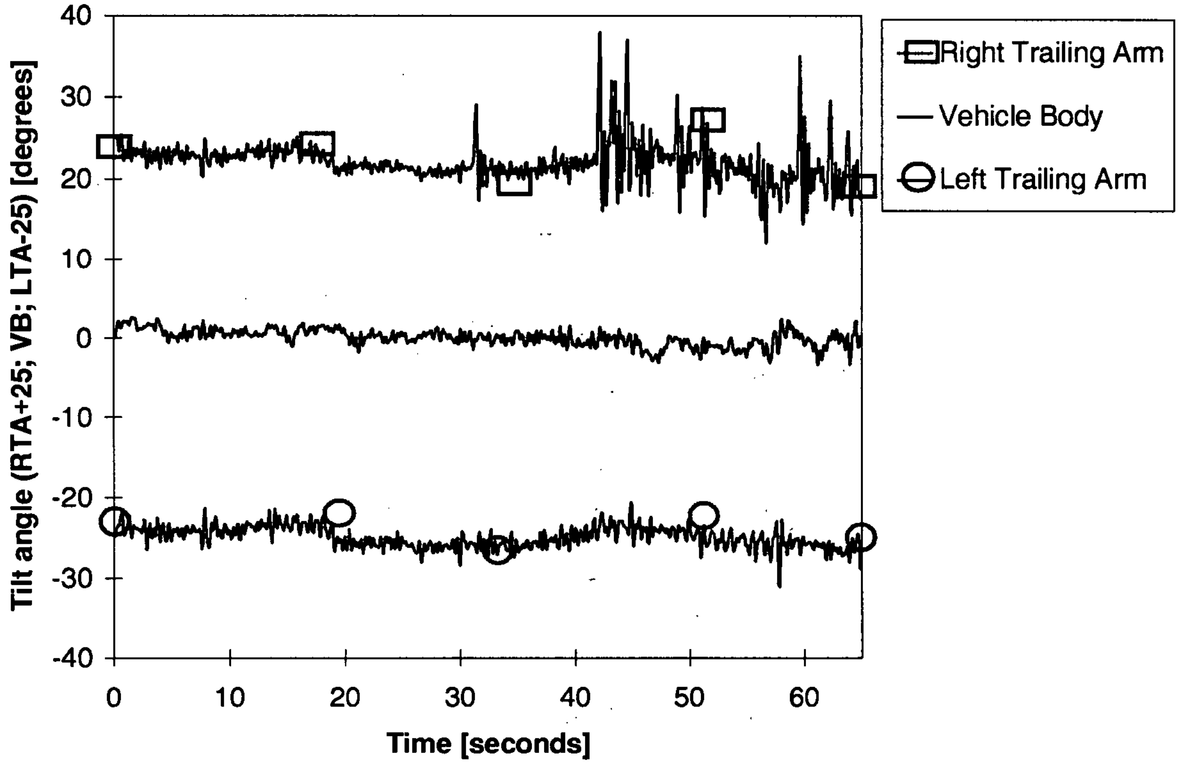
**Appendix D-4.** Vehicle roll sensing. Driving at moderate speed on relatively rough and straight road in UBC area

Plots based on 10,000 data samples recorded in separate files.

Data was filtered with a 2 Hz low-pass filter.




Right Trailing Arm
Vehicle Body
Left Trailing Arm
Tilt angle (RTA+25; VB; LTA-25) [degrees]
Time [seconds]
60
50
40
30
20
10
0
-10
-20
-30
-40
0
10
20
30
40
50
60



40
30
20
10
0
-10
-20
-30
-40
Tilt angle (RTA+25; VB; LTA-25) [degrees]
0
10
20
30
40
50
60
Time [seconds]
Right Trailing Arm
Vehicle Body
Left Trailing Arm