Rollover Prevention of Articulated Vehicles through Predictive Warning of Impending Roll Instability

Sandeep Kar

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ABSTRACT

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Articulated vehicles owing to their large weights and dimensions and consequentially higher cg heights, are more susceptible to rollover related accidents. The onset of roll instability in articulated vehicles, if assessed accurately and effectively, would lead to the reduction in occurrence of rollover accidents through early warning, thereby preventing the risk of potential loss of lives and property worth millions of dollars annually. The effective detection of an impending rollover hinges on development of reliable and feasible rollover metrics that are capable of providing good lead-time for open-loop rollover warning and control, over a wide range of design and operating conditions.

The state-of-the-art in rollover metrics, strategies and warning systems is discussed. The relative rollover condition for an articulated tractor-semitrailer combination is established as the premise for determination of an effective measure of roll instability for open-loop rollover warning. The reported concepts in open-loop rollover control are critically assessed in terms of their rollover detection potential, performance and reliability, ease of measurement and lead-time, for identifying an effective early warning and control strategy. The sensitivity analyses of the proposed measures to variations in design and operating conditions revealed considerable variations in all of the suggested measures. The roll safety factor (*RSF*) is identified as the most reliable indicator, which relates directly to the relative roll instability condition.

Owing to its poor measurability, an alternate measure that correlates well with *RSF* is explored. A new measure, termed as normalized roll-response of the semitrailer sprung mass (*NRSSM*), based upon the roll response of the sprung and unsprung masses is proposed as a superior measure of the onset of a relative roll instability. The measure incorporates the rearward amplification tendency of the combination and the lead unit in roll, roll deflection transmission characteristics of the lead unit and the response of the semitrailer sprung mass. Comprehensive relative assessments of the reported and the proposed rollover metrics are performed over a wide range of design and operating conditions. The results suggest that the *NRSSM* is least sensitive to variations in the design and operating conditions, and provided enhanced lead-time performance. A warning strategy based upon *NRSSM* together with the semitrailer lateral acceleration response is proposed to enhance the reliability and lead-time, while reducing the possibilities of false warnings. For this purpose, the minimum and mean threshold values of both measures are derived for the primary and secondary warnings, on the basis of results attained under wide variations in vehicle design parameters and maneuvers. An algorithm for the two-stage audio/visual warning is finally proposed to predictively warn the driver of an articulated vehicle of an impending rollover.

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NOMENCLATURE

Symbol	Description
a_{uj}	lateral acceleration of axle j
a_y	vehicle lateral acceleration
a_{y1}	tractor lateral acceleration
a_{y2}	semitrailer lateral acceleration
a_y^*	static rollover threshold
A_i	dual tire spacing
ABS	anti-lock braking system
COV	coefficient of variation
DRT	dynamic rollover threshold
DWS	differential wheel slip
EBS	electronic braking system
ELA	effective lateral acceleration
F	frequency of the steering input
F_{ij}	force of the suspension springs (i=1,2,3) and j=1,2 (left or right suspension spring)
$FL_{i/j}$	vertical tire forces on the left track tires of i/j^{th} axle
F_{Ri}	lateral force transmitted between the sprung and unsprung masses of axle i
$FR_{i/j}$	vertical tire forces on the right track tires of i/j^{th} axle
F_{sj}	suspension force acting at right angle to the Y_{uj} axis of the j^{th} axle
F_{yij}	lateral tire forces on the j^{th} tire of the i^{th} axle

F_{zij}	vertical tire forces on the j^{th} tire of the i^{th} axle
h	cg height of the vehicle
h_{sj}	sprung mass cg height of the j^{th} unit
h_{ui}	unsprung mass cg height the i^{th} axle
I_{xxsi}	mass moment of inertias about the X_{si} direction
I_{xxuj}	unsprung mass moment of inertia respectively for axle j
I_{yysi}	mass moment of inertias about the Y_{si} direction
I_{zzsi}	mass moment of inertias about the Z_{si} direction
K_{ai}	auxiliary roll stiffness of axle i
K_{ij}	equivalent linear spring constant
$KOVT_{ij}$	linear overturning stiffness of the j^{th} tire on the i^{th} axle
K_{TF}	torsional stiffness of the tractor frame
KT_{ij}	vertical stiffness of the j^{th} tire on the i^{th} axle
K_{T5}	torsional stiffness of the fifth wheel
KYT_{ij}	spring rates of the j^{th} tire of axle i
LTR	load transfer ratio
l_{si}	lateral velocity of the sprung mass of the i^{th} unit
M_i	roll moment transmitted through different roll sections owing to compliance of tractor frame and hitch mechanisms
m_{sj}	sprung mass of the unit j
m_{ui}	unsprung mass of the i^{th} axle
m_{uj}	unsprung mass of axle j
n_s	number of sprung units of the combination

n_u	total number of axles
NRSSM	normalized roll-response of the semitrailer sprung mass
p_{si}	pitch velocity of the sprung mass *i*
R_i	effective tire radius
RAR	rearward amplification ratio
RAR_{φ}	rearward amplification in roll
r_{si}	roll velocity of the sprung mass *i*
RSF	roll safety factor
SRT	static rollover threshold
SSF	static safety factor
s_i	half suspension lateral spread
T	half trackwidth of the vehicle
T_i	half track width of axle *i*
TTR	time to rollover
$URAR_1$	unsprung mass roll amplification ratio incorporating first axle roll angle response
$URAR_2$	unsprung mass roll amplification ratio incorporating tractor rear axle roll angle response
u_{tirejk}	forward velocity of the *k*-th tire on the *j*-th axle
v_{axlej}	lateral velocity of axle *j*
v_{si}	vertical velocity of the sprung mass of the i^{th} unit
W	weight of the vehicle
W_{AXLi}	normal load acting on the vehicle's i^{th} axle

W_i	weight on the i^{th} axle
W_{si}	sprung weight of the i^{th} unit
W_{VFS}	vertical shear force acting through the tractor frame
W_5	load on the fifth wheel
y_{si}	yaw velocity of the sprung mass i
z_{FRi}	distance from the tractor's front and rear suspension roll centers to the acting point of the lateral force due to W_{VFS}
z_{Ri}	distance between suspension roll center and sprung mass cg
z_{5i}	distance from the tractor's rear and trailer's suspension roll centers to the point of action of the lateral force due to W_5
z_{ui}	vertical motion of the axle i along the $\vec{z}_{ui}$ axis
α_{jk}	side-slip angle for the k-th tire on the j-th axle
β_{si}	euler pitch angle of sprung mass I
∂_i	angle made by the wheel plane with respect to the longitudinal axis of the sprung mass co-ordinate system
δ_{sw}	steering wheel angle
$\delta_{amplitude}$	amplitude of the steering angle
$\Delta\phi_{ST}$	increment in roll angle of the semitrailer sprung mass
φ	roll angle response of the vehicle
φ_i	lift-off angle of the i^{th} axle
φ_l	lift-off angle of the vehicle
ϕ_{s1}	tractor sprung mass roll angle
φ_{s2}	sprung mass roll angle response of the semitrailer

ϕ_{uui}	roll response of the i^{th} axle
Φ_{u1}	rearward roll amplification property of the lead unit
$\Phi_{u5/1}$	$URAR_1$
$\Phi_{u5/2}$	$URAR_2$
λ_{s2}	*NRSSM*
ψ_{si}	Euler angles in yaw for the i^{th} unit sprung mass
θ_{si}	Euler angles in pitch for the i^{th} unit sprung mass
φ_{si}	Euler angles in roll for the i^{th} unit sprung mass

CHAPTER 1

LITERATURE REVIEW AND SCOPE OF THE DISSERTATION

1.1 INTRODUCTION

Rollover of heavy commercial vehicles is a critical issue that involves the potential for loss of human lives and damage to property worth millions of dollars. A vehicle reaches a roll instability, whenever the overturning moment caused by the centrifugal forces associated with the directional movement of the vehicle, exceeds the net stabilizing moment. High center of gravity of heavy vehicles, specially when laden, yields lower roll stability limit, which is further related to the dynamic characteristics of the articulated vehicle and the non-linearities associated with the suspension and tires. Roll instability at low speeds is caused during steady turning maneuvers, when the centrifugal force acting on the vehicle exceeds the rollover threshold of the vehicle. Rollover at high speeds is caused by rapid steering inputs, when the driver is required to make an emergency evasive maneuver or change lanes rapidly at higher speeds.

Nearly sixty percent of the accidents involving heavy commercial vehicles reported in the US have been are associated with rollover [1]. Another study showed that nine percent of multiple vehicle accidents involving heavy freight vehicles resulted in rollover of the vehicle while one third of all the accidents involving single heavy freight vehicles resulted in the rollover of the vehicle [2]. Winkler et al. [3] reported that more than one tenth of fatal truck and bus accidents and almost 6 out of every ten accidents in which the truck driver was killed between 1992 and 1996 in the United States, was caused by rollover of the vehicle. Rakheja et al, [4] reported that rollover was an observed cause in forty five percent of all the accidents involving transportation of dangerous goods and forty percent of the accidents involving tanker vehicles. Large trucks accounted for 8 percent of the vehicles in fatal crashes. Eighteen thousand large trucks were involved in rollover related accidents, which accounted for 4.2 percent of all vehicles involved in rollover related crashes. Out of 18,000 trucks involved in rollover accidents 11,000 were combination trucks. Another study has reported that 436 out of 622 fatal crashes attributed to vehicle rollover in 2001 were associated with articulated vehicles [5]. The reported accident statistics clearly reveal that there exists a need for a comprehensive rollover prevention device, whether passive or active.

Some concepts in roll dynamics monitoring for generating an early warning for the driver have been proposed [6,7,8,9,10,11]. The proposed concepts however involve complex challenges in realizing a reliable monitoring device, namely the identification of key response variables related to onset of rollover, measurement and processing of the variables to be monitored and development of a reliable warning algorithm with minimal potential for false warnings. In articulated vehicles, the driver's action is limited to braking, acceleration and steering, and a combination of these. Studies on dynamics of heavy trucks indicate that the speeds considered safe by the drivers during a steady turn are actually close to the rollover threshold of the vehicle [6].

The perception and control performance of the driver and dynamic responses of the vehicle under wide variations in operating conditions form the primary challenges for development of a reliable monitor that could be seamlessly integrated within the system with an effective driver-vehicle interface for preventing potential rollover induced accidents. It was found that an advance warning of 0.5 seconds could prevent thirty to sixty percent of the accidents [7]. Such a system should sense in advance a risk of rollover and warn the driver so that the driver can take cognitive and corrective action in a timely manner. Once the warning signal is delivered the possibility of averting a rollover is greatly dependent on the reaction time, which is the time taken by the driver to perceive and respond to an emergency situation.

The increasing usage of electronic sensors, actuators, warning devices and high speed processors have lead to the seamless integration of various vehicle safety systems. The last two decades saw a large number of open loop and closed-loop control strategies proposed in the area of rollover warning and control systems [6,7,8,10,13,14]. The reliability of such systems has not been explored under the wide range of operating conditions, such as vehicle load, speed and maneuver. The usage of either active or passive warning and control system as well as open and closed loop control strategies hinge on the effective sensing of the rollover instability. Although considerable efforts have been made to identify response measures that relate to onset of a potential rollover, a generally applicable measure has not yet been identified. Furthermore, it is vital to establish comprehensive performance measures to assess the reliability and effectiveness of rollover detection and warning generation strategies. The performance measures must include sensitivity of the rollover metric to variations in load and speed conditions, ease of measurement of the rollover metric and its lead-time performance.

This dissertation research is directed towards a comparative analysis of various existing measures of rollover instability in terms of performance indicators considering variations in various vehicle design and operating parameters, to assess their relative reliability for application in a rollover prevention device. A set of performance measures is formulated to assess the reliability and lead-time of the rollover metrics. An alternate rollover metric based upon the roll deflections of the sprung and unsprung masses of the articulated units is proposed and analyzed. The effectiveness of the proposed measure is demonstrated through comprehensive sensitivity analyses. A warning and control strategy is then proposed and analyzed for implementation in an early warning rollover prevention system.

1.2 REVIEW OF RELEVANT LITERATURE

Prevention of a commercial vehicle rollover through detection and warning of impending rollover involves three systematic stages. The detection of the onset of a rollover forms the vital first stage, which requires identification and formulation of rollover metric that remains relatively insensitive to variations in design and operating conditions of the vehicle. The second stage necessitates thorough knowledge and analysis of the rollover metric in view of its reliability of prediction, measurability and lead-time performance to provide sufficient reaction time for the human driver. The final stage involves a design of the warning algorithm including selection of readily available sensors, signal processing modules and the form of warning that is easily perceived by the driver. This dissertation research thus requires thorough understanding of the vehicle directional dynamics, methods of analysis, range of vehicle parameters, measures of rollover, sensors, warning devices and human factor issues. The reported relevant studies are thus reviewed to build the needed background and the scope of this dissertation research. The highlights of these studies grouped within different subjects are summarized in the following subsections.

1.2.1 Vehicle Dynamics and Rollover Analyses

Roll dynamics of heavy vehicles have been extensively studied through development and analyses of an array of analytical models of varying complexities. Vlk [15] reviewed a large number of these studies and concluded that little importance has been placed on comparison of various simulation models and far more emphasis has been given to the development of various computer models for analyses of lateral dynamics of articulated vehicles. Jindra [16], Ellis [17], Vlk[18] and Nordstrom [19] developed linear and non-linear yaw plane models involving linear component models to study the lateral stability limits of articulated vehicles. Mallikarjunarao and Fancher [20] developed an analytical yaw plane model for lateral stability analysis of articulated vehicles with multiple axles, using eigenvalue analysis. The yaw plane models are incapable of assessing the roll and pitch dynamics of the vehicle. The roll stability of a vehicle is generally defined in terms of its rollover threshold, which is characterized by the maximum value of lateral acceleration, which the vehicle can withstand without experiencing a divergent roll response under a steady-turning manuever. Several roll plane models have been developed to analyze the static as well as dynamic rollover characteristics of heavy vehicles [6,13,21,22,23].

A number of more comprehensive non-linear models with large number of degrees-of-freedom evolved in the seventies. A more comprehensive model of an articulated vehicle was proposed by Strandberg [24]. Verma and Gillespie [21] presented a non-linear model for studying the roll motions of trucks and tractor–trailers. It was shown that the roll dynamics of such vehicles was related to the time history of the applied lateral force, and rollover can occur at lateral acceleration levels less than the static acceleration levels due to the roll resonance of the combination. On the basis of a simplified static force model, Miller and Barter [12] concluded that the rollover occurred when the summation of all forces acting on the semitrailer center of gravity (*cg*) falls outside the trailers outer wheels. The lower limits of roll stability could occur under transient steering maneuvers performed at high speeds. Winkler et al. [25] proposed that the rollover phenomenon could be conveniently divided into factors influencing the lateral translations of the sprung masses of the vehicle units and the factors influencing the center of gravity height to the track width ratio.

Malikarjunarao [22] developed a comprehensive static roll model, for deriving the rollover threshold of articulated vehicles during steady turning maneuvers. The static roll model was further validated using the data from tilt table tests. This model grouped the vehicle axles with similar suspension properties so as to represent the vehicle as a collection of 3 composite axles; the tractor front axle, an equivalent rear axle and an equivalent semi trailer axle. The sprung mass of the tractor was represented as a combination of two sprung masses coupled through torsional stiffness of the tractor frame. The sprung mass of the semi-trailer was coupled to the tractor mass by the torsional stiffness of the fifth wheel and the semi-trailer structure. The tires were represented as linear springs. The static equilibrium equations were solved for infinitesimal increments in the roll angle of the trailer sprung mass. Piché [26] concluded that the onset of roll instability could be directly related to the rearmost axle's roll angle, cornering forces and drive axle's tire sideslip angles.

A large number of dynamic models of articulated vehicles have been developed to analyze their roll dynamic performance under varying conditions. Gillespie and Verma

[23] analyzed the dynamic roll response of heavy tankers using single composite axle roll plane model. Rakheja et al. [13] proposed a dynamic model for multi-unit and multi-axle vehicle, to study the maneuver induced roll stability limits. Gillespie and McAdam [27] developed a three dimensional directional dynamic model of an articulated vehicle incorporating 52 degrees-of-freedom, referred to as Yaw/Roll model, for analyzing the dynamics of articulated vehicles engaged in constant speed directional maneuvers. A comprehensive three dimensional vehicle model called Phase IV, incorporating 71 degrees-of-freedom, was developed by UMTRI [28] to simulate the braking and steering dynamics of trucks, tractor-trailers, double and triple combinations.

Liu [6] modified the constant velocity Yaw/Roll model by shifting the sprung mass roll center to the outboard tire-road contact point when the inboard tire lost contact with the road and defined the dynamic rollover threshold of a heavy vehicle as the level of Effective Lateral Acceleration (*ELA*), corresponding to the vehicle's relative roll instability. The relative roll instability was considered to occur when wheels of the tractor rear and trailer axles lifted off the road in case of a tractor semi-trailer. A neural network model was suggested by Ghazizadeh [29], which uses the results of the yaw/roll plane model to train the neural network. Yang [30] studied the effectiveness of the trained recurrent neural network with two-step time delay by comparing the output of the neural network with results attained through analysis of the yaw/roll model. The study showed reasonably good agreements between the two models in terms of the yaw rate, roll angle and lateral acceleration response characteristics. Van Deusen [31] analyzed a linear thirty-eight degrees-of-freedom model in the frequency domain using transfer function concepts. The comparison of the theoretical and experimental data showed good agreements. Ruhl [32] investigated a roll plane model by combining the semitrailer roll plane model with a static yaw model of the articulation angle between the tractor and the trailer incorporating nonlinear articulation stiffness.

The roll analysis models found in the literature assume considerably small magnitudes of lateral forces developed at the inside tires in comparison with those developed at the tires on the outside at lateral acceleration values close to the rollover limit. The suspension springs are constrained to translate along the direction normal to the plane of the unsprung mass. The vertical and lateral stiffnesses of the tires are assumed to be constant. It is further assumed that the relative motions between the sprung and unsprung masses occurs about the roll center, located at a fixed vertical distance from the sprung mass *cg*. Roll angles of the sprung and unsprung masses are assumed to be very small. Dynamic roll responses of different units of an articulated heavy vehicle under different directional maneuvers are known to differ in both amplitude and phase [20]. The static rollover threshold may not thus describe the dynamic rollover situations. In dynamic analysis, five degrees-of-freedom are assigned to each sprung rigid body, namely: lateral, roll, yaw, pitch and vertical, while the axles are considered to have two degrees-of-freedom (roll and bounce). A rolling vehicle possesses kinetic energy, which deteriorates roll stability, invalidating the static rollover threshold analysis. Roll stability analysis with an energy approach was also highlighted by comparing the present roll energy with that required for rollover [72]. Both the potential energy required to bring the vehicle to a roll instability and the kinetic energy present in the system are expressed as a function of lateral acceleration. McAdam [33], and El Gindy and Ghazizadeh [29] utilized neural networks to model the dynamic behavior of heavy vehicles. Yang [30] used a neural network model to analyze the dynamic roll instability and suggested that the neural network model slightly overestimates the response under a ramp-step steer input. A comparative analysis of the different analytical and computer simulation models was presented by El Gindy and Wong [34] and it was inferred that different models presented different attributes and the choice of a relevant simulation model was dependent on the desired performance of the analysis.

1.2.2 Reported Rollover Metrics

Owing to the large variations in the operating conditions, the assessment of reliability of a detection algorithm is also a formidable task. The reported studies have explored various performance measures, which include the lead-time for the driver to undertake an action, reliability, measurability and minimization of false warnings. The majority of the studies consider rollover metrics based on different measures of rollover, which may differ considerably in terms of reliability, lead time generated, measurability and requirements of sensors. The roll instability of a heavy vehicle is primarily caused by the maneuver induced lateral acceleration and high centre of gravity of the sprung mass. Relative roll instability concerns the maximum limit in terms of lateral acceleration, which the vehicle can sustain without rollover in a steady turning maneuver. Thus it can be inferred that the vehicle can no longer remain stable under the action of a constant level of lateral perturbation in case of relative roll instability. The absolute roll instability on the other hand considers the maximum roll angle of the vehicle at zero tolerance as the measure of instability [6]. Early warning and control requires sufficient lead time to bring the vehicle back to stability and hence absolute roll instability cannot be considered for timely prediction and control.

Dunwoody [35] used the lateral acceleration of the trailer and the relative roll angle of the tractor-trailer as a measure of rollover. Ervin [36] suggested that the static roll instability of a vehicle could be described by its rollover threshold. The rollover threshold of a rigidly suspended vehicle is expressed as the ratio of half the track width to the cg height, when the vehicle is represented as a single degree-of-freedom system in roll. The equations of static equilibrium in roll moment reveal that the maximum stabilizing moment attained during rollover is reached when the wheel lift off occurs or when the load on the inner wheel approaches zero. The corresponding maximum lateral acceleration that the vehicle can sustain without suffering a divergent roll response is termed as Static Rollover Threshold (*SRT*). The *SRT* of a vehicle can be obtained through road and tilt-table tests and analytical methods. Winkler [37] concluded that a small increase in the static rollover threshold by 0.1g in the 0.4-0.7 g range could significantly reduce the frequency of rollover accidents.

An articulated vehicle in motion possesses kinetic energy and this fact brings into focus the issue of deteriorating roll stability while analyzing a rollover occurrence. Dynamic Rollover Threshold (*DRT*) has thus been proposed to describe the rollover propensity of a vehicle, while taking the dynamic characteristics into consideration. Dahlberg [38] suggested that the *DRT* is the worst-case measure of roll instability, and defined *DRT* as the magnitude of lateral acceleration that may cause the vehicle to rollover under transient directional maneuvers. Simulation results show that the vehicle may rollover under application of a lateral step input equal to the dynamic rollover threshold but lower than the *SRT* [38]. The *DRT*, however, is sensitive to the rate and magnitude of steering, and may yield an over or under estimation of the rollover threshold, which is mostly attributed to highly non-linear behavior of the vehicle components in the dynamic state. On the basis of simulation results under sinusoidal maneuvers in the 0.25 to 0.50Hz range, it has been suggested that the *DRT* value for all practical purposes can be considered almost equal to the *SRT* [6]. The *DRT* measure may thus also correlate with the relative rollover condition and could be employed to derive the relative rollover indicator under directional maneuvers.

Erwin [36] proposed a measure of dynamic roll instability termed as, Rearward Amplification Ratio (*RAR*), which is defined as the ratio of the peak lateral acceleration response of the trailer to that of the tractor, when the vehicle undergoes high speed maneuvers. In such a case larger lateral and roll response is obtained from the rearmost unit as compared to the lead unit. It has been suggested that the *RAR* of a tractor-trailer combination under a path change maneuver must not exceed 2.2 [32]. The extreme sensitivity of this measure to variations in design, configuration and operating conditions, however, inhibits the use of *RAR* as a reliable measure [6].

El Gindy [39] proposed the Load Transfer Ratio (*LTR*) as a measure to assess dynamic roll stability limits of heavy vehicles in North America. The measure is evaluated under a path change maneuver performed at 100 km/h which causes a lateral acceleration response of 0.15 g at the cg of the trailer. A target value of this parameter was recommended as 0.6 [26]. Preston–Thomas and Woodrooffe [40] concluded that *LTR* could serve as a reliable indicator for impending rollover. The *LTR* approaches a unity value when all the wheels of the vehicle on a single track lose road contact. The

LTR, however, does not relate to the relative rollover condition, which happens when one or more axles experience lift off depending on the vehicle configuration. A unique value of overall *LTR* with respect to the relative rollover condition does not exist [6,13]. Relative rollover condition of a tractor-semitrailer combination is reached, when the wheels on the tractor rear axles and trailer rear axles lift off the road [6]. An alternate measure that directly relates to the relative rollover condition and utilizes the concept of *LTR* was proposed by Liu [6]. This measure, termed Roll Safety Factor (*RSF*) was shown to be a more generalized measure of the relative rollover condition and thus the onset of roll instability as the *RSF* approaches a unity value.

Klein [41] proposed another measure of relative rollover instability called the Static Safety Factor (*SSF*), which closely correlates with actual rollover frequencies of road vehicles. This factor provides a crude estimate of the rollover propensity of heavy vehicles by ignoring the compliance due to suspension and tires such that the vehicle roll angle remains close to zero. Moreover, this simplified measure does not consider the contributions due to multi-axle vehicles. A compliance factor of 0.72 was proposed by Piché [26] to account for the reduction in rollover threshold of the tractor-semitrailer combinations due to compliance of different suspensions, tires and springs. Although the diminishing tire loads offer a direct assessment of a rollover condition, the measurement of loads in a vehicle in motion is difficult and hence cannot be considered feasible as concluded by Rakheja et al. [42]. Liu [6] proposed the dynamic rollover threshold of articulated vehicles as a credible measure on the basis of lateral acceleration and relative rollover criterion. It was concluded that *RSF* can serve as the most reliable indicator irrespective of the vehicle configurations but its measurement requires dynamic vertical load information of tractor rear and trailer axles, which makes its measurability an area of concern. Dunwoody [35] proposed that a combination of lateral acceleration response of the trailer and the relative roll angle of the tractor-trailer combination could serve as an effective measure of impending rollover.

The total roll moment caused by the centrifugal forces acting on the sprung and unsprung masses can be expressed in terms of Effective Lateral Acceleration (*ELA*)[6]. *ELA* was reported as a dynamic parameter that is dependent on load transfer and track width of each axle, the center of gravity heights of the sprung and unsprung masses, and the lateral displacement of sprung mass, which in turn is dependent on the suspension properties. *ELA* can also be represented as a moment-weighted average of lateral acceleration response characteristics of the vehicle units. In the case of articulated vehicles, the dynamic roll response of different mass units may differ significantly in terms of phase and amplitude. The *ELA* is approximately equal to the lateral acceleration of the trailer sprung mass in an articulated vehicle, when the effect of unsprung mass acceleration is considered negligible.

Chen and Peng [14] proposed a Time To Rollover (*TTR*) metric as a basis to assess rollover threat in articulated vehicles. A neural network was proposed to be trained to deliver *TTR* count down for a wide range of maneuvers. However the accuracy of the metric under various speeds and driving conditions and the requirements of a faster real time model were cited as the two conflicting requirements for implementation of the metric. Palcovics [8] suggested a method, which uses the electronic braking system platform of the vehicle and is based on difference in the wheel slips as the measure of roll instability. By stimulating the wheels of an articulated vehicle in very short intervals of time, through the application of a test pulse using the braking system, the difference in slip between the left and right wheels of the vehicle can be detected both qualitatively and quantitatively. Thus the proximity of the vehicle to roll instability can be determined if either the differential wheel slip or the longitudinal slip ratio is greater than a preset value corresponding to the rollover stability.

The roll dynamics of an articulated vehicle is a complex function of many design and operating factors, such as weights and dimensions, suspension and tire properties, loading practices, maneuvers, speed and road conditions. The sensitivity of a predictor to variations in design and operating conditions thus directly relates to the feasibility of an early warning monitor and the associated risks. Mikulcik [43] derived the boundaries of stability of a light vehicle-trailer system using the Routh's criterion. Troger and Zeman [44] concluded that the *cg* location of a semi-trailer relative to the articulation point is one of the most important parameters for the stability of articulated vehicles. Nalcez et al. [45] proposed that the heavy vehicles experience instability at lateral acceleration levels of 0.3-0.4 g.

The rollover threshold of heavy vehicles is a function of vehicle geometry, suspension properties and tire properties. On the basis of the dynamic responses of a tractor trailer combination, Schmid [46] concluded that the location of dolly cg in front of the trailer front axle, understeering of the truck, large wheelbase of the truck, high cornering stiffness have positive influences on the directional stability of the tractor-trailer combination. Fancher and Mathew [47] compared the simulation results of the roll dynamics of a wide range of heavy vehicles on the basis of *RAR*, *SRT* and handling performance. The study found that roll stiffness, yaw stiffness and location of vehicle couplings strongly effect the interactions between adjacent units of an articulated vehicle. The pintle hitch coupling decouples the roll motions of adjacent units and the roll stability of each unit can be evaluated separately, where the roll motion of units linked by a fifth wheel are strongly roll coupled. Pintle hitch coupling yields relatively higher values of rearward amplification ratio of the combination.

Blow et al. [48] performed an exhaustive simulation study of more than five hundred heavy vehicle combinations in view of the rearward amplification ratio and steady state roll stability. The study concluded that these two parametric indicators are most sensitive to vehicle weights, tire properties and vehicle coupling design. Kusters [49] suggested that the sprung mass, unsprung mass, the roll center height, height of unsprung mass cg, sprung mass cg heights, wheel tread and roll stiffness form the basic parameters for such analyses, while the suspension and tire stiffness, and roll stabilizer as important parameters. The semi-trailer cg height, track width of the vehicle, choice of suspension, auxiliary roll stiffness, roll centre height have all been found to greatly affect the rollover threshold of articulated vehicles [6,22].

1.2.3 Concepts in Early Warning Devices

The proposed control and warning strategies are based upon detection of the prevailing conditions and their comparisons with a predefined value of the rollover threshold. The identification of a preset threshold value, however, remains the primary challenge for the control, whether passive or active. Preston-Thomas and Woodrooffe [39] conducted a study of the feasibility of rollover warning device and concluded that forty two percent of more than two thousand vehicle rollovers involving heavy commercial vehicles could be avoided by systems that could warn the driver of the onset of a risky situation. Sampson et al. [50] suggested that active stability systems avoid rollovers more efficiently by the application of active braking at one or more wheels, than passive systems enabling dynamic control more precisely by a driver. However, the prevention of rollover through active control requires high forces and inertia, the passive control strategies are thus considered as potential methods for rollover prevention.

An active control strategy includes actions like reducing vehicle speed, lessening of lateral force components, and action on the axles by hydraulic actuators and differential braking. The use of active suspension systems holds a possibility of active roll control, which needs to be further explored. Low power and low priced hydraulic actuators can be used to actively control rollover following an effective detection of a rollover threat. One of the earliest road vehicles with active roll control was a three wheeled motorcycle based machine designed and built by MIT in 1968 [51]. The prototype used a single feedback control scheme based on a tuned pendulum to measure body roll angle and a hydraulic servo actuator to apply roll moment between the body and the rear axle. The steady state performance was judged to be good, while the transient performance was limited due to lack of sensors with sufficiently fast response. The large sized heavy mechanical roll angle sensor used in the study introduced unwanted lag into the closed loop system dynamics.

Karnopp [52] investigated the potential of using an active control of load levelers to regulate low frequency automobile body motions. It was concluded that proportional and derivative control of load leveler deflection could be used to reduce the roll and pitch motions under the influence of cornering and braking, respectively. Sharp and Hassan [53] proposed a system based on rotary hydraulic actuators incorporated within the anti-roll bars, to provide active restoring moment using the lateral acceleration feedback. Mizuno et al. [54] designed an active roll control system using the feedback from the relative roll angle and the roll rate between the sprung and unsprung masses. The system could regulate the relative roll angle but regulation with respect to the ground was not possible because of the tire compliance.

Kushahara et al. [55] proposed an active control system comprising anti-roll bars and double acting hydraulic actuators linking the front and rear end linked to the vehicle frame to control the vehicle roll angle. Wheel speed and steering angle sensors were used to estimate the lateral acceleration of the vehicle, which served as a feedback for the proportional feedforward controller to produce actuator force demand signals. The controller could switch between several modes for different loading conditions by obtaining the static suspension deflection. The system reduced body roll by sixty seven percent in steady state cornering and high-speed lane change maneuvers.

Dunwoody and Froese [35] investigated the advantages of using active roll control systems to increase the steady state roll stability of a tractor-trailer by using simulations. The input signal was the lateral acceleration signal from an accelerometer mounted on a trailer. The authors stated that such a system could increase the rollover threshold by twenty to thirty percent. Sampson et al. [50] proposed a modified hybrid suspension system for semi-trailers, comprising active U-shaped anti-roll bars, controlled by hydraulic actuators, to control the roll motion of each axle. The maximum steady state actuator force required to hold the mass at zero roll angle during a 0.5 g steady state run was estimated as 110 kN. It was proposed that fast transient response of the active control system could be realized using lateral acceleration feedback. Active control and stabilizers generally require high force, which tend to increase the weight and cost.

Eisele and Peng [11] proposed a Vehicle Dynamics Control (VDC) system that utilizes differential braking to affect vehicle response. Using the simulation results of a human in the loop linear model, it was shown that the vehicle accurately follows the drivers intended path with enhanced roll stability. Wielenga [51] proposed the use of acceleration sensors in combination with suspension deflection sensors to determine the rollover risk severity. The front brakes are activated by the system to lower vehicle speed but this induces understeer, which could conflict with the driver's intention to stay on the road. The system proposed by Palcovics [8], stimulates the wheels through a brake system by the application of a test pulse in the brake system when the calculated lateral acceleration exceeds the threshold value. The system is appealing because it does not require any additional sensors, while a reliable preset value still needs to be identified. A closed-loop controller was proposed to undertake a corrective action in the form of engine braking. Holler and McNamara [56] proposed a trailer based rollover prevention system utilizing EBS and ABS systems as proposed by Palcovics [8]. The study however did not address the lead-time issue that may cause delays associated with the filling up of brake chambers through the traction valve, relay valve and the wheel end modulator line.

The impairment of the driver to sense lateral acceleration response raises the need for some form of warning device for the driver when the lateral acceleration or other potential rollover indicator's response approaches a threshold value. An online detection of impending roll stability at the trailer axle can provide early warning to the driver and ensure a timely corrective action. Sparks and Berthlot [9] categorized rollover accidents as preventable, potentially preventable and non-preventable, and it was inferred that some of the accidents could be avoided if a warning device was utilized. Moreover, a potential instability mostly initiates at the rearmost axle of the vehicle, which is not readily perceived by the driver. A substantial delay in the driver's perception of the risk and the reaction delay most likely constitute the main cause for the drivers failure to arrest a rollover threat and undertake a preventive action. The delivery of information of a potential risk with sufficient perception and reaction time thus forms the main challenge.

Palcovics and Fries [57] classified chassis electronic systems in three classes: chassis installed systems that could assist the driver in undertaking corrective actions while the driver remains in the loop; autonomous systems that override the driver; and the driver triggered systems involving direct actuation from the driver through advisory information supplied to the driver. The systems within the last category are considered desirable as they allow the driver to remain in the control loop.

Sensing of a selected rollover metric is an integral part of a warning and control system for preventing rollover induced accidents. The sensors must be precise as the variations in some metrics could be small, rugged to operate in a vehicular environment and preferably possess integrated conditioning and signal filters. The modern technologies use angular rate sensors to measure the vehicles angular velocity insensitive to the linear accelerations in an automotive environment. These inertial sensors are piezoelectric crystals and variations of micro machined structures [54,58]. The sensing algorithm processes the sensor measurements and in turn estimates the vehicle dynamic states. An important consideration is that the sensing system must function properly under a wide range of severe conditions. Moreover sensitivity to the linear acceleration is critical to ensure robust sensing of roll rates and deflections. Steiner et al. [59] discussed some elementary rollover measuring sensors like spirit level detector, tilt cone sensor and low range acceleration sensors. These sensors, being gravity based, exhibit larger response time and greater sensitivity to linear accelerations. Position and rate gyroscopes have also been proposed for measurement of angular positions and velocities.

Nordstrom [60] suggested the use of wheel load, steering wheel angle, tire pressure and acceleration sensors for detecting the rollover propensity. Load sensors in each wheel help to gauge the available overturning stability of the vehicle. With the aid of navigational systems with road curvature information, the driver can be advised about the speed with which he should pass the next curve. Wallner and Schaffman [61] proposed a rollover sensing module that can predict in advance the angular position of the vehicle prior to reaching a critical angular threshold. The module uses three orthogonally oriented accelerometers and three angular rate sensors to enhance the reliability of the measure.

Considerable advances have been made to realize the usage of glass and silicon micro machining techniques for fabrication of micromachined sensors for angular measurements. Such sensors are considered to be rugged as silicon, quartz and other monocrystalline materials possess long life and superior shock survivability [58,59,62]. The strain gauge hubs suggested by Rupp [63] and the rotating wheel dynamometer systems [54] that use quartz force sensors, could be applied to monitor individual wheel loads. Such sensors, however, are very expensive. Ehlbeck et al. [10] suggested that a roll advisory and control system should utilize the ABS platform to house and operate the necessary electronics and software. The authors further proposed the utilization of existing vehicle database and exclusion of any external sensor to maximize reliability. The device allows the driver to monitor the driving performance through access to cumulative rollover risk, while the warning is issued by visual means. The system consists of three independent elements. A roll stability advisor, roll stability control and a hard braking event detector. The roll stability advisor and the hard braking event detector components provide information only while the roll stability control identifies critical rollover situation. The system uses measured lateral acceleration to detect the onset of a roll instability, which is known to be sensitive to variations in vehicle design and operating variables.

Generally three causes lead to the rollover of heavy commercial vehicles; sudden course deviation often in combination with braking and high initial speed; high speed cornering; and excessive load transfer. The first two relate to the driver's action and the lateral stability of the vehicle. The stability of an articulated vehicle depends on both the vehicle and its driver. The driver controls the motion of the vehicle by deciding on the path in response to a perception of risk through corrective maneuvering, while the vehicle provides the disturbance overcoming capabilities, higher roll stability limits and minimum driver effort based on superior design [65]. Rothengatter [66] suggested that the drivers tend to have specific preferences for the type and modality of the information provided. A warning system for the driver, must be designed to provide warning through suitable modes so as to accommodate the reaction time of the driver.

1.3 SCOPE AND OBJECTIVES OF THE PRESENT RESEARCH

From the review of the relevant literature, it is evident that extensive efforts have been made on the subject of roll dynamics of heavy commercial vehicles including the enhancement of the roll stability limits. A number of concepts in open-loop and closed-loop rollover control have emerged during the past two decades. The suitability of the open loop concepts in terms of the prediction reliability, ease of measurement and lead-time, however, has not been thoroughly investigated. The reported concepts in open-loop rollover control thus need to be critically reviewed in terms of their rollover detection methodology, performance and reliability, ease of measurement, lead-time for warning generation, etc. The identification of measurable indicators of a potential rollover, however, is a formidable task due to relatively large variations in the operating conditions of commercial vehicles

The assessment of a potential rollover predictor for articulated vehicles would necessitate a thorough investigation of its sensitivity to variations in vehicle design and operating conditions and its lead-time performance. The risk of rollover related accidents can be greatly reduced by enhancing the driver's perception of the onset of a roll instability or by utilizing control systems that would act automatically, when a potential roll instability is detected. This necessitates the identification of a rollover metric that is least sensitive to operating conditions or design variations, and correlates well with the well established measures of rollover. Moreover, the metric should be easily measurable and provide reasonable lead time for the driver to undertake a timely corrective action with minimal false warnings. A comprehensive early warning and control strategy may then be formulated using the most effective metric(s) for detecting the onset of a potential roll instability and providing effective warning for the driver. The primary objectives of this dissertation research concerns with identification of rollover metric(s) with enhanced reliability and sufficient lead time, and formulation of a warning generating algorithm for preventing a potential rollover. The specific objectives are:

(1) Perform critical analyses of various reported measures of vehicle rollover to assess their ability to predict the onset of a potential roll instability in a reliable manner.

(2) Formulate performance criteria to assess relative potentials of different measures of vehicle rollover.

(3) Investigate the reliability of the reported measures to predict the onset of a rollover through extensive sensitivity analyses under wide ranges of parametric variations in the vehicle design and operating conditions.

(4) Explore alternate metric(s) that exhibit least sensitivity to variations in design and operating conditions, and yield reasonably good 'time to rollover' performance.

(5) Establish threshold values of desirable metric(s) for implementation in either single or dual-mode warning mechanisms.

(6) Propose an open-loop early warning and control strategy on the basis of threshold values of desirable metrics and assess its potential performance under variations in important design and operating variables.

1.4 ORGANIZATION OF THE THESIS

In chapter 2, the mechanics of rollover of an articulated vehicle is briefly discussed to identify the roll instability conditions related to onset of a potential roll instability. The widely used Yaw/Roll model of articulated heavy vehicles is described as a simulation tool for analyzing the roll dynamic responses of the vehicles and performing relative assessments of various rollover metrics.

In chapter 3, the concepts of open loop rollover controls are critically reviewed and a wide range of measures for the detection of impending rollover is analyzed .The measures that are related to relative roll instability criteria are identified. The Roll Safety Factor (*RSF*) is established, as a premise for identifying the relative roll instability condition. Based on the shortcomings of the existing measures and the premise for relative roll instability criterion, the required attributes of a generalized measure of impending rollover are outlined. The rollover metrics pertinent for open-loop rollover warning are shortlisted for subsequent parametric sensitivity analysis.

In chapter 4, a baseline 5-axle tractor semitrailer vehicle is described and a simulation matrix is established that encompasses a wide range of variations in vehicle design and operating conditions. Simulations are performed to assess variations in both the static and dynamic roll responses under wide variations in design and operating variables and to understand the effect of variations in design and operating parameters on the roll stability of articulated vehicles.

The simulation results are subsequently analyzed in Chapter 5 to identify desirable rollover indices. A set of criterion including the metric's sensitivity to variations in design and operating variables, lead-time and measurability is applied to evaluate the relative performance potentials of different measures. An alternate metric is proposed to reliably predict the onset of a potential rollover with least sensitivity to variations in design and operating conditions.

In chapter 6, the identified metric(s) are integrated with an already existing measure of roll instability and the performance of these metrics at various stages of roll instability is investigated to identify the level of relative roll instability in terms of *RSF*, most suitable for initiating the warning mechanism. A comprehensive warning strategy based on the threshold values of the identified rollover metrics, using a two-stage warning and control algorithm is proposed. The human factors issues for effective delivery of the warning signals to the driver are discussed.

The major highlights of the investigation and conclusions drawn are summarized in chapter 7. The scope for further investigations in roll control of heavy commercial vehicles in order to prevent rollover-induced accidents is presented.

CHAPTER 2

MECHANICS OF ROLLOVER OF ARTICULATED VEHICLES

2.1 INTRODUCTION

The rollover of articulated vehicles is a complex process due to interactions between the coupled units, while the properties of many key components, such as tire and suspension, are highly nonlinear. The roll stability limits under steady-turning could be conveniently studied in the laboratories. Although, rollover is typically a dynamic phenomenon, an in-depth investigation of the static rollover properties is required to understand the process of dynamic rollover. This study investigates rollover mechanics of a five axle tractor-semi trailer, as more than three fourth of all the commercial vehicles operating in the highways in Canada are five-axle tractor semi-trailers and are most prone to rollover instability [67]. Articulated vehicles reach roll instability condition at low speeds during cornering or during directional maneuvers at high speeds, while in the latter case, roll instability maybe reached at low levels of lateral acceleration, in comparison to static rollover threshold.

Liu [6] suggested that the rollover mechanics of articulated vehicles could be classified into two conditions of roll instability: Relative Roll Instability and Absolute Roll Instability. All possible roll instability criteria must be identified and critically examined for their effectiveness in predicting rollover. A carefully selected and widely applicable roll performance measure based on the most effective roll instability condition can be utilized for the development of an early warning and control strategy.

In order to study the mechanics of roll stability and examine the influence of parameters, it is first essential to develop or select mathematical models that can realistically simulate the roll motions of the vehicle units. For the case of static roll stability, a roll plane model is adequate to simulate the steady state responses to lateral acceleration in a steady turn. For the simulation of roll behavior under dynamic maneuvers, however, a more comprehensive model with yaw motion consideration is more appropriate.

In this chapter, the rollover mechanics of articulated vehicles is discussed along with the criteria for roll instability. Articulated vehicle models are developed and identified that can be utilized for the simulations of roll performance under both static and dynamic maneuvers. The constant velocity Yaw/Roll model is selected for the simulation of static and dynamic analysis in this dissertation is presented and discussed. Since the duration of the lateral perturbance under a dynamic maneuver plays an important roll, relative rollover condition is established as the premise for effective and predictive warning of impending rollover of an articulated vehicle. In the following chapters the model is extensively used to simulate the response of an articulated vehicle for a wide range of design parameters and operating conditions.

2.2 ROLL MECHANICS OF ARTICULATED VEHICLES

A survey revealed that of all the articulated vehicles operating in Canada, around 77% of the vehicles were tractor semi-trailers [68]. A typical five-axle tractor semitrailer combination consists of a towing unit (tractor) with one axle in the front and tandem axle at the back, and a towed unit (semitrailer) with tandem axle in the back. The two units are joined together through a coupling, commonly known as fifth wheel. The fifth wheel allows freedom in yaw motion without any constraints, while in roll, the units are coupled with a degree of lash present.

The simplest form of rollover mechanics for a vehicle with suspension can be illustrated using a lumped model where all the masses and axles are lumped as shown in Figure 2.1. When the vehicle with weight W and cg height h undertakes a steady state or transient steering maneuver, the lateral acceleration (a_y) acting on the vehicle causes a primary overturning moment ($W.h.a_y$) to act on the vehicle's cg about the ground plane thereby causing the vehicle to roll over a point on the ground plane by an angle φ, termed as roll angle. This in turn results in the center of gravity of the vehicle to be laterally displaced giving rise to further moment of lateral displacement ($W.h.\varphi$). The load transfer from the inside to outside tires causes the generation of a stabilizing force, which increases until one of the inner wheels lift off the ground. Before the wheel lift off, the moment balance leads to an equation that can be expressed as:

$$(F_{zL} - F_{ZR}).T \ - \ W.h.\varphi = W.h.a_y \tag{2.1}$$

where T is the half-trackwidth, and F_{zL} and F_{ZR} are the wheel vertical loads for the left and right wheels respectively. The vehicle experiences roll instability when the lateral acceleration reaches a level large enough to provide a primary overturning moment which exceeds the net stabilizing moment generated by load transfer, the term on the left hand side of equation (2.1).

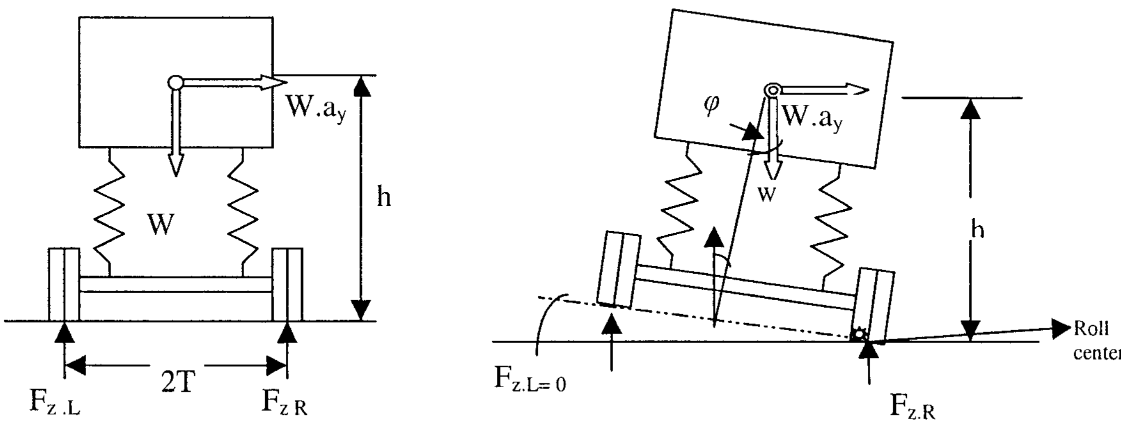


Fig. 2.1: Lumped roll plane representation of a vehicle before and after the wheel lift off

The gradual load transfer from the inner to the outer wheels leads to the load on the inner wheel approaching zero and the load on the outer wheel approaching the weight of the vehicle when the inner wheel experiences lift off. At this point the maximum value of lateral acceleration the vehicle can sustain without experiencing a divergent roll angle response is called the Static Rollover Threshold. At the point of wheel lift-off, the maximum restoring moment is obtained, which corresponds to $F_{zL}= 0$ and $F_{ZR}=W$. Hence the corresponding roll moment equation (2.1) yields:

$$W.T - W.h.\varphi_l = W.h.a_y^* \quad (2.2)$$

where a_y^* is the static rollover threshold and φ_l is the corresponding sprung mass roll angle. The Static Rollover Threshold (*SRT*) can therefore be expressed as:

$$SRT= a_y^* = (T/h) - \varphi_l \quad (2.3)$$

The roll moment and motion expressed in equation (2.1) can also be conveniently represented in graphical form as shown in Figure 2.2. A lumped mass model as presented above tends to overestimate the *SRT* value as it corresponds to lift off of a wheel that represents all axles. For a multiple axle unit combination such as tractor semitrailer, all axles do not provide same restoring moments and thus the *SRT* may be reached prior to all axles experiencing a wheel lift off.

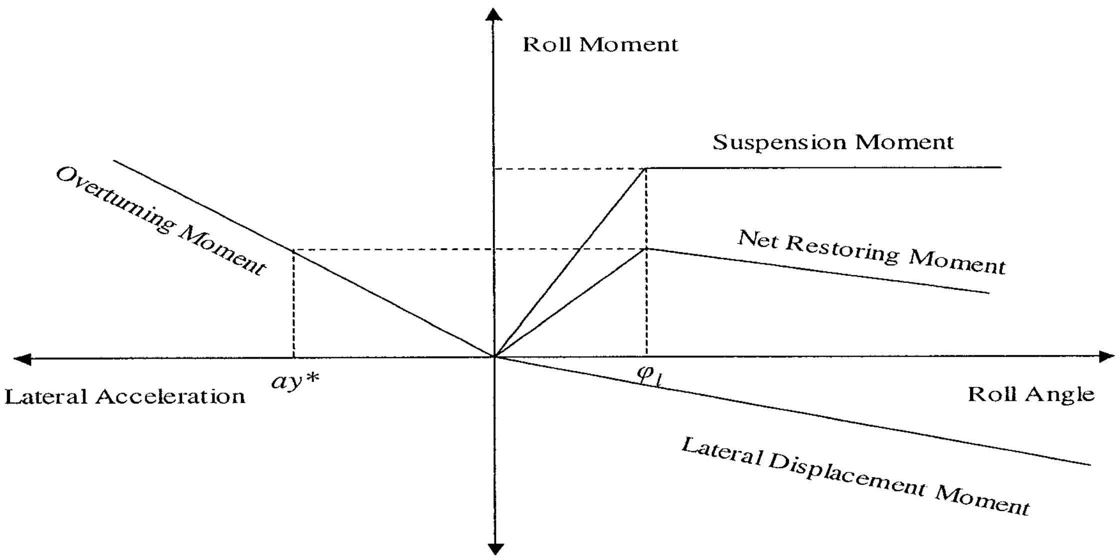


Figure 2.2: Roll moments and response of lumped mass and axle model.

Neglecting fifth wheel compliance, a tractor semitrailer combination can be expressed in a simplified manner by a set of three composite axles based upon the suspension characteristics of the tractor front, tractor rear and semi-trailer axles. The initiation of rollover in a tractor semi-trailer begins with the wheel lift off of the semi-trailer when the sprung mass roll angle is φ_3 and corresponding suspension moment is W_3T_3. This is followed by the wheel lift off at the tractor rear axle when the roll angle is φ_2 and corresponding suspension moment is W_2T_2. The tractor front axle having the softest suspension exhibits wheel lift off at the larger value of roll angle (φ_1) with corresponding moment W_1T_1. Figure 2.3 illustrates the roll moments for the three composite axles. Similar to Fig. 2.2, the diagram also represents total overturning moment $\sum W_i h_i a_y$ and lateral displacement moment $\sum W_i h_i \varphi_i$ where $i=1,2,3$.

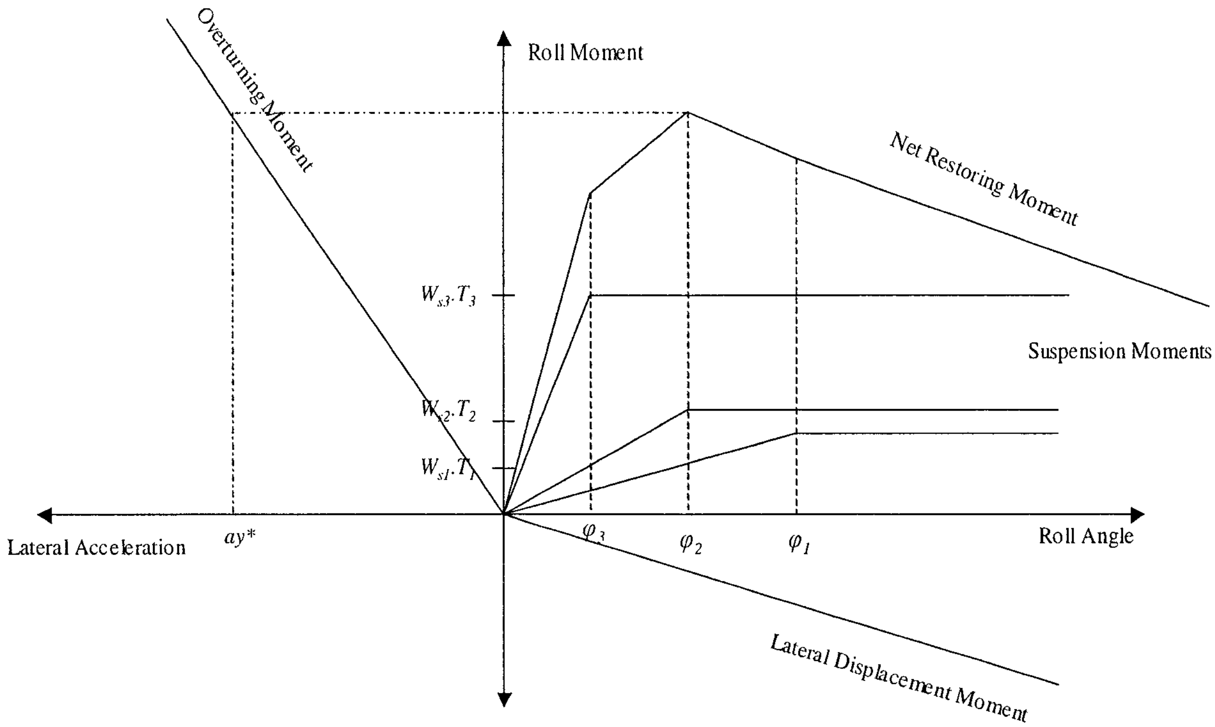


Figure 2.3: Roll moments and response of a three axle vehicle representation.

The *SRT* in this case can be obtained by equating peak net restoring moment to the overturning moment as shown in Fig. 2.3. Beyond this point the slope of the net restoring moment is negative which indicates that the tractor front wheels at this point is incapable of generating a restoring moment to negate the lateral displacement moment. The threshold in this case therefore corresponds to the lift off of the inside wheel of the tractor rear axle. Depending on the suspension properties, it is thus essential that a comparison of performance measures with respect to both the trailer axles and the tractor rear axle be undertaken to obtain the rollover threshold that would be more pertinent.

Considering a steady turning maneuver, under the influence of a constant level of lateral perturbance the vehicle experiences two states of roll instability. In the primary stage, the tires on the inner wheels of the vehicle experience lift off, which corresponds to the maximum lateral acceleration the vehicle can withstand without rolling over [6]. As can be seen from Figure 2.3 for a tractor semitrailer combination, this stage corresponds to the point when the inner wheels of the trailer and tractor rear axles experiences wheel lift off. This state of roll instability is defined as the *Relative Roll Instability* condition. Following the attainment of this condition, a further increase in the vehicle roll angle yields a rapid decrease in a_y in a highly non-linear manner, which is attributed to insufficient restoring moment developed at the front axle. The vehicle thus quickly approaches a condition termed as *Absolute Roll Instability* condition under a continued lateral acceleration field. After the attainment of the relative roll instability condition, with continued application of the lateral perturbation the vehicle reaches tip-over position at which state the center of mass of the vehicle lies vertically above the contact point of the outer tires with the road surface and at this stage the vehicle cannot tolerate the lateral perturbation leading to an actual rollover.

The actual rollover of an articulated vehicle is thus preceded by the vehicle reaching relative roll instability condition, when the vehicle cannot sustain the lateral force while remaining stable; followed by the absolute rollover condition, when it possesses no tolerance to the lateral force. An open-loop rollover prevention strategy based upon early warning therefore must detect the onset of potential rollover on the basis of relative roll instability condition. While this strategy would be more reliable, the possibility of false warnings exists, which makes the identification of a reliable and robust metric imperative.

2.3 STATIC ROLL PLANE ANALYSIS OF TRACTOR SEMITRAILER COMBINATION

In the case of tractor semitrailers, although composite axles can be justified, it is essential to consider compliances such as those of frame and fifth wheel. Figure 2.4 shows the side view representation of a tractor semitrailer with composite axles. The tractor frame possesses a torsional stiffness K_{TF} which couples the two sprung weights of the tractor, W_{s1} and W_{s2}. The torsional stiffness of the fifth wheel and semitrailer structure, K_{T5}, couples the sprung weight of the semitrailer to that of the tractor.

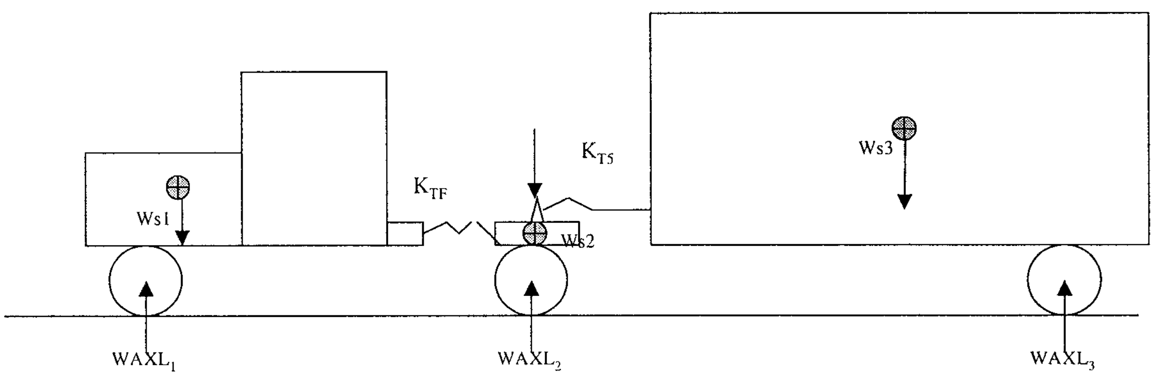


Figure 2.4: Composite axle representation of the tractor semitrailer combination

The roll plane view of the i^{th} unit (i=1,2) of the tractor semitrailer is shown in Figure 2.5. The sprung masses rotate about their respective roll centers, the roll angles of the sprung and unsprung masses are assumed to be small. The relative motion between the sprung and unsprung masses occurs about the roll centers, which slide freely relative to the axles perpendicular to the $\vec{y}_{ui}$ axis. The suspension springs are constrained to translate vertically and remain parallel to the $\vec{z}_{ui}$ axis and transmit elastic forces only. The tires are assumed to posses linear stiffness KT_{ij}, where i to the axle and j is the tire on a given axle (i= 1…3, j= 1,2..4). Owing to load transfer at higher levels of lateral acceleration, the lateral forces developed by the tires on the outside of the turn are considered significant in comparison with that on the inside of the turn. The effect of articulation angle on the roll response of the vehicle is considered insignificant. The pitch motion of the vehicle is incorporated in the model although the variations in the axle loads due to the pitch motion of the vehicle is considered negligible. The total vehicle load supported by each composite axle is thus assumed to remain constant during the roll process.

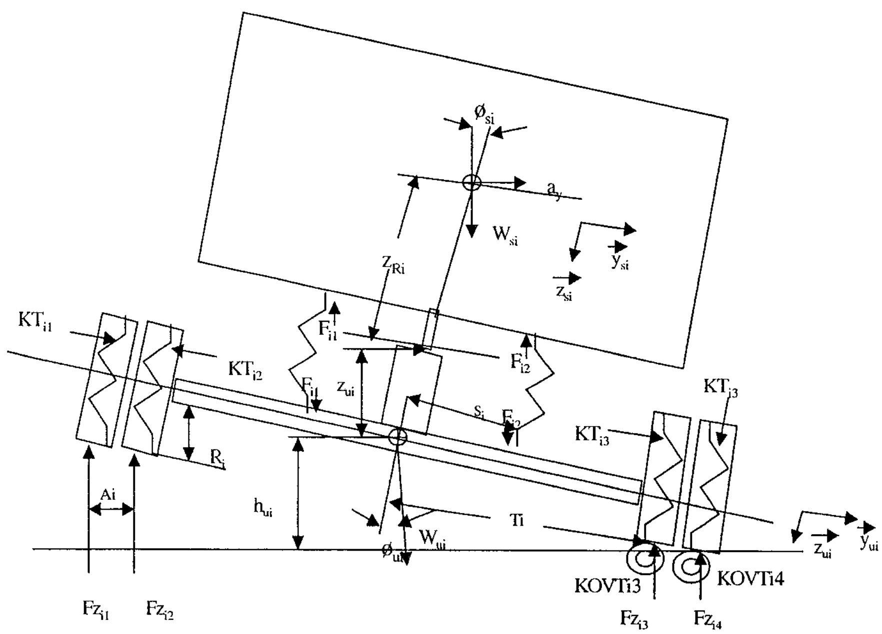


Figure 2.5: Motion and forces experienced by the i[th] unit of the vehicle in roll plane.

The geometrical variables used in the model include the sprung mass roll angle φ_{si} (i=1,2), unsprung mass roll angle φ_{ui} (i=1,2,3). The half trackwidth is represented by T_i (i=1,2,3) while the half suspension spread is represented by s_i (i=1,2,3). The distance between the axle cg and suspension roll center is represented by z_{ui} while the distance between the ground surface and the cg of the axle is represented by h_{ui}. The distance between the suspension roll center and the cg of the sprung mass is represented by z_{Ri}. The vertical shear force acting through the tractor frame is represented by W_{VFS} where $W_{VFS}= W_5 +W_{S2} - W_{AXL2}$, where W_{AXL2} is the normal load acting on the tractor's rear axle, W_{S2} is the sprung mass weight on the tractor's rear axle and W_5 is the load on the fifth wheel. The dual tire spacing is represented as A_i and the effective tire radius by R_i. The vertical and lateral tire forces on the j^{th} tire of the i^{th} axle are represented by F_{zij} and F_{yij} respectively. The static roll analysis is performed by balancing the roll moments acting on the sprung and unsprung masses, suspension forces and tire forces.

2.3.1 Roll Moment Analysis of Sprung Masses

The moments due to the suspension forces, lateral forces acting through the roll centers, torsional stiffness of the tractor and semi-trailer structures and lateral components of gravity forces constitute the moments acting on the sprung masses. The sum of roll moments acting in the roll plane of the tractor semitrailer as shown for ith unit illustrated in Figure 2.5 yields:

$$F_{i1}[s_i + z_{Ri}.\sin(\varphi_{si} - \varphi_{ui})] - F_{i2}[s_i - z_{Ri}.\sin(\varphi_{si} - \varphi_{ui})] + M_i - F_{Ri}.\, z_{Ri}.\cos(\varphi_{si} - \varphi_{ui})$$
$$- K_{ai}.\,(\varphi_{si} - \varphi_{ui}) = 0 \qquad (2.4)$$

where F_{ij} is the force of the suspension springs (i=1,2,3) and j=1,2 (left or right suspension spring). F_{Ri} denotes the lateral force transmitted between the sprung and unsprung masses of axle i. M_i denotes the roll moment transmitted through different roll sections owing to compliance of tractor frame and hitch mechanisms. K_{ai} represents the auxiliary roll stiffness of axle i, s_i is the half suspension lateral spread of the i^{th} axle. The roll moments for the different sections are:

$$M_1 = K_{TF}(\varphi_{s2} - \varphi_{s1}) + W_{VFS}(a_y - \varphi_{s1}).z_{FR1}$$

$$M_2 = K_{T5}(\varphi_{s3} - \varphi_{s2}) - K_{TF}(\varphi_{s2} - \varphi_{s1}) - (W_{VFS}.z_{FR2} + W_5.z_{52}).(a_y - \varphi_{s2})$$

$$M_3 = W_5.z_{53}.(a_y - \varphi_{s3}) - K_{T5}.(\varphi_{s3} - \varphi_{s2}) \tag{2.5}$$

where z_{5i} (i= 2,3) is the distance from the tractor's rear and trailer's suspension roll centers to the point of action of the lateral force due to W_5. z_{FRi} $(i= 1,2)$ denotes the distance from the tractor's front and rear suspension roll centers to the acting point of the lateral force due to W_{VFS} . The lateral acceleration acting on the vehicle is denoted by a_y. Since the roll angles are assumed to be small, equation 2.4 reduces to:

$$(F_{i1} - F_{i2}).s_i + [(F_{i1} + F_{i2}).\ z_{ui} - K_{ai}](\varphi_{si} - \varphi_{ui}) + M_i - F_{Ri}.\ z_{ui} = 0 \tag{2.6}$$

At the condition of equilibrium, the effect of a small change in the sprung mass roll angle yields:

$$(\Delta F_{i1} - \Delta F_{i2}).s_i + (\Delta F_{i1} + \Delta F_{i2}).\ z_{ui}.(\varphi si - \varphi ui) + [(F_{i1} + F_{i2}).\ z_{ui} - K_{ai}](\Delta\varphi_{si} - \Delta\varphi_{ui}) + \Delta M_i - \Delta F_{Ri}.\ z_{ui} = 0 \tag{2.7}$$

where $\Delta M_i\,(i=1,2,3)$ are $\Delta M_1 = K_{TF}(\Delta\varphi_{si} - \Delta\varphi_{ui}) + W_{VFS}.(\Delta ay - \Delta\varphi_{s1}).\ z_{FR1}$;

$\Delta M_2 = K_{T5}(\Delta\varphi_{s3} - \Delta\varphi_{s2}) - K_{TF}(\Delta\varphi_{s2} - \Delta\varphi_{s1}) - (W_{VFS}.z_{FR2} + W_5.z_{52}).(\Delta a_y - \Delta\varphi_{s2})$;

and $\Delta M_3 = W_5.z_{53}.(\Delta a_y - \Delta\varphi_{s3}) - K_{T5}.(\Delta\varphi_{s3} - \Delta\varphi_{s2})$

The variations in the deflections such that $\Delta\varphi_{s1}$, $\Delta\varphi_{ui}$ and Δz_{ui} are related to the changes in suspension forces for a given axle i as :

$$\Delta F_{ij} = \frac{\partial F_{ij}}{\partial \phi_{si}} \cdot \Delta\phi_{si} + \frac{\partial F_{ij}}{\partial \phi_{ui}} \cdot \Delta\phi_{ui} + \frac{\partial F_{ij}}{\partial z_{ui}} \cdot \Delta z_{ui} \qquad \text{where } i=1,2,3 \text{ and } j=1,2 \tag{2.8}$$

The equations of the left-hand side and right-hand side suspension forces of the suspension springs are represented as:

$$\Delta F_{i1}= K_{i1}\ .\ [\Delta z_{ui} - s_i\ (\Delta\varphi_{si}\ -\Delta\varphi_{ui})]$$

$$\Delta F_{i2}= K_{i2}\ .\ [\Delta z_{ui} + s_i\ (\Delta\varphi_{si}\ -\Delta\varphi_{ui})] \qquad (2.9)$$

where K_{ij} represents the equivalent linear spring constant and z_{ui} is the vertical motion of the axle i along the $\vec{z}_{ui}$ axis . Thus the total suspension sprung force developed along the $\vec{z}_{ui}$ axis is expressed as:

$$F_{i1}+F_{i2}= (WAXL_i\text{-}W_{ui})(a_y.sin\varphi_{ui} + cos\varphi_{ui}) \qquad (2.10)$$

The variations in the total spring forces, considering small angles, is thus expressed as:

$$\Delta F_{i1}+\Delta F_{i2}= (WAXL_i\text{-}W_{ui})(a_y.\Delta\varphi_{ui} + \Delta a_y.\varphi_{ui}) \qquad (2.11)$$

The change in the lateral force F_{Ri}, acting through the roll center parallel to the $\vec{y}_{ui}$ axis is represented as:

$$\Delta F_{Ri} = (WAXL_i\text{-}W_{ui})(\Delta a_y - \Delta\varphi_{ui}) \qquad i=1,2,3 \qquad (2.12)$$

Substituting equations 2.9, 2.11 and 2.12 in equation 2.7 the equilibrium of roll moments acting on the sprung mass supported by the front axle of the tractor is obtained as:

$$(K_{11}\text{-}K_{12}).s_1.\Delta\ z_{u1}\ -\ (K_{11}+K_{12})s_1^2\ \Delta\varphi_{s1}\ +\ (K_{11}+K_{12})s1^2\ \Delta\varphi_{u1}\ +\ (F_{11}+F_{12})z_{R1}(\Delta\varphi_{s1}\ -\Delta\varphi_{u1})+ K_{TF}(\Delta\ \varphi_{s2}\text{-}\Delta\varphi_{s1})+\ (WAXL_1\text{-}W_{u1})(\ \varphi_{s1}\ -\varphi_{u1})\ z_{R1}(a_y.\Delta\varphi_{u1}\ +\Delta a_y.\varphi_{u1})\text{-}\ (WAXL_1\text{-}W_{u1}).\ z_{R1}.(\ \Delta a_y - \Delta\varphi_{u1})\ +\ W_{VFS}.(\Delta a_y\text{-}\Delta\varphi_{s1}).\ z_{FR1} = 0 \qquad (2.13)$$

Similarly, the equations for the remaining sprung masses are obtained as:

$$\text{-}W_{VFS}.(\Delta ay\text{-}\Delta\varphi_{s2}).\ z_{FR2}\ +\ K_{T5}.(\ \Delta\ \varphi_{s3}\text{-}\Delta\varphi_{s2})\ - W_5.\ (\Delta a_y\text{-}\Delta\varphi_{s2})\ z_{52}\ +\ (K_{21}\text{-}K_{22}).s_2.\ \Delta\ z_{u2}\ - (K_{21}+K_{22}).s_2^2.\ \Delta\varphi_{s2}\ +\ (K_{21}+K_{22}).s_2^2.\ \Delta\varphi_{u2}\ +\ (F_{21}+F_{22})z_{R2}(\Delta\varphi_{s2}\ -\Delta\varphi_{u2})\ -\ K_{TF}(\Delta\ \varphi_{s2}\text{-}\Delta\varphi_{s1})\ + (WAXL_2\text{-}W_{u2})(\ \varphi_{s2}\ -\varphi_{u2})\ z_{R2}(a_y.\Delta\varphi_{u2}\ +\Delta a_y.\varphi_{u2})\text{-}\ (WAXL_2\text{-}W_{u2}).\ z_{R2}.(\ \Delta a_y - \Delta\varphi_{u2})=0 \qquad (2.14)$$

and

$(K_{31}-K_{32})s3\,\Delta z_{u3} - (K_{31}+K_{32})s3^2\,\Delta\varphi_{s3} + (K_{31}+K_{32})\,s3^2\,\Delta\varphi_{u3} + (F_{31}+F_{32})\,z_{R3}.(\Delta\varphi_{s3} - \Delta\varphi_{u3})-$

$K_{TF}(\Delta\varphi_{s3}-\Delta\varphi_{s2}) + (WAXL_3-W_{u3})(\varphi_{s3}-\varphi_{u3})\,z_{R3}(a_y.\Delta\varphi_{u3} + \Delta a_y.\varphi_{u3}) - (WAXL_3-W_{u3}).\,z_{R3}..(\Delta a_y -$

$\Delta\varphi_{u3}) - W_{VFS}.(\Delta a_y - \Delta\varphi_{s3}).\,z_{53} = 0$ (2.15)

The roll moments caused by small deviations from the equilibrium condition acting on the three composite axles are represented by equations 2.13, 2.14 and 2.15.

2.3.2 Roll Moment Analysis of Unsprung Masses

The moments caused by suspension, tire and lateral forces acting through the roll center and lateral forces developed at the tires constitute the roll equilibrium equations of the unsprung masses. The roll moment equation for the i^{th} unsprung mass can be expressed as:

$$(F_{zi1}+F_{zi2}+F_{zi3}+F_{zi4}).R_i.\sin\varphi_{ui} + (F_{zi1}+F_{zi4})(T_i+A_i)\cos\varphi_{ui} + (F_{zi2}-F_{zi3}).Ti.\cos\varphi_{ui} -$$

$$(F_{zi3}+F_{zi4}).y_i.\cos\varphi_{ui} - (F_{i1}-F_{i2}).s_i + F_{Ri}.z_{ui} + F_{yi}.h_{ui} + OVT_{i3} + OVT_{i4} = 0 \quad (2.16)$$

Considering small increment in the roll angle the equation can be rewritten as:

$$(\Delta F_{zi1}+\Delta F_{zi2}+\Delta F_{zi3}+\Delta F_{zi4}).R_i.\varphi_{ui} + (F_{zi1}+F_{zi2}+F_{zi3}+F_{zi4}).R_i.\Delta\varphi_{ui} + (\Delta F_{zi1}+\Delta F_{zi4})(T_i+A_i)+$$

$$(\Delta F_{zi2}-\Delta F_{zi3}) - (\Delta F_{zi3}+\Delta F_{zi4}).yi + (\Delta F_{i2}\Delta F_{i1}).si + \Delta F_{Ri}.z_{ui} + F_{yi}.\Delta h_{ui} + \Delta OVT_{i3} + \Delta OVT_{i4} = 0 \quad (2.17)$$

Increase in the loads ΔF_{zij} influences the unsprung mass roll angle $\Delta\varphi_{ui}$ and the unsprung mass bounce Δh_{ui}. The changes in the normal forces acting on the tires of a composite axle are expressed as:

$$\Delta F_{zi1} = -KT_{i1}.\,((T_i+A_i)\,\Delta\varphi_{ui} - \Delta h_{ui})$$

$$\Delta F_{zi2} = -KT_{i2}.\,(T_i\Delta\varphi_{ui} - \Delta h_{ui})$$

$$\Delta F_{zi3} = KT_{i3}.\,((T_i+y_i)\,\Delta\varphi_{ui} - \Delta yi.\,\varphi_{ui} + \Delta h_{ui})$$

$$\Delta F_{zi3} = KT_{i4}.\,((T_i+A_i-y_i)\,\Delta\varphi_{ui} - \Delta yi.\,\varphi_{ui} + \Delta h_{ui}) \quad i=1,2,.. \quad (2.18)$$

The increment in the roll resisting moment at the tire road contact is given by

$\Delta OVT_{i3} = - KOVT_{i3}. \Delta\varphi_{ui}$ and $\Delta OVT_{i4} = - KOVT_{i4}. \Delta\varphi_{ui}\ i=1,2,3..$ (2.19)

where $KOVT_{ij}$ is the linear overturning stiffness of the j-th tire on the i-th axle. Substituting equations 2.9, 2.12, 2.18 and 2.19 in 2.17 yields:

$$-[(WAXL_i-W_{ui})(R_i+z_{ui})+W_{ui}.h_{ui}]\Delta a_y + (K_{i1}+K_{i2})s_i^2\ \Delta\varphi_{si} - [(Ki1+K_{i2})s_i^2 - WAXL_i.R_i - (WAXL_i-W_{ui}).z_{ui}+KOVT_{i3} + KOVT_{i4} + KT_{i1}(T_i+A_i)^2 + KT_{i2}.T_i^2 + KT_{i3}(T_i-y_i)^2 + KTi4(Ti+Ai-yi)^2]\ \Delta\varphi_{ui} + [(WAXL_i-W_{ui})(a_y - \varphi_{ui}) -(K_{i1}-K_{i2}).si]\ \Delta\ z_{ui} + [(KT_{i1}-KT_{i4})(T_i+A_i)+(KT_{i2}-KT_{i3}).T_i + (KT_{i3}+KT_{i4})yi+WAXLi.\ a_y]\ \Delta h_{ui} - \{Fz_{i3}+Fz_{i4}+[(KT_{i3}+KT_{i4}).(T_i-y_i)+KT_{i4}.Ai]\ \varphi_{ui}\}\Delta y_i= 0 \quad (2.20)$$

2.3.3 Vertical Suspension Forces and Tire Forces

The forces generated by the compression and extension of the suspension springs are obtained by substituting equation 2.9 in equation 2.11 to maintain equilibrium along the $\vec{z}_{ui}$ axis, which yields:

$$(K_{i1}+K_{i2})\Delta z_{ui} -(K_{i1}-K_{i2}).s_i\ \Delta\varphi_{si} + (K_{i1}-K_{i2}).s_i\ \Delta\varphi u_i + (WAXL_i-W_{ui})\varphi u_i\ \Delta a_y - (WAXL_i-W_{ui})\ \Delta\varphi_{ui}.a_y= 0 \qquad i=1,2,3 \quad (2.21)$$

Assuming that the load on each composite axle remains constant, the vertical tire forces are expressed as: $F_{zi1}+F_{zi2}+F_{zi3}+F_{zi4}= WAXL_i$ (2.22)

Considering small deviations about an equilibrium condition:

$$\Delta F_{zi1}+\Delta F_{zi2}+ \Delta F_{zi3}+\Delta F_{zi4}=0 \quad (2.23)$$

Thus equation 2.17 can be substituted in 2.22 to get:

$$[(-KT_{i1}+ KT_{i4}).\ (T_i+A_i)+ (-KT_{i2}+KT_{i3}).T_i - (KT_{i3}+KT_{i4})yi].\ \Delta\varphi u_i + (KT_{i1}+KT_{i2}+KT_{i3}+ KT_{i4}).\ \Delta h_{ui} - (KT_{i3} + KT_{i4})\ \varphi u_i\ \Delta y_i= 0 \quad (2.24)$$

At the lateral acceleration limit in the vicinity of the rollover threshold, the outside tires on the turn experience almost the entire axle load. The lateral equilibrium equation for axle i can be expressed in terms of the outside tire stiffness alone, which for small roll angle is:

$$WAXL_{i.}\ \Delta a_y = (KYT_{i3}+KYT_{i4})\ \Delta y_i \qquad (2.25)$$

where KYT_{i3} and KYT_{i4} are the spring rates of the outside tires of axle i.

The equilibrium equations for the tractor semitrailer can be represented in matrix form as:

$$[A] = \{\Delta x\} = \{b\}\Delta\phi_{ST} \qquad (2.26)$$

where $\Delta\phi_{ST}$ is the increment in roll angle of the semitrailer sprung mass. The set of equation (2.26) are solved to obtain the variation in lateral acceleration and roll angles of the sprung and unsprung masses related to the increment in roll angle of the last sprung weight. $[A]$ and $\{b\}$ are (15X15) and (15X1) matrices which contain the vehicle parameters and $\{\Delta x\}$ is the vehicle response variable vector given by:

$$\{\Delta x\}^T = (\Delta a_y,\ \Delta\varphi_{si,},\ \Delta\varphi u_i,\ \Delta z_{ui,},\ \Delta h_{ui},\ \Delta y_i) \qquad (2.27)$$

where, $i=1,2,3$ for unsprung masses and $i=1,2$ for sprung masses. A decrease in slope in the a_y- φ_{ST} plot with increase in the semi-trailer roll angle depicts the relative rollover condition for the tractor semitrailer combination.

2.4 RELATIVE ROLL PLANE ANALYSIS OF ARTICULATED VEHICLES IN DYNAMIC DIRECTIONAL MANUEVERS

The static roll instability analysis deals with the roll instability of the vehicle when it experiences steady lateral acceleration at a constant speed steady turning maneuver. However, under a dynamic maneuver, the attainment of the *SRT* does not guarantee a rollover, which occurs if the lateral acceleration acts constantly for a definite period of time. Moreover, the dynamic roll response characteristics of the different units of a tractor semitrailer are quite different in both amplitude and phase angle. The UMTRI model commonly known as the yaw/roll model [27] is ideally suited for the simulation of roll response of an articulated vehicle. The model can simulate a vehicle system with upto 52 degrees of freedom with the limitation that the forward velocity has to remain constant. The constant velocity yaw/roll model is utilized in this investigation to study the response of articulated vehicles under static and dynamic maneuvers. It is however essential to understand the methodology used by the model for accurate entry of vehicle data as well as analysis and interpretation of results. In this study of relative roll instability of a tractor semitrailer combination it is assumed that each unit consists of a rigid sprung mass, and the sprung and unsprung masses are connected by compliant suspension system. The relative roll angle between the sprung and unsprung masses as well as the pitch angles of the sprung masses is assumed to be small. For a given axle, the relative motion between the sprung and unsprung masses are assumed to occur about the roll centre corresponding to the axle. The location of the roll center, assumed to be underneath the sprung mass, is free to move in the vertical axis of the unsprung mass. The principal axes of inertia for the sprung and unsprung masses are assumed to coincide with their respective co-ordinate systems. The suspensions are assumed to be independent of each other thereby neglecting the inter-axle load transfer.

The sprung masses are treated as rigid bodies with five degrees of freedom – roll, yaw, pitch, lateral and vertical. Each axle is modeled as a two degree of freedom system and is free to move in the roll and bounce modes. For a five axle tractor semitrailer combination the axis system can be described by a co-ordinate system attached at the sprung mass cg of each unit i.e X_{si}, Y_{si},, Z_{si} ($i=1,2$) and by a co-ordinate system attached to the center of each axle represented by X_{uj}, Y_{uj},, Z_{uj} ($j=1,2,3..5$).

The global co-ordinate system relative to which the location of the vehicle is specified is represented by X_R, Y_R and Z_R. Euler angles ψ_{si},, θ_{si} and φ_{si} are used to represent the relationship between the body fixed axis system and the inertial system X_R, Y_R and Z_R. To describe the orientation of the sprung mass axis system the Euler angles are used to derive the transformation equation between sprung and inertial mass axis systems. As the axis system of the sprung masses is alike, the transformation equations are shown for one sprung mass.

Considering yaw rotation: $\{\vec{x}_R, \vec{y}_R, \vec{z}_R\}^T = [a_{ij}]\ \{\vec{x}'_R, \vec{y}'_R, \vec{z}'_R\}^T$ (2.28)

Considering pitch rotation : $\{\vec{x}'_R, \vec{y}'_R, \vec{z}'_R\}^T = [b_{ij}]\ \{\vec{x}''_R, \vec{y}''_R, \vec{z}''_R\}^T$ (2.29)

Considering roll rotation : $\{\vec{x}''_R, \vec{y}''_R, \vec{z}''_R\}^T = [c_{ij}]\ \{\vec{x}_R, \vec{y}_R, \vec{z}_R\}^T$ (2.30)

Combining the equations of yaw, pitch and roll rotations the transformation matrix is obtained as:

$$\{\vec{x}_R, \vec{y}_R, \vec{z}_R\}^T = [A_{ij}]_i\ \{\vec{x}_{si}, \vec{y}_{si}, \vec{z}_{si}\}^T \tag{2.31}$$

where $[A_{ij}]_i=[a_{ij}]\ [b_{ij}]\ [c_{ij}]$, denotes the transformation matrix from the i[th] sprung mass body fixed co-ordinate system to the inertial co-ordinate system. Assuming small pitch angles during directional maneuvers the transformation matrix can be expressed as:

$$[A_{ij}]_i = \begin{bmatrix} \cos\psi_{si} & -\sin\psi_{si}\cos\phi_{si} + \theta_{si}\cos\psi_{si}\sin\phi_{si} & \sin\psi_{si}.\sin\phi_{si} + \theta_{si}\cos\psi_{si}\cos\phi_{si} \\ \sin\psi_{si} & \cos\psi_{si}\cos\phi_{si} + \theta_{si}\sin\psi_{si}\sin\phi_{si} & -\cos\psi_{si}\sin\phi_{si} + \theta_{si}\sin\psi_{si}\cos\phi_{si} \\ -\theta_{si} & \sin\phi_{si} & \cos\phi_{si} \end{bmatrix} \tag{2.32}$$

The inertial axis system can be related to the sprung mass system by:

$$\{\vec{x}_{si}, \vec{y}_{si}, \vec{z}_{si}\}^{T} = [A_{ij}]_{i}^{-1}\{\vec{x}_{R}, \vec{y}_{R}, \vec{z}_{R}\}^{T} \tag{2.33}$$

where

$$[A_{ij}]_{I}^{-1} = \begin{bmatrix} \cos\psi_{si} & \sin\psi_{si} & -\theta_{si} \\ -\sin\psi_{si}\cos\phi_{si} + \theta_{si}\cos\psi_{si}\sin\phi_{si} & \cos\psi_{si}\cos\phi_{si} + \theta_{si}\sin\psi_{si}\sin\phi_{si} & \sin\phi_{si} \\ \sin\psi_{si}.\sin\phi_{si} + \theta_{si}\cos\psi_{si}\cos\phi_{si} & -\cos\psi_{si}\cos\phi_{si} + \theta_{si}\sin\psi_{si}\sin\phi_{si} & \cos\phi_{si} \end{bmatrix} \tag{2.34}$$

The axles on each unit are constrained to bounce and roll with respect to the corresponding sprung mass. The transformation equation of the sprung and unsprung mass axis system is given by:

$$\begin{Bmatrix} \vec{x}_{u} \\ \vec{y}_{u} \\ \vec{z}_{u} \end{Bmatrix} = \begin{bmatrix} 1 & \theta_{si}.\sin\phi_{si} & \theta_{si}\cos\phi_{si} \\ -\theta_{si}\sin\phi_{u} & \cos(\phi_{si} - \phi_{u}) & -\sin(\phi_{si} - \phi_{u}) \\ -\theta_{si}\cos\phi_{u} & \sin(\phi_{si} - \phi_{u}) & \cos(\phi_{si} - \phi_{u}) \end{bmatrix} \tag{2.35}$$

where for the tractor unit (i=1),j=1,2,3 and for the semitrailer unit (i=2);j=4,5

2.4.1 Equations of Motion for Each Sprung Mass

Five equations of motion encompass the equations of motion for the sprung masses, these include equations of motion for the roll, pitch, yaw, lateral and vertical motions which can be expressed by:

Equation of vertical motion:

$$m_{si}.v_{si}' - m_{si}.(p_{si}.v_{si} - r_{si}.l_{lsi}) = \sum Z_{si} \tag{2.36}$$

Equation of lateral motion:

$$m_{si}.l_{si}' - m_{si}.(r_{si}.v_{si} - y_{si}.v_{lsi}) = \sum Y_{si} \tag{2.37}$$

Equation of roll motion:

$$I_{xxsi}.r_{si}' - (I_{yysi} - I_{zzsi}).p_{si}.l_{si} = \sum R_{si} \tag{2.38}$$

Equation of pitch motion:

$$I_{yysi}.p_{si}' - (I_{zzsi} - I_{xxsi}).r_{si}.y_{si} = \sum S_{si} \tag{2.39}$$

Equation of yaw motion:

$$I_{zzsi}.y_{si}' - (I_{xxsi} - I_{yysi}).r_{si}.p_{si} = \sum T_{si} \tag{2.40}$$

where m_{si} represents the sprung mass, I_{xxsi}, I_{yysi}, I_{zzsi} represent the mass moment of inertias about the X_{si}, Y_{si} and Z_{si} directions respectively. r_{si}, p_{si} and y_{si} represent the roll, pitch and yaw velocities of the sprung mass i and l_{si} and v_{si} represent the lateral and vertical velocities, while $\sum Z_{si}$, $\sum Y_{si}$ represent the net external forces and moments acting in Z and Y co-ordinates while $\sum R_{si}$, $\sum S_{si}$ and $\sum T_{si}$ represent the forces and moments corresponding to the roll, pitch and yaw motions respectively.

2.4.2 Equations of Motion for Each Unsprung Mass

The equations of motion for each unsprung mass can be described by the equation of vertical motion and the equation of roll motion. The various parameters in each equation represents the property of the given axle *(j=1,2,3,4,5)*.

Equation of Vertical Motion

$$m_{uj}.\ v_{uj}' = \sum Z_{uj} \tag{2.41}$$

Equation of Roll Motion

$$I_{xxuj}.\ r_{uj}' = \sum R_{uj} \tag{2.42}$$

where m_{uj} and I_{xxuj} are the unsprung mass and the unsprung mass moment of inertia respectively for axle j and $\sum Z_{uj}$ and $\sum R_{uj}$ are the net external force and moment in the vertical and roll directions which include the gravitational forces. The yaw/roll model also considers the lateral forces and moments at the tire-road interface, suspension forces and moments due to left and right suspension springs, vertical forces and moments at the tire road interface and auxiliary roll moments of the suspension springs for each axle considered.

2.4.3 Constraint Equations

The sprung masses in the yaw/roll model are rigidly coupled in translation. The forces that get transmitted through the coupling can be obtained using kinematic constraints [6]. The acceleration at the coupling point thus is the same for both units. The constraint moments are obtained by considering the roll compliance and the transmitted moments are thereby obtained from the relative roll displacement between the two units. For a vehicle with n sprung masses and m unsprung masses the equations of motion can be represented by k (where $k=5n+2m$) differential equations of motion such that:

$$M\ddot{\vec{x}} = \vec{y} + N\vec{f_c} \tag{2.43}$$

where M is the (k X k) inertial matrix $\ddot{\vec{x}}$ is the acceleration vector of length k , $\vec{y}$ is a position vector of size k comprising of gravitational , suspension and roll center forces, N is the (k X j) matrix of vehicle dimensions and $\vec{f_c}$ is a vector of unknown constraint forces. The kinematic constraints posted by the various hitch points can be written as acceleration constraint equation: $B\ddot{\vec{x}} = \vec{c}$ (2.44)

where B is a (j X k) matrix function of the vehicle dimensions and $\vec{c}$ is a vector of size j and a function of $\vec{x}, \dot{\vec{x}}$ and vehicle dimensions. Solving equations 2.43 and 2.44 yields the constraint force vector:

$$f_c = -[BM^{-1}N]^{-1}\{\vec{c} - BM^{-1}\vec{y}\} \qquad (2.45)$$

The conventional fifth wheel connection can be obtained by considering it as a roll coupler as shown in Figure 2.6. The X_{si}, Y_{si}, Z_{si} (i=1,2) co-ordinates are attached to the unit 1or 2. The co-ordinate system [X'_{si}, Y'_{si} and Z'_{si} (i=1)] has similar yaw and pitch angles as the lead unit while having different roll angle ϕ'_{s1}. K_{T5} represents the fifth wheel roll stiffness rate. The roll moment acting on the fifth wheel is represented by M_{x1} where $M_{x1}=K_{T5}.(\phi'_{s1} - \phi_{s1})$. The pitch axis $\vec{Y}'_{s1}$ is assumed to be perpendicular to the yaw axis $\vec{Z}_{s2}$ which yields:

$$\vec{y}'_{s1}.\vec{z}_{s2} = 0 \qquad (2.46)$$

where $\vec{y}'_{s1}$ and $\vec{z}_{s2}$ are unit vectors in $\vec{Y}'_{s1}$ and $\vec{Z}_{s2}$ directions. The X_R, Y_R and Z_R co-ordinate system is fixed to the ground, and define the vehicle trajectory and motion. The constraining roll and pitch moments acting on the semitrailer sprung weight are determine by the coordinate transformation whereby: $M_{x2}=-M_{x1}.cos(\psi_{s2}-\psi_{s1})$ and

$M_{Y2}= -M_{x1}.sin\varphi_{s2}cos(\psi_{s2}-\psi_{s1}) - cos\ \varphi_{s2}.sin(\psi_{s2}-\psi_{s1})-\theta.sin\varphi_{s2}$ (2.47)

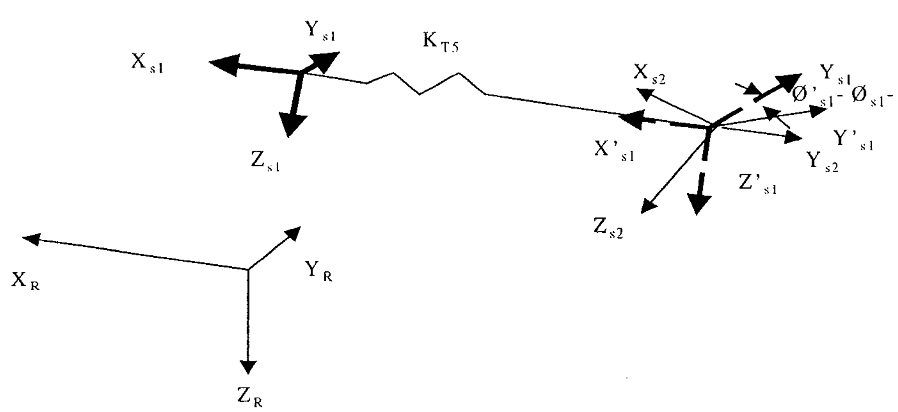


Figure 2.6: Illustration of the axes systems at the articulation point.

Through Euler angles in yaw (ψ_{si}), pitch (θ_{si}) and roll (φ_{si}), the inertial axis system is related to the body fixed axis system. The transformation matrix that relates the co-ordinate system is expressed as:

$$ {}^{R}_{si}T = T_{zsi}.(\psi_{si}).T_{ysi}(\theta_{si}).T_{xsi}(\phi_{si}) $$
$$ = \begin{bmatrix} \cos\psi_{si}.\cos\phi_{si} & \cos\psi_{si}.\sin\theta_{si}.\sin\phi_{si} - \sin\psi_{si}.\cos\phi_{si} & \cos\psi_{si}.\sin\theta_{si}.\cos\phi_{si} + \sin\psi_{si}.\sin\theta_{si} \\ \sin\psi_{si}.\cos\theta_{si} & \sin\psi_{si}.\sin\theta_{si}.\sin\phi_{si} + \cos\psi_{si}.\cos\phi_{si} & \sin\psi_{si}.\sin\theta_{si}.\cos\phi_{si} - \cos\psi_{si}.\sin\phi_{si} \\ -\sin\theta_{si} & \cos\theta_{si}.\sin\phi_{si} & \cos\theta_{si}.\cos\phi_{si} \end{bmatrix} \quad (2.48) $$

and ${}^{si}_{R}T = {}^{R}_{si}T^{T}$ =

$$ \begin{bmatrix} \cos\psi_{si}.\cos\phi_{si} & \cos\theta_{si}.\sin\psi_{si} & -\sin\theta_{si} \\ \cos\psi_{si}.\sin\theta_{si}.\sin\phi_{si} - \sin\psi_{si}.\cos\phi_{si} & \sin\psi_{si}.\sin\theta_{si}.\sin\phi_{si} + \cos\psi_{si}.\cos\phi_{si} & \cos\theta_{si}.\sin\phi_{si} \\ \cos\psi_{si}.\sin\theta_{si}.\cos\phi_{si} + \sin\psi_{si}.\sin\theta_{si} & \sin\psi_{si}.\sin\theta_{si}.\cos\phi_{si} - \cos\psi_{si}.\sin\phi_{si} & \cos\theta_{si}.\cos\phi_{si} \end{bmatrix} \quad (2.49) $$

where ${}^{R}_{si}T$ transforms co-ordinate system (X_{si}, Y_{si}, Z_{si}) into to inertial system (X_R, Y_R, Z_R) and ${}^{si}_{R}T$ is the transpose of ${}^{R}_{si}T$. The unit vectors are related by:

$$ \begin{bmatrix} \vec{x}_{si} \\ \vec{y}_{si} \\ \vec{z}_{si} \end{bmatrix} = {}^{si}_{R}T \begin{bmatrix} \vec{x}_{R} \\ \vec{y}_{R} \\ \vec{z}_{R} \end{bmatrix} \quad (2.50) $$

Substituting 2.50 into 2.46 we get the roll angle of the reference co-ordinate system [$X'_{si}, Y'_{si}, Z'_{si}$] for the lead unit ($i=1$) as:

$$ \phi_{s1} = \arctan\left[\frac{\sin\phi_{s2}.\cos(\psi_{s2}-\psi_{s1}) - \theta_{s2}.\cos\phi_{s2}.\sin(\psi_{s2}-\psi_{s1})}{\theta_{s1}.\sin(\psi_{s2}-\psi_{s1}).\sin\phi_{s2} + \cos\phi_{s2}}\right] \quad (2.51) $$

The roll and pitch moments transmitted through the articulation point are obtained in terms of pitch and roll displacements between the adjacent units and therefore the acceleration constraint approach is deemed redundant to solve for the pitch and roll moments.

2.4.4 Suspension and Tire Forces

The equations of motion corresponding to sprung and unsprung masses show that the terms on the right hand side of the equations 2.36-2.40 and 2.41-2.42 involve suspensions, roll center, tire and constraint forces. The suspension forces F_{sj} constitutes of lateral force F_{Lj} and vertical forces acting at right angle to the Y_{uj} axis of the j-th axle such that:

$$F_{sj} = \begin{bmatrix} 0 \\ F_{Lj} \\ -(F_{j1}+F_{j2}) \end{bmatrix}^T \begin{bmatrix} \vec{x}_{uj} \\ \vec{y}_{uj} \\ \vec{z}_{uj} \end{bmatrix} \tag{2.52}$$

Equation 2.52 can be written with respect to the co-ordinate system fixed on the sprung mass i as:

$$F_{sj} = \begin{bmatrix} 0 \\ F_{Lj} \\ -(F_{j1}+F_{j2}) \end{bmatrix}^T {}^{uj}_{si}R \begin{bmatrix} \vec{x}_{si} \\ \vec{y}_{si} \\ \vec{z}_{si} \end{bmatrix} \tag{2.53}$$

where ${}^{ui}_{si}R$ is the relation matrix of the unsprung mass j with respect to the sprung mass i presented as:

$${}^{ui}_{si}R = \begin{bmatrix} 1 & \beta_{si}.\sin\phi_{si} & \beta_{si}.\cos\phi_{si} \\ -\beta_{si}.\sin\phi_{uj} & \cos(\phi_{si}-\phi_{uj}) & -\sin(\phi_{si}-\phi_{uj}) \\ -\beta_{si}.\cos\phi_{uj} & \sin(\phi_{si}-\phi_{uj}) & \cos(\phi_{si}-\phi_{uj}) \end{bmatrix} \tag{2.54}$$

where β_{si} is the Euler pitch angle of sprung mass i. Considering the dynamic equations of the axle forces along the Y_{uj} direction the lateral suspension force can be computed as :

$$F_{Lj} = -m_{uj}.a_{uj}.\vec{y}_{uj} + \sum(F_{yj}.\cos\phi_{uj}) - \sum(F_{zj}.\sin\phi_{uj}) + m_{uj}.g.\sin\phi_{uj} \tag{2.55}$$

where a_{uj} is the lateral acceleration of axle *j*. $\sum F_{yj}$ and $\sum F_{zj}$ are the total lateral and vertical forces at the tire-road interface.

The yaw/roll model allows for the measured tire data to be used in obtaining the lateral force and aligning moments at the road-tire interface. At each step of integration, the vertical load, slip angle and the resulting lateral forces and aligning torques are obtained. The vertical load and side- slip angles are obtained in terms of the velocities and displacements of the sprung and unsprung masses. The side-slip angle is described in terms of the body fixed velocities of the sprung mass and the axles. For the *k*-th tire on the *j*-th axle, the side slip angles can be expressed as:

$$\alpha_{jk} = \tan^{-1}\left(\frac{v_{axlej}}{u_{tirejk}}\right) - \partial_i \tag{2.56}$$

where v_{axlej} represents the lateral velocity of axle j and u_{tirejk} represents the forward velocity of the *k*-th tire on the *j*-th axle while ∂_i represents the angle made by the wheel plane with respect to the longitudinal axis of the sprung mass co-ordinate system. The vertical load acting on the *k*-th tire of the *j*-th axle, F_{zjk}, can be computed from the compliance of the KT_{jk}, which represents the tire vertical stiffness of *k*-th tire of the *j*-th axle, and vertical deflection, Δ_{jk} and expressed as:

$$F_{zjk} = KT_{jk}.\ \Delta_{jk} \tag{2.57}$$

For any given steering input, the model solves the differential equations to establish different forces and moments at the tire-road interface, suspension forces and moments, as well as constraint forces and moments. These forces in turn are used in the the equations of motion to obtain the response of the vehicle through numerical methods.

The Yaw/Roll model developed by UMTRI uses this approach for simulation of rollover for articulated vehicles [27]. All simulations for dynamic maneuvers in this investigation is carried out using the UMTRI Yaw/Roll model described in this section.

2.5 SUMMARY

General rollover mechanics as well as the mechanics of rollover involving tractor semitrailer combination is analyzed and the relevant roll instability conditions: relative and absolute are identified. The criteria for attainment of these conditions is obtained and the pertinence of utilizing either of these roll instability criteria is discussed. The static roll plane model for tractor semitrailer combination is developed in accordance with the Yaw/Roll model in order to derive the equations of static roll equilibrium based upon forces and moments involving sprung and unsprung masses, suspension and tire forces. The solution of the equilibrium equations would yield the relationship between the different response parameters and help to simulate the roll dynamics of the vehicle combination. UMTRI yaw/roll model is proposed for the dynamic roll response characteristics of the tractor semitrailer combination. The solution of these equations through numerical methods would provide the desired simulation results when the vehicle is simulated as per given input parameters in terms of steering input and design and operating conditions. The models developed and discussed in this chapter are used extensively in the following chapters for simulation of static and dynamic rollover performance of articulated vehicles.

CHAPTER 3

ROLLOVER INDICES

3.1 INTRODUCTION

Articulated vehicles tend to approach their roll stability limits during low-speed cornering or high-speed directional maneuvers. Reported analyses of accident data suggest that more than three-fourth of the rollover accidents in Canada could be classified as maneuver-induced [71]. The measures of roll instability reported in literature are generally derived from analysis under steady turning or dynamic directional maneuvers. Early warning of potential roll instability requires the establishment of a rollover criterion and identification of motion cues that are directly related to the onset of rollover under both static and dynamic conditions. The relative roll instability criterion as discussed in the previous chapter is more relevant for early detection of roll instability in articulated vehicles as it provides sufficient lead time as compared to the absolute roll instability criterion. The measures based on absolute rollover criterion are thus not considered due to their limitations for early warning and control.

The development of an early warning and control algorithm necessitates the evaluation of the relative roll instability performance measures in terms of lead-time issue, reliability and measurability. A wide range of measures for the detection of an impending rollover have been suggested, such as the static rollover threshold (*SRT*), sprung and unsprung mass roll angles, load transfer ratio (*LTR*), roll safety factor (*RSF*), rearward amplification ratio (*RAR*), effective lateral acceleration (*ELA*), differential wheel slip, steering rate, and wheel loads [6,8,13,36,39,41,74].

In this chapter, these indices of relative roll instability, classified into measures of static and dynamic roll instability, are reviewed and assessed for their potential use in predicting the onset of a rollover. Those measures that correlate well with the relative roll instability criterion are identified. The limitations of the proposed measures are described and the required attributes of a generalized measure of impending rollover are outlined. The rollover indices that warrant investigation through an intensive sensitivity analysis are identified.

3.2 MEASURES OF ROLLOVER

The effective detection of a potential rollover risk is greatly dependent on the measures of roll instability that could be continuously monitored and processed in an efficient manner. The identification of a rollover metric that could be applied to establish the onset of an instability of the vehicle forms the primary requirement for the development of a generally applicable detection and warning algorithm. Several measures based on static and dynamic roll stability criteria have been reported in the literature [38,39,41,72,73], which may be applied for prediction of an impending relative rollover. The reported measures, classified into static and dynamic measures, are discussed below.

3.2.1 Measures of Static Roll Instability

The static roll stability of articulated vehicles have been widely proposed as the basis for prediction of rollover propensity [6,36,39,41,74]. The measures of static roll instability mostly describe the magnitude of lateral acceleration corresponding to the maximum net restoring moment beyond which the vehicle experiences a divergent roll response. The measures based upon static roll instability include the static rollover threshold acceleration (*SRT*) and static safety factor (*SSF*). These measures characterize the roll stability limits of vehicles under a steady lateral acceleration field arising from a steady turning maneuver.

Static Rollover Threshold (*SRT*)

The Static Rollover Threshold (*SRT*) refers to the limiting value of the steady lateral acceleration beyond which the vehicle cannot sustain a lateral perturbation and could experience rollover. The *SRT* has been widely adopted as a measure of rollover immunity of heavy vehicles, which directly relates to the relative rollover condition [39,73,74]. A lower limit in the range of 0.35-0.40g has been generally accepted for commercial freight vehicles [9,74]. Assuming a single degree-of -freedom, the *SRT* of a road vehicle can be estimated from:

$$SRT = \frac{T}{h} - \phi_0 \tag{3.1}$$

where *SRT* is the limiting value of lateral acceleration in 'g', T is half-track width, h is the cg height of the vehicle and ϕ_0 is the vehicle roll angle at the wheel lift off position [23]. The *SRT* at the relative rollover condition also corresponds to the lateral acceleration, when the rate of change of the lateral acceleration with respect to the sprung mass roll angle approaches a negative value.

The tilt table tests are widely used to obtain the *SRT* [37,74]. In this method, an articulated vehicle mounted on a tilt table platform is gradually tilted until relative rollover condition is realized. The tangent of the angle of inclination at which the vehicle experiences relative rollover condition yields the *SRT*. The *SRT* estimate however considers the variations in the cg height under increasing roll inclination. A number of analytical models and computer simulation tools have been developed for predicting the *SRT* of a vehicle combination as summarized in Chapter 1 [22, 27, 46, 74]. The effect of non-linearities of various vehicle sub-systems needs to be characterized to obtain an accurate value of *SRT* for a given vehicle. The *SRT* can however provide an estimation of the lateral acceleration that the fully loaded vehicle can withstand before experiencing a divergent roll response. Once the *SRT* of a vehicle is established through analytical or tilt-table tests, the lateral acceleration response could be constantly and easily monitored using accelerometers and compared with the *SRT* value to estimate the rollover propensity of the vehicle. The reliability of this measure in predicting the onset of a rollover under variations in design and operating conditions has been investigated in a few studies [36, 74].

Static Safety Factor (*SSF*)

The *SSF*, defined as the ratio of half-track width to *cg* height, provides a crude estimate of the rollover propensity of the vehicle, while assuming negligible contribution due to compliance of the suspension and tire. It has been shown that the *SSF* closely correlates with the actual rollover frequencies of road vehicles [41]. Although the half trackwidth remains constant during a steady turning roll process, the cg height of the vehicle can vary owing to varying loading conditions. On the basis of simulations performed for an articulated vehicle with different combinations of axle suspensions and tires, a compliance factor of 0.72 was proposed to account for compliance of the suspension springs and tires employed in tractor-semitrailer combinations [26], such that

$$SRT = c.\frac{T}{h} \tag{3.2}$$

where c is the compliance factor and the ratio T/h defines the *SSF*.

3.2.2 Measures of Dynamic Roll Instability

Articulated vehicles exhibit rollover instability at somewhat different levels of lateral accelerations in dynamic directional maneuvers. The roll response characteristics of different units of an articulated vehicle undertaking a dynamic maneuver exhibit considerable differences in amplitude and phase depending upon the nature of the maneuver [24]. The vehicle rollover occurs only when the lateral acceleration acts for a sufficient period of time to bring into effect the dynamic forces and moments. The static rollover threshold is thus considered inadequate under dynamic conditions [6]. A number of different measures have been defined to assess the dynamic roll instability of articulated vehicles with very little or no consensus on their general applicability.

Rearward Amplification Ratio (*RAR*)

Rearward Amplification Ratio (*RAR*), defined as the ratio of the peak lateral acceleration response of the trailer to that of the tractor, has been proposed to assess the relative roll performance of articulated vehicles under high speed steering maneuvers [36,39]. Articulated vehicles exhibit strong dependence on the rate or frequency of steering input. Figure 3.1 illustrates, as an example, the acceleration response time histories of the tractor and semitrailer sprung masses under a sinusoidal steering input. As is evident from the figure, the rearmost unit exhibits relatively higher response. The *RAR* of the combination is then computed as:

$$RAR = \frac{\left|a_{y2(peak)}\right|}{\left|a_{y1(peak)}\right|} \tag{3.3}$$

where $\left|a_{y2(peak)}\right|$ and $\left|a_{y1(peak)}\right|$ are the peak values of absolute lateral acceleration responses of the semitrailer and tractor respectively.

El Gindy [39] suggested that the *RAR* of a tractor-trailer combination undertaking a path change maneuver must not exceed 2.2. Liu [6] derived the peak lateral acceleration of the tractor and the *RAR* of the tractor-trailer combination, corresponding to a relative rollover condition for different suspension and steering frequencies to show their sensitivity to variations in suspension properties and rate of steer inputs. It was concluded that both the peak lateral acceleration and the *RAR* are maneuver sensitive. An increase in the steering frequency increases the *RAR* and the characterization of roll behavior through *RAR* requires a thorough knowledge and consideration of the maneuver.

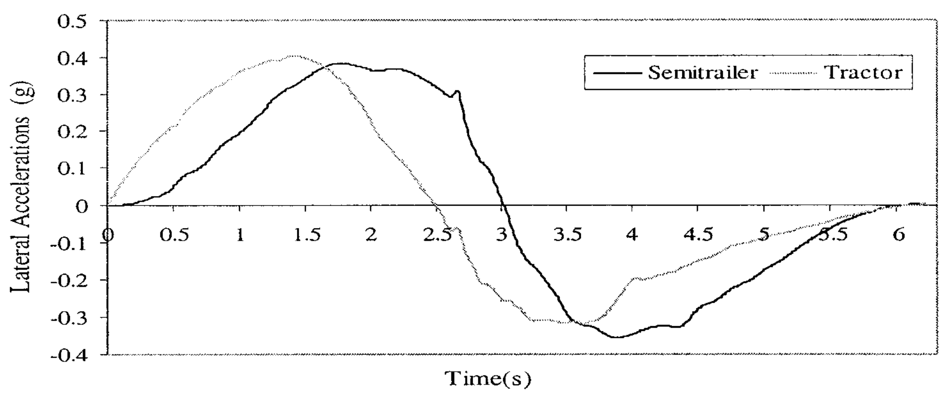


Figure 3.1: Acceleration responses of the tractor and semitrailer subjected to a sinusoidal steering input

Load Transfer Ratio (*LTR*)

The *LTR* measure based upon the lateral load transfer between the outboard and inboard tires has been proposed for assessing the dynamic roll stability limits of heavy vehicles [40]. The *LTR* is given by:

$$LTR = \left| \frac{\sum_{i=1}^{n} (FL_i - FR_i)}{\sum_{i=1}^{n} (FL_i + FR_i)} \right| \tag{3.4}$$

where, FL_i and FR_i are the vertical tire forces on the left and right track tires of axle i, and n is the total number of axles. A limiting value of 0.6 under a standardized path change maneuver at 100 km/h has been further recommended [39]. It has been suggested that the *LTR* could serve as a reliable indicator for impending rollover [40]. Considering that the *LTR* relates to the magnitude of load transfer of tires of all the axles, it does not directly relate to the relative roll instability condition. A unique value of *LTR* corresponding to relative rollover condition has not been identified. Furthermore, on-line measurement of *LTR* would involve considerable complexities and extremely high cost.

Roll Safety Factor (*RSF*)

The *RSF* measure, which directly relates to the relative rollover conditions for heavy vehicles, has been proposed as an extension to *LTR* [13]. The *RSF* is based on the ratio of load transfer of all axles with the exception of the first axle, which is considered to contribute only little to the net restoring moment. The *RSF* is given by:

$$RSF = \frac{\sum_{j=1}^{m}(FL_j - FR_j)}{\sum_{j=1}^{m}(FL_j + FR_j)} \tag{3.5}$$

where m is the number of axles that should experience loss of wheel-road contact in order to approach relative rollover condition. For a tractor-semitrailer combination, $m=n-1$. The dimensionless *RSF* measure attains a value of ± 1 as the vehicle approaches relative rollover condition, irrespective of the vehicle configuration, which makes it superior in terms of the reliability. The measure may thus be considered to be most effective for predicting relative rollover condition of different heavy vehicle combinations. Similar to *LTR*, a direct measurement of *RSF* is quite complex; an alternate measure that correlates well with the *RSF* and is easily measurable is thus highly desirable for developing an early warning roll control strategy.

Differential Wheel Slip (*DWS*)

The difference in slip between the left and right wheels, measured under application of short duration pulses to the braking system, has been proposed as a potential measure of the vehicle roll instability [8]. On the basis of the simulation results, it is suggested that *DWS* can predict the onset of a roll instability, when it exceeds a preset threshold value. Although the *DWS* can be easily measured from the *ABS* sensors, its reliability in relation to other defined measures of rollover is yet to be established. Moreover, a threshold value of this measure is not yet known.

Dynamic Rollover Threshold (*DRT*)

The rollover of heavy vehicles during a transient maneuver occurs at lateral acceleration levels lower than the *SRT* [38]. This acceleration value is termed as *DRT* [6], expressed in terms of effective lateral acceleration (*ELA*) and represented as:

$$ELA = \frac{\sum_{i=1}^{m}(FL_i - FR_i)T_i - \sum_{j=1}^{n_S} m_{sj} g \Delta_{sj}}{\sum_{j=1}^{n_S} m_{sj} h_{sj} + \sum_{i=1}^{n_U} m_{ui} h_{ui}} \tag{3.6}$$

where, m is the number of axles required to experience loss of wheel-road contact in order to approach relative rollover condition for the specific combination chosen, T_i is the half track width of axle i (i=1,..,m), m_{sj} is the sprung mass of the unit j and h_{sj} is its *cg* height, and n_s defines the number of sprung units of the combination. The unsprung mass of the i^{th} axle is represented by m_{ui} with its cg height h_{ui}, and n_u is the total number of axles. The lateral displacement of the sprung mass is represented by Δ_{sj}, which in turn is dependent on the suspension properties.

The *ELA* is based on the relative roll instability condition, however it is found to be sensitive to the steering frequency and the vehicle's operating conditions [6]. It has been shown that the *DRT* can provide either an over or under estimation of a potential roll instability depending upon the rate of steering and design and operating conditions [6,38]. Under most driving situations, the *DRT* is found to be quite close to the *SRT*, while it provides a poor measure of the yaw-induced rollover caused by rearward amplification tendencies [13,40].

Roll Angles of Sprung and Unpsrung Masses

The roll angle responses of sprung and unsprung masses of an articulated vehicle have also been proposed to detect the onset of roll instability under steady as well as transient steering maneuvers [6,26]. On the basis of the simulations performed for a tractor-semitrailer combination under a wide range of design and operating conditions, it was proposed that the unsprung mass roll angle, specifically that of the rearmost axle, correlates well with the *SRT* and thus the condition of relative roll instability. The roll angle responses alone, however, cannot be considered as reliable measures of the roll instability in the lower ranges as these can reach the threshold values without the vehicle approaching its relative rollover condition. A combination of the trailer sprung mass roll angle and *SRT* have also been proposed as a basis for a rollover warning system [26].

3.3 A PRELIMINARY ASSESMENT OF THE EXISTING ROLLOVER METRICS FOR SESNITIVITY ANALYSES

The effective detection of a rollover risk is highly dependent on the measure(s) of rollover employed at the detection stage. Thus it becomes imperative to assess the various rollover metrics in terms of performance measures that are pertinent in analyzing the effectiveness of the measure and validating its usage in the detection, warning and control algorithm. The effectiveness of a rollover metric strongly relies upon its ability to accurately predict the onset of roll instability in a timely manner over a range of variations in vehicle design and operating conditions. The prediction and control strategy would further necessitate direct measurement of the metric using low-cost and reliable sensors.

A few studies conducted on static and dynamic roll performance of heavy vehicles have concluded that the lateral acceleration, and sprung and unsprung mass roll angle responses vary considerably with variations in maneuvers undertaken, loading, suspension properties, roll center heights, etc. [25,36,70]. The lateral acceleration, however, can be easily measured using an accelerometer requiring simple signal conditioning. A poor correlation between the tractor lateral acceleration response, and the *LTR* and *RSF* has been observed under transient directional maneuvers [6]. This measure would thus be considered as less reliable for detection of onset of a roll instability. While the *DWS* measure offers monitoring with far greater ease, its correlation with *RSF* or onset of relative rollover has not yet been established and a reliable threshold value of *DWS* is not yet known. Of the proposed measures, the *RSF* is considered to be the most reliable for predicting the onset of roll instability based upon the relative rollover condition.

A unity value of *RSF* directly relates to the relative rollover condition, irrespective of the design, operating and environmental conditions considered. Owing to the extreme difficulties associated with its measurement, it would be desirable to identify alternate directly measurable indicators that correlate well with the *RSF*. The *LTR* reaching a unity value in a transient maneuver would more or less relate to the absolute rollover condition, which would make the induction of a warning and control system futile. However, the *RSF* reaching unity value translates into the attainment of relative roll instability criterion irrespective of the vehicle design, configuration or operating condition. The measure based on *RSF* is thus judged to be the most reliable indicator of relative roll instability for assessing the suitability of various measures.

The measurement of *ELA* requires prior knowledge of *cg* heights of the sprung and unsprung masses, and the measurement of wheel loads, which translates into poor measurability. The measures based upon roll deflections of sprung and unsprung masses offer superior measurability, which could be measured using micromachined inertial sensors and miniature gyroscopes. The developments in the area of micro-machined angular position sensors have facilitated accurate measurement of roll angles of both sprung and unsprung masses [58,62].

The detection of impending rollover is a two-step process, which involves the online measurement of a rollover indicator and the comparison of the obtained instantaneous value with a preset threshold value of the indicator. A rollover indicator can be considered to be reliable if its threshold value does not vary significantly in daily operations of the vehicle, and its threshold can be predicted with little estimation errors under different operating conditions. It needs to be emphasized that commercial vehicles undergo extreme variations in the payload and thus the axle loads, which would cause significantly different roll stability limits. Furthermore, these vehicles employ considerably different suspension and tires with varying restoring and dissipating properties.

On the basis of the briefly reported results on the measurability and sensitivity of various measures to variations in design and operating conditions, it can be concluded that only limited knowledge exists on the reliability of the existing measures of roll instability. The *RSF,* however, forms an exception as it directly relates to the relative rollover condition, irrespective of variations in the design and operating conditions. It can be further concluded that the trailer lateral acceleration and the sprung and unsprung mass roll angles offer relatively good measurability, while their threshold values corresponding to the unity value of *RSF* have not been established over a range of operating conditions. Comprehensive sensitivity analyses are thus needed to investigate the threshold values of these measures together with their reliability under arrange of design and operating conditions. Such an analysis would help to identify effective measures of relative roll instability for developing a reliable and effective early warning and control algorithm.

3.4 SUMMARY

The existing measures related to the relative roll instability condition of articulated vehicles, categorized into static and dynamic measures, are discussed. A preliminary assessment of *SRT, SSF*, roll angles of sprung and unsprung masses, *LTR, RSF, ELA* and *DWS* is presented in view of their perceived reliability and measurability. This assessment suggests that the sensitivity of these measures to variations in design and operating variables needs to be thoroughly examined in order to identify potentially usable measures. The lateral acceleration response of the semitrailer sprung mass as well as the roll deflections of the sprung and unsprung masses were found to provide good measurability as well as lead-time but their sensitivity to variations in design and operating conditions necessitates further analyses. It was further concluded that the *RSF* is the most reliable measure of the relative roll instability. Owing to its poor measurability, alternate metrics that correlate with the *RSF* would be most desirable. The *RSF* measure is thus taken as the basic reference for assessing the different measures for predicting the onset of a rollover.

CHAPTER 4

PARAMETRIC SENSITIVITY ANALYSES OF ROLLOVER INDICES

4.1 INTRODUCTION

A review of existing measures of impending rollover indicators revealed that most of the measures would be sensitive to variations in design and operating conditions of the vehicle and thus cannot be considered reliable. A few studies have reported the influence of design variations on the roll stability limits of commercial vehicles [25, 35, 37, 47, 48, 49, 75, 41]. El Gindy and Hosameldeen [70] identified certain parameters that influence the static rollover threshold of tractor-semitrailers, these include track width, cg. heights of sprung masses, suspension spring rates, etc. However, an extensive analysis of the influence of variations in design and operating conditions on the roll stability of articulated vehicles subjected to both steady as well as transient maneuvers is required to assess the relative reliability of various measures. The identification of the crucial design parameters, which can reduce or improve the roll stability, will further help in understanding the effect of these parameters on roll stability of the vehicle and establish the design goals within the statutory limits. The sensitivity of the measures of roll stability to the vehicle design parameters also needs to be understood in order to establish a reliable measure of rollover that can be considered generic. Considering that the articulated vehicles when fully loaded, exhibit lower stability limits, the sensitivity analysis of different measures should be effectively performed under the conditions of rated loads. The response of the vehicle and suitability of a particular measure needs to be studied under varying loading conditions in order to understand the effect of loading on the response parameters and minimize the possible occurrences of false detections.

The sensitivity analysis under variations in design and operating conditions could help identify vital response vectors that can, either singularly or in conjunction with other measures of roll instability, form a rollover metric that provides better lead-time and proves to be an effective rollover metric in-terms of measurability and reliability. In this chapter a baseline 5-axle tractor semi-trailer vehicle is considered to perform the sensitivity analysis of the rollover indices. The essential design parameters are identified together with the range of variations to formulate the simulation matrix. The simulations are performed to derive the effectiveness of different measures in the static as well as dynamic rollover threshold tests under ramp and sinusoidal steer inputs. The results obtained from the simulations are analyzed to study the sensitivity of the measures to variations in selected design and operating conditions. The results are discussed in view of the suitability of different measures of roll instability for applications in design of an early warning device.

4.2 BASELINE VEHICLE

A five-axle tractor semi-trailer combination is chosen as the baseline vehicle, which comprises a three-axle tractor and a two-axle semi-trailer. The choice of this configuration is supported by its relatively higher population in the freight transportation sector. It has been reported that approximately 77% of all the heavy commercial vehicles in Canada were tractor semi-trailers [68]. Furthermore, the tractor semi-trailer combinations are most frequently involved in highway accidents; nearly 60% of the fatal accidents have been associated with this vehicle combination, which in-part is attributed to its relatively high population [77]. The vehicle is considered to be loaded as per the maximum load limit prescribed by the Ministère des Transport du Québec [78]. The baseline vehicle is described in terms of number of axles, tandem spread, sprung and unsprung masses and mass moments of inertia of the constituent units, track widths, wheelbases of tractor and semi-trailer, cg heights of sprung and unsprung masses, roll center heights, articulation parameters, suspension parameters, tire properties, etc. [6, 25, 26, 36, 47, 77, 79], which are summarized in Tables 4.1 to 4.5. The components of articulated vehicles are further described in this section. The components are classified into geometric dimensions, masses and mass moments of inertia, suspension components, steering characteristics and tire properties.

4.2.1 Design Parameters

The salient geometric parameters that influence the directional dynamics of a tractor semi-trailer are trackwidth of the vehicle, its wheel base, the longitudinal and vertical location of the hitch, the suspension lateral spread, the tandem spread of the axles, axles locations and dual tire spacing. The nominal values of these parameters for the baseline vehicle are summarized in Table 4.1. The inertial properties of the sprung and unsprung masses of the combination are presented in Table 4.2. These include: the sprung and unsprung masses, axle loads, the cg locations and the roll, yaw and pitch moments of inertia of the sprung masses and the roll mass moment of inertia of the unsprung masses.

Table 4.1 : Geometric Parameters of the Baseline Vehicle

Geometric Parameters	Tractor			Semi-Trailer	
Wheelbase (m)	5.72			12.5	
Tandem axle spread (m)	1.52			1.52	
Longitudinal location of articulation from cg (m)	-3.0			5.98	
Articulation Height (m)	1.22			1.22	
Axles →	**Axle 1**	**Axle 2**	**Axle 3**	**Axle 4**	**Axle 5**
Half-Trackwidth(m)	1.08	0.75	0.75	0.75	0.75
Half-suspension spread(m)	0.406	0.482	0.482	0.482	0.482
Axle location from cg(m)	-1.52	2.67	4.19	4.98	6.50
Dual tire spacing(m)	0.0	0.33	0.33	0.33	0.33

Table 4.2: Inertial Properties of the Baseline Vehicle

Inertial Parameters	Tractor			Semi-Trailer	
Sprung mass(kg)	5353			31890	
cg height(m)	1.11			2.06	
Roll mass moment of inertia(kgm^2)	2938.0			26996.87	
Yaw mass moment of inertia(kgm^2)	18645.0			593900.80	
Pitch mass moment of inertia(kgm^2)	18645.0			603864.0	
Axles →	**Axle 1**	**Axle 2**	**Axle 3**	**Axle 4**	**Axle 5**
Axle load (kN)	54.95	89.92	89.92	89.92	89.92
Roll mass moment of inertia (kgm^2)	418	576.3	576.3	464.2	464.2
Unsprung weight (kN)	5.025	10.328	10.328	8.278	8.278
Unsprung mass cg height (m)	0.49	0.51	0.51	0.51	0.51

The roll dynamics of an articulated vehicle is greatly dependent on the choice of suspension system. A hard suspension reduces the lateral load transfer of the vehicle during a directional maneuver by limiting the roll deflection, and thus yields higher rollover threshold of the vehicle. A hard suspension however could induce higher dynamic tire loads to the pavement and yield poor ride performance. Auxiliary roll stiffeners are thus employed to realize a better compromise between the ride and roll performances [79]. While considering the roll dynamics of a vehicle, the composite roll stiffness proves to be a contributing factor that directly influences the roll stability of the vehicle. The composite roll stiffness depends on the individual spring rates, lateral spring spread and auxiliary roll stiffness of the vehicle [79]. The suspension properties of a vehicle are expressed in terms of vertical suspension stiffness or force-deflection characteristics, damping ratio, roll center heights above ground, auxiliary roll stiffness, roll steer coefficient and suspension coulomb friction. The force deflection properties of the tractor front and rear, and trailer axle suspension springs are illustrated in Figures 4.1(a) to 4.1 (c) respectively.

Table 4.3: Suspension Parameters (Axle wise)

Parameter	**Axle 1**	**Axle 2**	**Axle 3**	**Axle 4**	**Axle 5**
Roll center height(m)	0.58	0.74	0.74	0.74	0.74
Auxiliary roll stiffness(Nm/rad)	24760	194249	550373	388200	388200
Damping (Ns/m)	4904	14886.0	14886.0	14886.0	14886.0
Suspension coulomb friction(kg)	2107	1563	1563	1612	1612

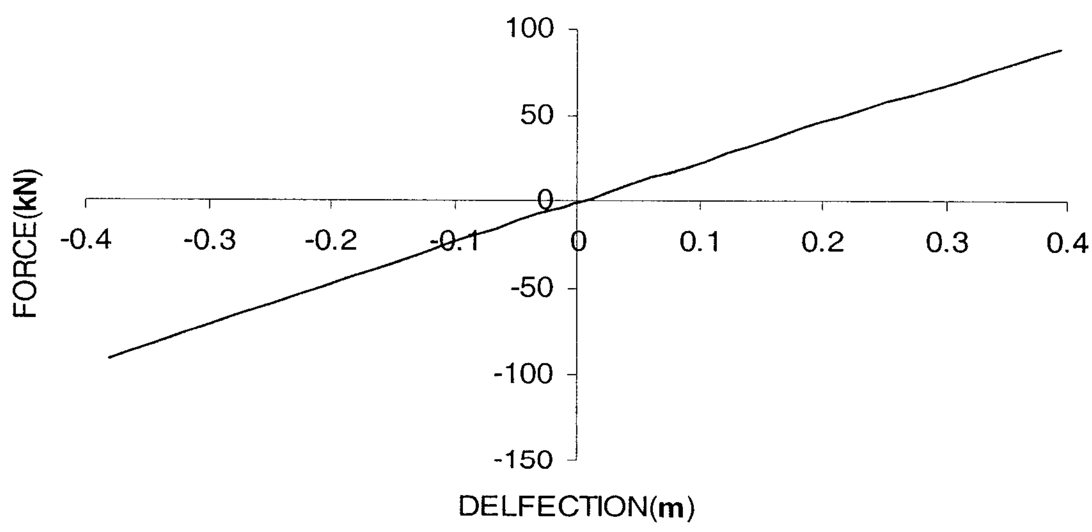


Figure 4.1(a): Force-Displacement relationship for the tractor front axle suspension springs (International Harvester- Leaf) [80] .

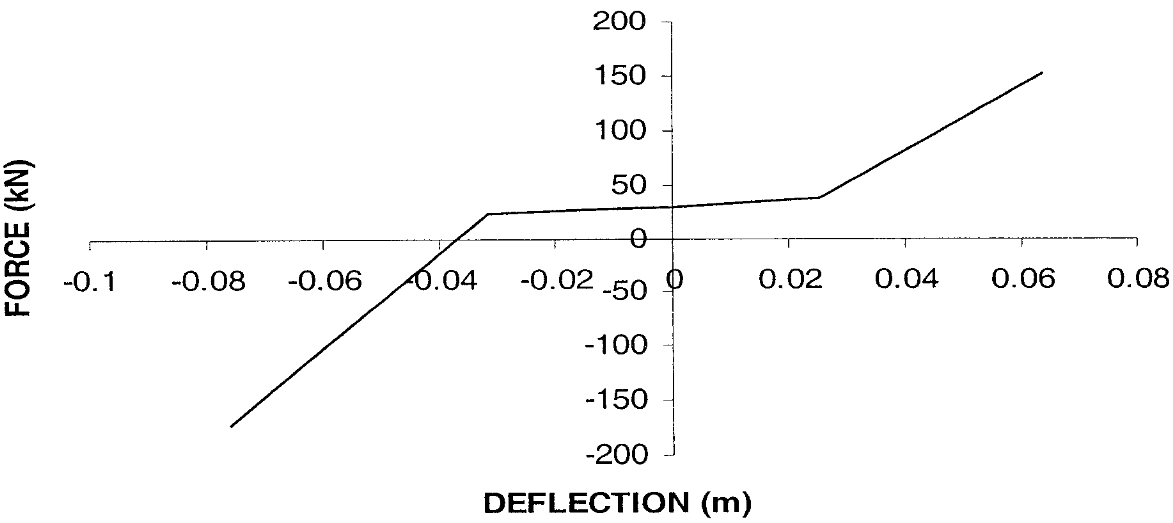


Figure 4.1(b): Force-Displacement relationship for the tractor rear axle suspension springs (Neway ARD 244-Air) [79]).

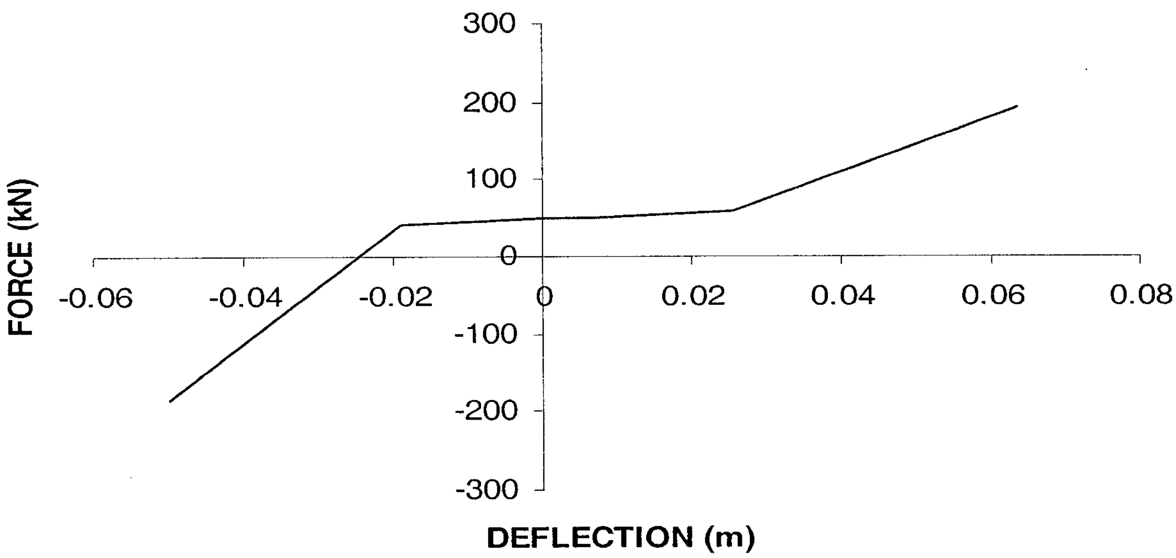


Figure 4.1(c): Force-Displacement relationship for the semitrailer axle suspension springs (Neway AR 95-17- Air) [79]).

The handling properties of the vehicle pertinent for analyzing the roll stability of a vehicle include the steering gear ratio, steering system stiffness, tie rod stiffness, mechanical trail and roll steer coefficients of different axles. These parameters for the baseline vehicle are summarized in Table 4.4.

Table 4.4: Handling parameters for the baseline vehicle

Parameter	Value
Steering gear ratio	25
Steering system stiffness (kNm/rad)	698.0
Tie rod linkage stiffness (kNm/rad)	698.0
Mechanical Trail (m)	0.0254
Roll Steer Coefficient (Axles1 to 5)	0.0, 0.22, 0.23, 0.23, 0.23

The directional dynamic behavior of a vehicle is strongly influenced by the traction, cornering and aligning properties of tires. For the constant speed analyses considered in this study, the cornering and self-aligning properties of tires are of utmost importance, which are non-linear functions of the side-slip angle and the normal load. The simulation program used in this study utilizes three-dimensional look-up tables to compute both the cornering force and self-aligning moments corresponding to the slip angle and tire normal load. Figures 4.2 (a-b) illustrate the cornering and self-aligning properties of a heavy vehicle tire (11R22.5) as functions of the side slip angle and normal load. Apart from the cornering properties of the tires, the tire vertical stiffness also contributes to the effective roll stiffness. The vertical stiffness of the tires is chosen to be 1050kN/m corresponding to an inflation pressure of 758.5 kPa.

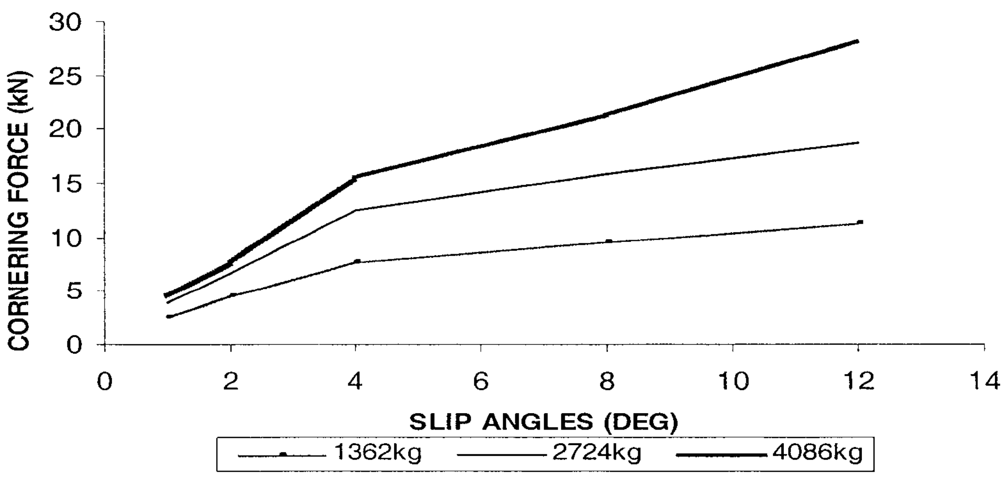


Fig 4.2(a): The cornering force property of the selected tire for the baseline vehicle.

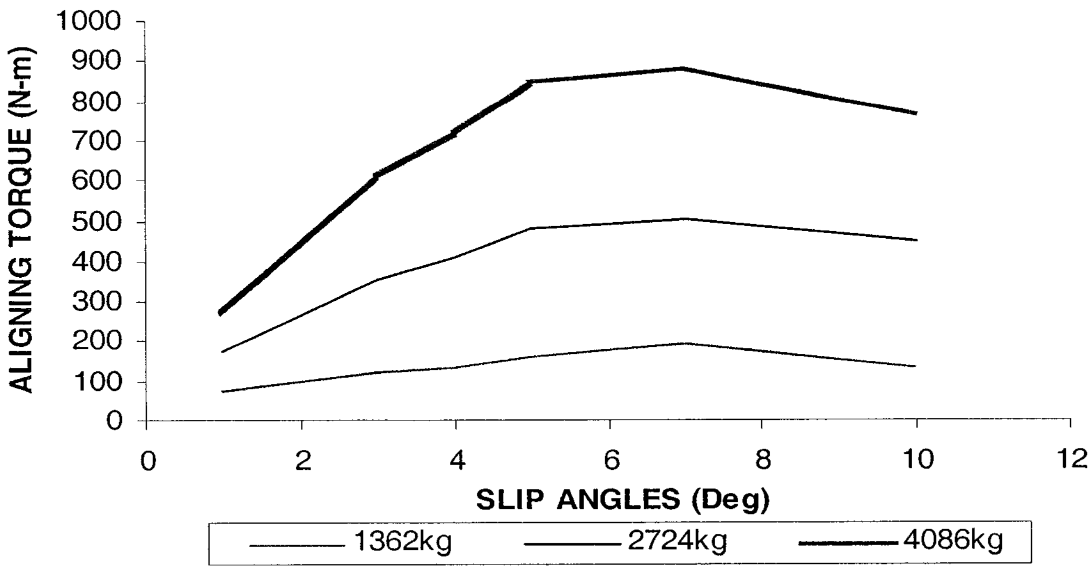


Fig 4.2(b): The aligning torque property of the selected tire for the baseline vehicle.

4.3 SIMULATION MATRIX FOR PARAMETRIC VARIATION ANALYSIS

The simulation parameters described in the previous section correspond to the baseline vehicle, which is considered to be fully loaded in accordance with MTQ weights and dimensions regulations. Furthermore, the tires and suspension properties are chosen for commonly used 11R22.5 tires, and axle suspensions (IH reference for front axle, Neway ARD 244 Air for tractor rear axles and Neway AR 95-17 Air for semitrailer axles). The design and operating parameters of different vehicles in service, however, may vary considerably from the baseline vehicles. The loading practices and nature of cargo could significantly influence the inertial properties, cg locations, and suspension and axle loads. The variations in tire inflation properties and tire wear may affect the vertical stiffness and cornering properties of tires. The vehicle operators may also choose different suspensions depending upon the preferences, cost and load requirements. Moreover, considerable variations in the operating conditions, such as speed and nature of steering can also be expected. Considering that the roll dynamic responses of the vehicle strongly rely upon many of these parameters, the identification of a reliable rollover metric would necessitate systematic analyses over wide ranges of design and operating conditions that may be encountered in practice.

In order to obtain a holistic perspective of the effect of design and operating conditions variations, each parameter should be varied within a range that spans positive as well as negative increments of the rating of the component considered. A comprehensive simulation matrix thus needs to be formulated, where each element of the matrix would represent variations in a particular design and operating condition, and would yield the desired responses for sensitivity and reliability analyses of the rollover metrics.

4.3.1 Maneuvers for Analysis of Static and Dynamic Roll Instability

The directional response characteristics are known to be strongly dependent on the maneuver, such as speed, the magnitude and rate of steering, and intensity of braking. Various analytical and experimental studies performed on assessment of directional performance of heavy vehicles have evolved into a set of recommended maneuvers. Fancher [47,79] and El Gindy [39,70] have proposed different measures to perform relative assessment of vehicles in view of static and dynamic roll properties, yaw dynamics and braking responses. It has been suggested that static rollover immunity levels of heavy vehicles can be obtained by subjecting the vehicle to a low rate of ramp steer input at a steady speed of 100 km/h and a steering wheel angle of two degree per second which is considered to represent a steady turning maneuver. A 2.2m (approx.) path change maneuver at a forward speed of 100 km/h has been recommended to assess the dynamic roll and yaw directional performance of the vehicle in terms of lateral load transfer ratio (*LTR*) and rearward amplification (*RA*). The recommended maneuver, however, does not consider the rate of steering that may vary with the gate length used by the driver.

Alternatively, Lam [80] suggested that sinusoidal steering functions could effectively represent lane change and evasive maneuvers, where the frequency of input would represent the steering rate. The steering frequencies of 0.25Hz, 0.33Hz and 0.50Hz are considered to cover the range of steering inputs that drivers of articulated vehicles could provide while undertaking evasive or lane change maneuvers. The sinusoidal maneuvers considered to represent open-loop path change maneuver are thus expressed as:

$$\delta_{sw} = \delta_{amplitude} \cdot \sin(2\pi f t) \qquad 0 \le t \le 1/f \tag{4.1}$$

where δ_{sw} is the steering wheel angle, $\delta_{amplitude}$ is the amplitude of the steering angle and f is the frequency of the steering input. The static as well as dynamic roll response under the influence of the said maneuvers can be studied using the Constant Velocity Yaw/Roll model, which is described in Chapter 2. The roll instability condition is obtained by gradually increasing the vehicle speed until the relative rollover condition is realized for the given design and operating condition.

4.3.2 Simulation Matrix

The sensitivities of various rollover metrics to variations in design and operating parameters are investigated through simulations over a wide range of parametric variations. The simulation matrix is formulated upon consideration of variations in vehicle design parameters, operating conditions, maneuvers, sprung weights and cg heights. The simulations are performed on the basis of the proposed matrix to assess both the static and dynamic rollover thresholds and relative rollover condition. Variations in the design parameters include varying tire stiffness, tire trackwidth, articulation roll stiffness, suspension lateral spread, axle tandem spread, suspension stiffness and damping, auxiliary roll stiffness, roll center height and tire cornering stiffness. All the design variations are considered in conjunction with simultaneous variations in the cg height, which is considered to be most significant factor affecting static and dynamic roll stability.

Table 4.5 summarizes the simulation matrix comprising variations in design parameters including: tire stiffness and cornering properties, trackwidth, articulation roll stiffness, suspension stiffness, damping ratio, suspension spread, roll center height, tandem spread. The table also lists the ranges of operating parameters considered in the simulation. These include the variations in the operating speed and payload, apart from the steering inputs as discussed in section 4.3.1. Each simulation run represents a particular variation in a design or operating condition, while all the other parameters are left unchanged. The design and operating condition of the baseline vehicle is regarded as the baseline configuration, and each design or operating parameter is changed relative to this configuration.

For each parametric variation in design and operating condition, three cg heights are considered for static rollover analysis, which involves the vehicle being subjected to a 2°/s ramp steer input at an operating speed of 100 km/h. This resulted in 90 simulation runs considering the 30 different parametric variations, with no simulations performed for variation in operating velocity, as the static rollover analysis is studied for constant velocity ramp-steer maneuvers. Each variation further involves 3 different cg heights (1.8m, 2.0m, 2.2m).

Table 4.5: Simulation Matrix for Parametric Sensitivity Analyses

PARAMETER	VARIATION	SIMULATION RUNS	
		STATIC	DYNAMIC
Baseline Vehicle	X	1-3	1-9
Trackwidth	2.59m	4-6	10-18
Axle Spread	1.22m	7-9	19-27
	1.83m	10-12	28-36
Tire Stiffness	787 kN/m	13-15	37-45
	875 kN/m	16-18	46-54
	972 kN/m	19-21	55-63
Articulation Point Stiffness	4800 kNm/rad	22-24	64-72
	7200 kNm/rad	25-27	73-81
Suspension Spread	0.57m (TF), 0.66m (AO)	28-30	82-90
Damping Ratio	0.025	34-33	91-99
	0.05	37-36	100-108
	0.1	40-39	109-117
Roll Center Height	0.60m (ST)	43-42	118-126
	0.88m (ST)	46-45	127-135
	0.46m (TF), 0.60m (TR)	49-48	136-144
	0.70m (TF), 0.88m(TR)	52-51	145-153
Auxiliary Roll Stiffness	155kNm/rad (TR1), 440kNm/rad(TR2)	55-54	154-162
	233kNm/rad (TR1), 660kNm/rad(TR2)	58-57	163-171
	311kNm/rad (ST)	61-60	172-180
	466 kNm/rad (ST)	64-63	181-189
Suspension Stiffness	Semi-Trailer Baseline-20%	67-66	190-198
	Semi-Trailer Baseline+20%	70-69	199-207
	Reyco (ST)	73-72	208-216
	Tractor Rear-20%	76-75	217-225
	Mack (TR)	79-78	226-234
Tire Cornering Stiffness	Dry Radial	82-81	235-243
	½ Tread Radial	85-84	244-252
	1/3 Tread Radial	88-87	253-261
Operating Load	50% Payload	88-90	262-270
Operating Speed	Critical Rollover Speed+20km/h	X	271-279
Abbreviations: TF= *Tractor Front Axle;* AO= *All Other Axles;* ST= *Semi-Trailer Axles;* TR=*Tractor Axles;* TR1= *Tractor Rear Axle;* TR2= *Tractor Rearmost Axle*			

For dynamic roll analyses, three steering different frequencies are considered (0.25,0.33 and 0.50Hz) for each parametric variation which when coupled with 3 cg heights (1.8m, 2.0m and 2.2m respectively), result in 9 simulation runs for every parametric variation. A total of 31 combinations of design and operating conditions considered for dynamic rollover analyses yield a total of 279 simulation runs.

Each simulation run corresponds to a given variation in the design or operating parameter along with a variation in steering frequency and cg height. The simulation run numbers corresponding to a given variation in design or operating condition are presented in the table. The simulations are performed to achieve the relative rollover condition of the vehicle, and the corresponding values of the selected rollover metrics. The relative rollover condition corresponding to dynamic rollover tests under sinusoidal maneuvers is identified by gradually increasing the vehicle speed until the condition is achieved which is characterized by the *RSF* reaching unity value.

4.4 EFFECTS OF VEHICLE DESIGN AND OPERATING PARAMETERS ON ROLL DYNAMICS OF ARTICULATED VEHICLES

The Constant Velocity Yaw/Roll Model developed by UMTRI was used to derive the roll dynamic responses of the articulated vehicle corresponding to each parametric variation. The simulation response characteristics are further evaluated to derive the responses in terms of different rollover metrics or measures, such as lateral accelerations of the tractor and semi-trailer, sprung mass roll angles of the tractor and semi-trailer, unsprung mass roll angles of different axles, the load on each wheel of the vehicle, load transfer ratio and roll safety factor. Considering that the rearward amplification tendency of a vehicle is highly dependent on the vehicle configuration and the maneuver [6] this metric has been excluded from the parametric analysis. The simulation results attained under steady ramp and dynamic sinusoidal maneuvers are discussed to emphasize the reliability of sensitivity of both the static and dynamic rollover metrics.

4.4.1 Static Rollover Threshold Tests

The static rollover threshold limits are characterized by simulating a 2°/s constant velocity ramp steer input at a speed of 100 km/h. The simulations are performed until a given vehicle configuration approaches its relative rollover condition, or when the sprung mass roll angle reaches 0.5 radians. The static rollover threshold is obtained by analyzing the roll performance signature, which describes the variation in the lateral acceleration of the semi-trailer unit to changes in the roll angle of the semi-trailer sprung mass. The lateral acceleration (a_{y2}) generally increases with increase in roll angle (φ_{s2}), and approaches its maximum value corresponding to the relative rollover condition. The a_{y2} then tends to decrease since the restoring moment developed by the front axle alone is insufficient to overcome the primary overturning moment, as is evident from Figure 4.3. The static rollover threshold of the vehicle is thus identified when slope of the roll performance signature approaches a negative value ($\frac{\partial a_{y2}}{\partial \varphi_{s2}} < 0$).

Figure 4.3 illustrates the dynamic responses of the baseline vehicle to a ramp steering input. The figure shows the time histories of lateral acceleration (a_{y2}) and roll angle (φ_{s2}) of the semitrailer sprung mass, roll signature (a_{y2} vs. φ_{s2}), unsprung mass roll response and vertical tire loads. The results show that the relative rollover condition, associated with loss of road contact of tires on the inner track of the axles 2 to 5, is attained near t = 25.63 s. This condition corresponds to peak lateral acceleration (a_{y2}) and $\frac{\partial a_{y2}}{\partial \varphi_{s2}} < 0$, as shown in the figure. Furthermore, the magnitudes of roll angle responses of the sprung and unsprung masses increase abruptly as the relative rollover condition is realized.

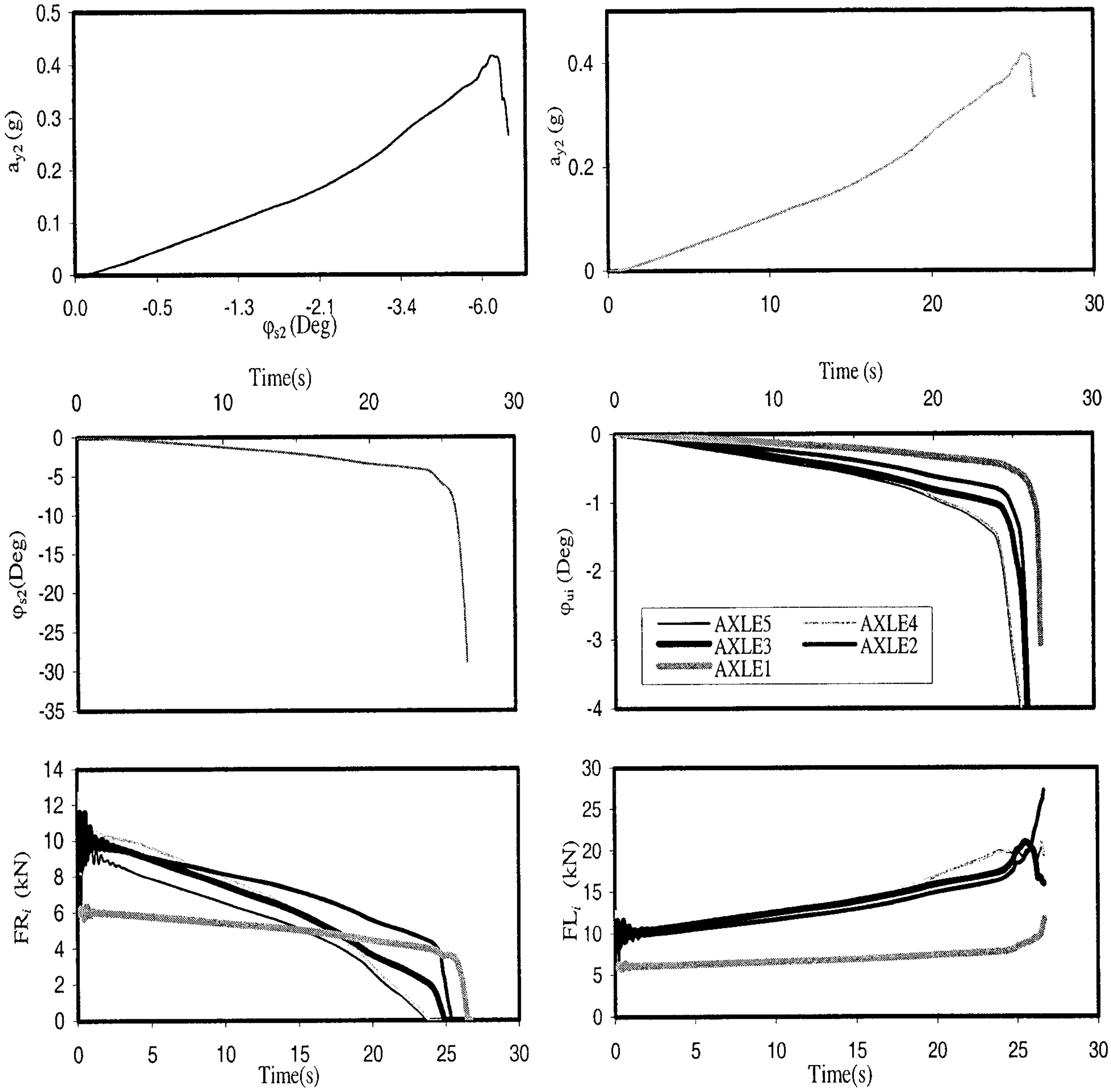


Fig 4.3: The response parameters to the static rollover threshold test for the baseline vehicle with center of gravity height of 2.0m.

The lateral acceleration threshold, and the sprung and unsprung mass roll angles corresponding to the relative rollover condition may be considered as possible rollover metrics for predicting the onset of a roll instability. The sensitivities of these measures (a_{y2} and φ_{s2}) to variations in design and operating conditions are evaluated to assess their reliability for applications in the open-loop roll control strategy. Tables 4.6 (a-c) represent the variations in static rollover threshold (*SRT*) and semi-trailer sprung mass roll angle (φ_{s2}) with respect to variations in geometric and design parameters, tire and suspension parameters, and operating load, respectively.

Table 4.6 (a): Effect of variations in geometric and design parameters on ***SRT*** and φ_{s2}

Vehicle Configuration		***SRT (g)***			φ_{s2} ***(Deg)***		
		cg=1.8m	**cg=2.0m**	**cg=2.2m**	**cg=1.8m**	**cg=2.0m**	**cg=2.2m**
Baseline Vehicle		0.509	0.417	0.385	6.278	6.603	6.697
AxleTandem Spread	**1.22 m**	0.514	0.418	0.385	6.262	6.560	6.661
	1.83 m	0.504	0.417	0.386	6.328	6.634	6.716
Artn. Roll Stiffness	**4800kNm/rad**	0.508	0.417	0.384	6.380	6.737	6.860
	7200kNm/rad	0.509	0.417	0.386	6.217	6.489	6.607
Trackwidth	**2.59m**	0.571	0.469	0.424	5.538	6.609	6.757

Table 4.6 (b): Effect of variations in tire properties on ***SRT*** and φ_{s2}

Vehicle Configuration		***SRT (g)***			φ_{s2} ***(Deg)***		
		cg=1.8m	**cg=2.0m**	**cg=2.2m**	**cg=1.8m**	**cg=2.0m**	**cg=2.2m**
Baseline Vehicle		0.509	0.417	0.385	6.278	6.603	6.697
Tire Vertical Stiffness	**972 kN/m**	0.504	0.415	0.383	6.424	6.735	6.817
	875 kN/m	0.500	0.412	0.378	6.595	6.877	6.989
	787 kN/m	0.495	0.407	0.383	6.791	6.490	7.188
Tire Cornering Stiffness	**Radial-Dry**	0.493	0.417	0.385	6.288	6.581	6.699
	Radial- ½Tread	0.486	0.417	0.384	6.325	6.597	6.695
	Radial-1/3Tread	0.467	0.416	0.383	6.134	6.583	6.681

Table 4.6 (c): Effect of variations in suspension properties on ***SRT*** and φ_{s2}

Vehicle Configuration		***SRT (g)***			φ_{s2} ***(Deg)***		
		cg=1.8m	**cg=2.0m**	**cg=2.2m**	**cg=1.8m**	**cg=2.0m**	**cg=2.2m**
Baseline Vehicle		0.509	0.417	0.385	6.278	6.603	6.697
Susp. Spread= Baseline+20%		0.526	0.432	0.398	5.536	5.790	5.855
Damping Ratio	**0.1**	0.509	0.418	0.386	6.287	6.584	6.703
	0.05	0.51	0.416	0.386	6.301	6.616	6.700
	0.025	0.512	0.418	0.386	6.287	6.576	6.705
Aux. Roll. Stiffness.	**Baseline(ST)-20%**	0.509	0.415	0.383	6.284	6.600	6.704
	Baseline(ST)+20%	0.509	0.419	0.386	6.286	6.600	6.691
	Baseline(TR)-20%	0.495	0.412	0.382	6.020	6.680	7.010
	Baseline(TR)+20%	0.515	0.420	0.388	6.180	7.020	7.310
Suspension Stiffness	**Baseline(ST)-20%**	0.505	0.416	0.384	6.298	6.594	6.719
	Reyco (ST)	0.512	0.415	0.382	6.296	6.596	6.696
	Baseline(ST)+20%	0.513	0.417	0.384	6.322	6.568	6.688
	Mack (TR)	0.500	0.420	0.382	6.466	6.949	7.220
	Baseline(TR)-20%	0.489	0.415	0.383	6.710	6.990	7.107
Roll Center Height	**Baseline(ST)-20%**	0.497	0.412	0.379	6.311	6.607	6.718
	Baseline(ST)+20%	0.518	0.420	0.389	6.270	6.589	6.682
	Baseline(TR)-20%	0.494	0.403	0.377	6.491	6.781	6.853
	Baseline(TR)+20%	0.522	0.425	0.394	6.179	6.422	6.559

Table 4.6 (d): Effect of variations in operating load on ***SRT*** and φ_{s2}

Vehicle Configuration		***SRT (g)***			φ_{s2} ***(Deg)***		
		cg=1.8m	**cg=2.0m**	**cg=2.2m**	**cg=1.8m**	**cg=2.0m**	**cg=2.2m**
Operating Load	**Fully Loaded**	0.509	0.417	0.385	6.278	6.603	6.697
	50% Payload	0.526	0.458	0.429	4.155	4.510	4.621

For a non-compliant vehicle, the *SRT* value is given by the ratio of half-track width to the sprung mass *cg* height. The *SRT* is thus directly dependent on the trackwidth as can be observed from Table 4.6(a). The compliant properties of the suspension, tires and articulation mechanism also influence the *SRT* and the sprung mass roll angle response of the semitrailer, as is evident from Tables 4.7 (a-c). Considering the baseline vehicle, fully loaded as per *MTQ* guidelines, the *SRT* values are attained as 0.509g, 0.417g and 0.385g for cg heights of 1.8m, 2.0m and 2.2, respectively; while the roll angles are 6.278, 6.603 and 6.697 degrees for cg heights of 1.8m, 2.0m and 2.2m respectively. Figures 4.4 to 4.7 further illustrate the effects of the parametric variations on the SRT and φ_{s2}.

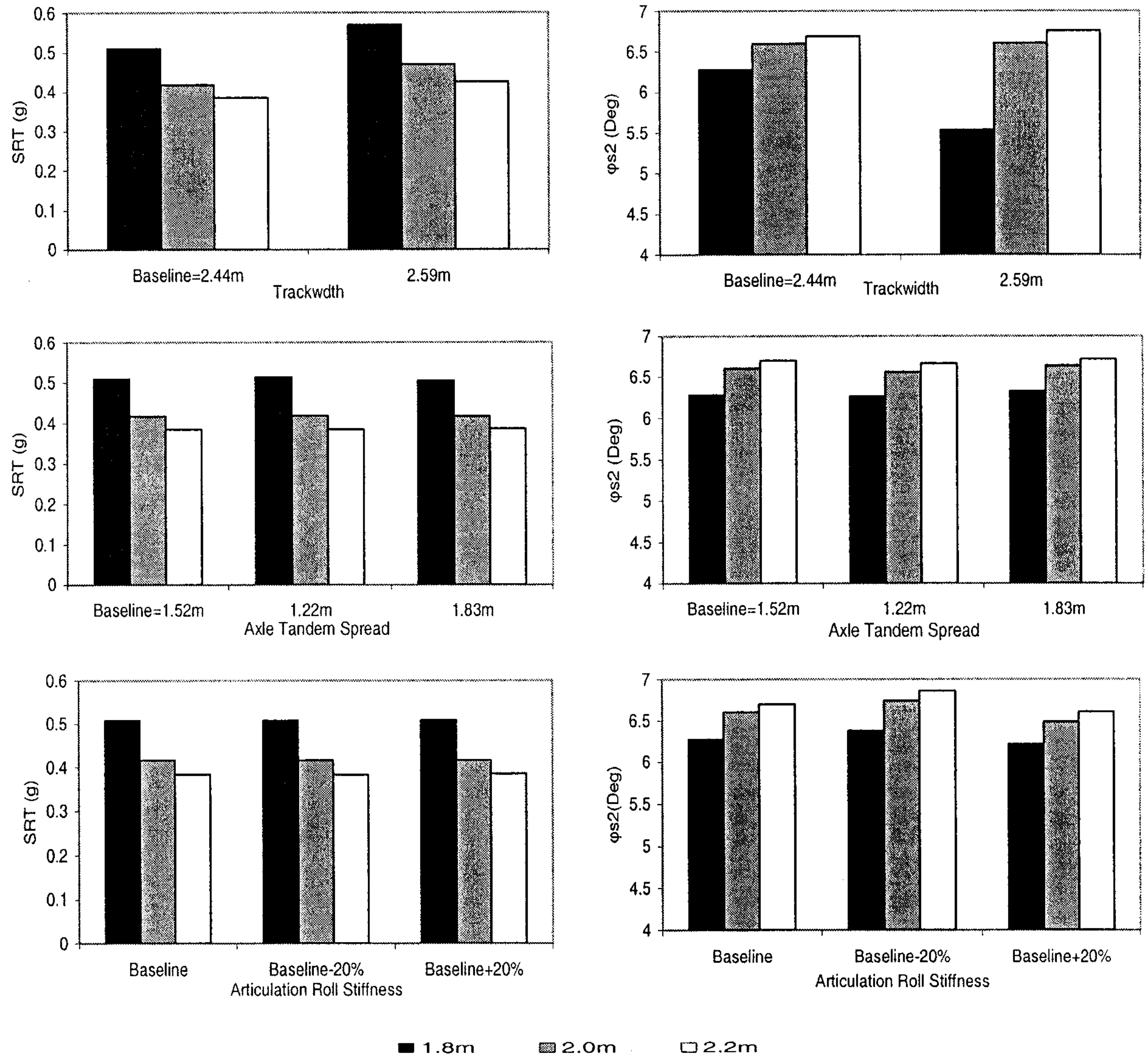


Fig 4.4: Influence of variations in trackwidth, axle spread and articulation roll stiffness on SRT and φ_{s2}.

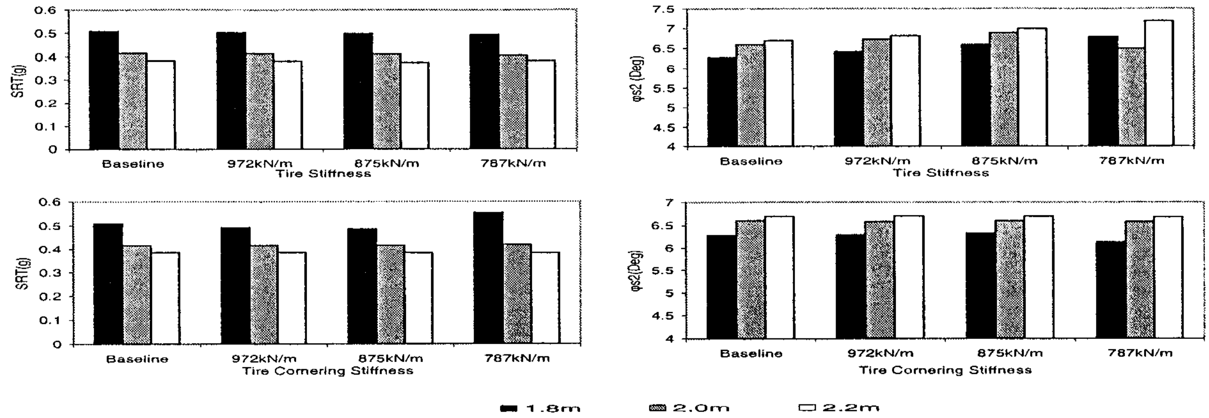


Fig 4.5: Influence of variations in tire vertical stiffness and tire cornering stiffness on *SRT* and φ_{s2}.

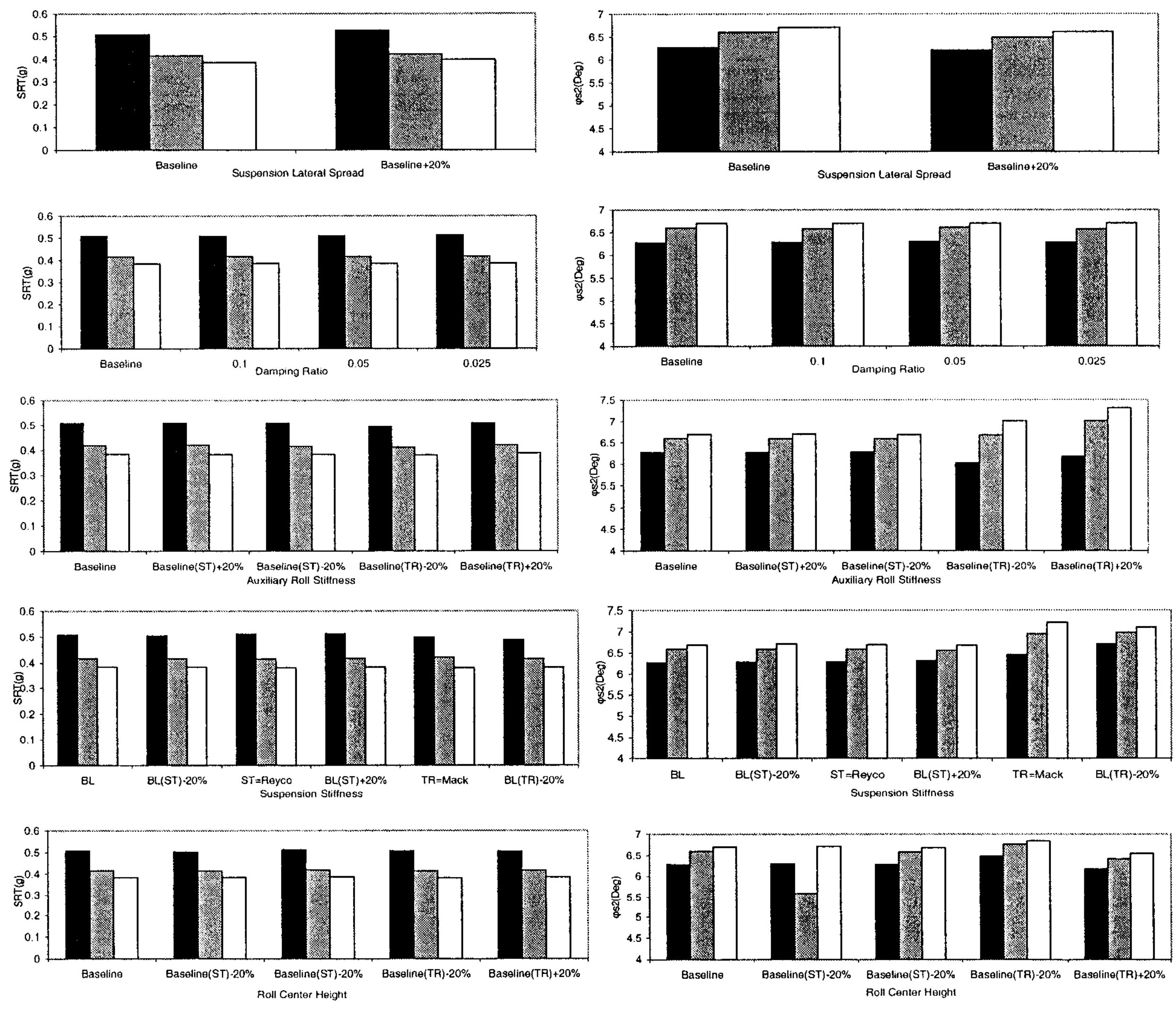


Fig 4.6: Influence of variations in lateral suspension spread, damping ratio, auxiliary roll stiffness, suspension stiffness and roll center height on *SRT* and φ_{s2}.

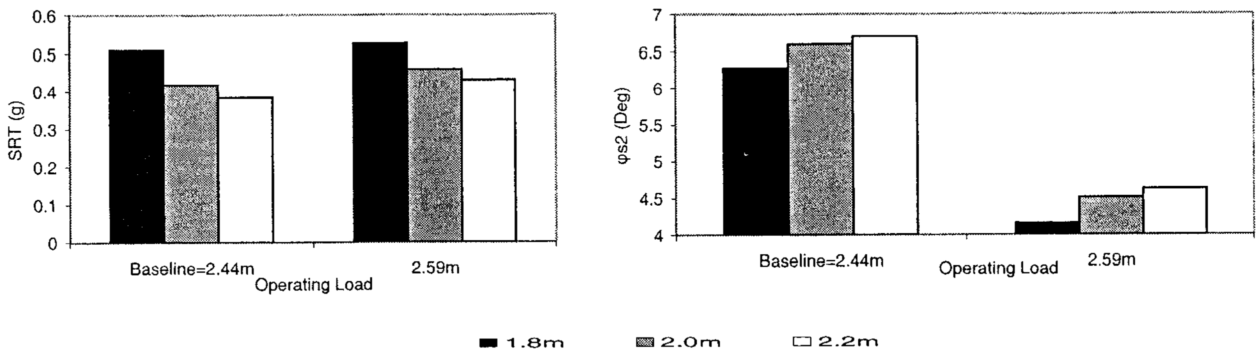


Fig 4.7: The response of *SRT* and φ_{s2} to variations in operating load

The results of simulation presented in Table 4.6 and Figures 4.4 to 4.7 show that most of the design as well as operating variations affect the static roll stability of the baseline vehicle significantly. It can be inferred that the increase in cg height decreases the roll stability of the vehicle and that the roll angle at relative roll instability condition increases with increase in cg height, although the static roll threshold decreases. Some design parameters affect the static roll stability of the vehicle more than others and these parameters are considered to be crucial to enhance the roll stability of articulated vehicles through modifications in design. These parameters are discussed in detail and the effect of variations of these parameters on the dynamic roll stability of the vehicle is presented in the subsequent sections.

4.4.2 Parametric Analysis Based on Static Rollover Threshold Test

Influence of Vehicle Trackwidth: The restoring roll moment developed by the vehicle and thus the static rollover threshold acceleration response is directly related to the vehicle trackwidth. Two values of trackwidth, 2.44m (baseline vehicle) and 2.59m (variation), as permitted by the road regulations are considered. Since the cg height is the most significant factor affecting the *SRT*, the simulation results are attained for variations in both cg height and the trackwidth. The results show that the increase in trackwidth from 2.44 m to 2.59 m has a significant effect on the *SRT* of the vehicle. An increase in trackwidth of 0.15m increases the SRT of the baseline vehicle by 12.18%, 12.47% and 10.12% respectively for cg heights of 1.8m, 2.0m and 2.2m. The roll angle of the semi-trailer sprung mass varies in the range of 6.278 to 6.697 degrees for trackwidth of 2.44m, while it ranges from 5.538 degrees to 6.697 degrees when the trackwidth is 2.59 degrees. An increase in trackwidth is seen to increase the static roll stability of the vehicle.

<u>Influence of Articulation Roll Stiffness:</u> The articulation roll stiffness was varied by 20% above and below the nominal value of 6000 kNm/rad and it was found that a lower articulation roll stiffness (80% of baseline) yields a slightly lower value of *SRT*, while the higher stiffness did not affect the *SRT* of the vehicle. The variations in the roll angle were also observed to be small. It is thus concluded that the articulation roll stiffness has only minute effect on the static roll stability of the vehicle.

<u>Influence of Axle Tandem Spread:</u> An increase in the axle tandem spread is seen to reduce the static roll stability of the vehicle. Increasing the axle spread from 1.52m in all the axles to 1.83m reduces the *SRT* by one percent, while the roll angle increases slightly. This decrease is seen to be more prominent, when the semi-trailer cg height is in the lower range. A decrease in the axle spread shows a moderate increase in the static roll stability and decrease in the roll angle of the semi-trailer at the relative roll instability condition. Thus the influence of axle tandem spread on the static roll stability of the vehicle is seen to be modest.

<u>Influence of Tire Vertical Stiffness:</u> The vertical stiffness of the tires of the vehicle was chosen to reflect a highly stiff tire (1050 kN/m) for the baseline vehicle and the stiffness

was gradually reduced to 787 kN/m. The results reveal that a decrease in tire vertical stiffness results in decrease in the roll stability limit of the vehicle, which is not desirable. A decrease of tire stiffness by 263 kN/m results in a decrease in *SRT* by 2.75%, 2.39% and 3.11% for cg heights of 1.8m, 2.0m and 2.2m respectively. Thus it can be inferred that the increase in tire pressure results in an increase in roll stability of the vehicle albeit marginal. The increase in the tire stiffness, however, tends to increase the semi-trailer sprung mass roll angle moderately at the relative rollover condition.

<u>Influence of Tire Cornering Stiffness:</u> The cornering stiffness of the tires is seen to have medium effect on the *SRT* and the roll angle of the vehicle, at low cg heights, as can be seen from Table 4.6(b). At higher cg heights, the effect is seen to be insignificant. Dry radial tires provide higher static roll stability as compared to the tires with half tread or one-third tread. The baseline vehicle is assumed to possess radial tires with tread between half and one-third tread. The roll angle at the relative roll instability condition, for a given cg height, is seen to be consistent, irrespective of the variations in the cornering stiffness. Except for the cg height of 1.8m, where the variation in *SRT* is seen to be around 5%, when the tires are varied from dry radial to 1/3 radial tires, the variations in *SRT* for higher cg heights are found to be moderately insignificant.

<u>Influence of Suspension Lateral Spread:</u> The suspension lateral spread is seen to have significant impact on the static roll stability of the vehicle. An increase of 20% in the suspension lateral spread, increases the *SRT* by 3.3%, 3.59% and 3.37% respectively for the given cg heights of 1.8m, 2.0m and 2.2m. The variation in roll angle at relative roll instability condition subject to an increase in the suspension lateral spread is seen to be more profound. An increase in suspension lateral spread by twenty percent results in

decrease in the roll angle by 11.81%, 12.31% and 14.41%, respectively, for the cg heights considered.

Influence of Damping Ratio: The effect of variations in damping ratio on the static roll stability of the vehicle is seen to be modest. The damping ratios of 0.1, 0.05 and 0.025 were chosen as variations to that of the baseline configuration, with suspension damping ratio of 0.08. The results show that the *SRT* of the vehicle does not vary significantly with variations in the damping ratio. The variation in the semitrailer roll angle was also observed to be small, less than 0.5% in all cases. Thus it can be concluded that the effect of variations in the damping ratio on the static roll stability of the vehicle is insignificant.

Influence of Auxiliary Roll Stiffness: The auxiliary roll stiffness of the semitrailer axles was decreased and increased by 20% first and then the same was done for the tractor rear axles. The results indicate that variation in the auxiliary roll stiffness of the semitrailer axles have lesser effect on the *SRT* than the variations in the tractor axles auxiliary roll stiffness. A decrease in the auxiliary roll stiffness of the semi-trailer axles by 20% decreases the *SRT* of the vehicle approximately half of a percent considering all three cg heights. The roll angle at which the vehicle reaches relative instability condition also does not vary significantly as can be seen from Table 4.6(c). However the variation in the auxiliary roll stiffness of the tractor axles has some impact on the static roll stability and sprung mass roll angle of the semi-trailer as observed from the simulation results.

Influence of Roll Center Height: A variation of 20% in the roll center height of the vehicle above the ground level is seen to show moderate effect on the static roll stability of the vehicle. The effect of such variation in the tractor axles is seen to be greater than the same variations in the trailer axles. The mentioned increase in the tractor rear axles resulted in an increase in *SRT* by 2.5%, 2.0% and 2.3% for cg heights of 1.8m, 2.0m and 2.2m respectively, while the decrease in roll center height resulted in a decrease in *SRT* by 3.0%, 3.3% and 2.3% for the same cg heights. The change in the distance between the cg location and the roll center has more profound effect on the static roll stability as reported in the literature [6] and as can be seen in Table 4.6(c).

Influence of Suspension Vertical Stiffness: The suspension stiffness influences both the static rollover threshold as well as the roll angle to an extent. The variations in suspension stiffness were considered by varying the suspension force by ± 20% in the force-deflection characteristics of the chosen suspension spring for a given set of axles. A 20% reduction in all the axle suspension stiffnesses revealed only slight reduction in *SRT* and increase in the roll angle of the semitrailer sprung mass However, the variations in the tractor rear axle suspension stiffness were seen to affect the static rollover threshold more than the variations in the semi-trailer axles. A decrease in the suspension stiffness by 20% resulted in the reduction of static rollover threshold by 3.92%, 0.37% and 0.52% for cg heights of 1.8m, 2.0m and 2.2m, while the corresponding increase in the roll angle was 6.88%, 5.86% and 6.12%, respectively. The Mack Camel suspension chosen for the tractor rear axles reflected a softer suspension for the axles than the baseline vehicle. It showed a change of less than 2% in the static rollover threshold and a change of 3.0%, 5.2% and 7.8% respectively in the roll angle, considering the c.g heights of 1.8m, 2.0m and 2.2m. It is thus concluded that the tractor rear axle suspension affects the roll stability of the vehicle more than the trailer axle suspension. Furthermore, the effect of variations in the suspension stiffness on the static rollover threshold is more profound at lower cg heights, while the effect on the roll angle is more profound under a higher cg height.

Influence of Operating Load: The influence of the load carried by the vehicle significantly affects the roll stability of the vehicle as it can be seen from Table 4.6(d). The same cg heights were considered for both the loading conditions. A decrease in the payload carried by the vehicle significantly increases the *SRT*, almost by around 10% or more in all cases. The influence of variations in operating load is more profound on the roll angle, which decreases significantly, almost by 30% in all cases. A decrease in the payload leads to a decrease in the axle loads and the yaw, pitch and roll mass moments of inertia of the vehicle thereby significantly decreasing the roll angle at the relative roll instability condition.

From the above analyses, it can be concluded that the trackwidth, cg height, suspension properties and operating load have considerable effect on the static roll stability limit of the articulated vehicles. Table 4.7 summarizes the effect of the parametric variations on the *SRT* and the roll angle responses of a 5-axle tractor semitrailer combination in a qualitative manner.

Table 4.7: Effect of design variations on the static roll stability of the baseline vehicle

Variations Considered	Effect of Variations on SRT	Effect of Variations on Roll Angle
Trackwidth	Significant	Significant at low cg heights
Roll Center Height	Significant	Significant
Suspension Lateral Spread	Significant	Significant
Tire Stiffness	Moderate	Significant
Cornering Stiffness	Significant at low cg heights	Moderate
Suspension	Significant	Significant
Damping	Moderate	Moderate
Axle Spread	Moderate	Moderate
Auxiliary Roll Stiffness	Moderate for variations in tractor axles, Negligible for semi-trailers	Moderate for both tractor and semi-trailers
Articulation Point Roll Stiffness	Negligible	Negligible
Operating Load	Significant	Significant

4.4.3 Dynamic Roll Analyses

As suggested by Lam [81], most path change and lane change maneuvers can be simulated through variations in the steering frequency. Three steering frequencies of 0.25 Hz, 0.33 Hz and 0.50Hz are chosen for the analyses to assess the sensitivity of measures to the rate of steering. The simulations are initiated at low velocities and the velocity is gradually increased until the vehicle approaches the relative rollover condition and finally experiences a rollover. The values of the rollover metrics corresponding to the relative rollover condition are obtained from the results to study the sensitivity of the metrics to variations in design and operating conditions.

Figure 4.8 illustrates the vehicle responses to a sinusoidal steering input (0.25Hz), as an example, at a forward speed below critical speed. The results show that the lateral acceleration response of the semitrailer, roll angle, *RSF*, *LTR*, wheel loads of left and right wheels and the axle roll angles all vary in a sinusoidal manner. The *RSF* and the *LTR* are seen to reach their respective peak values when the wheels on the right side of the vehicle lift off or when the magnitudes of tire loads approach their peak values.

The *RSF* reaching a unity value indicates the relative roll instability condition of the vehicle, while the *LTR* reaching a unity value indicates that all the wheels on a given side of the vehicle have lifted off. Fig. 4.9 shows the response of the baseline vehicle subject to the same maneuver but at a speed above the critical velocity. The time history of *RSF* shows that the *RSF* reaches a unity value before the *LTR* reaches its peak value thereby suggesting that *RSF* can provide an early warning of the onset of a rollover with greater lead-time as compared to *LTR*.

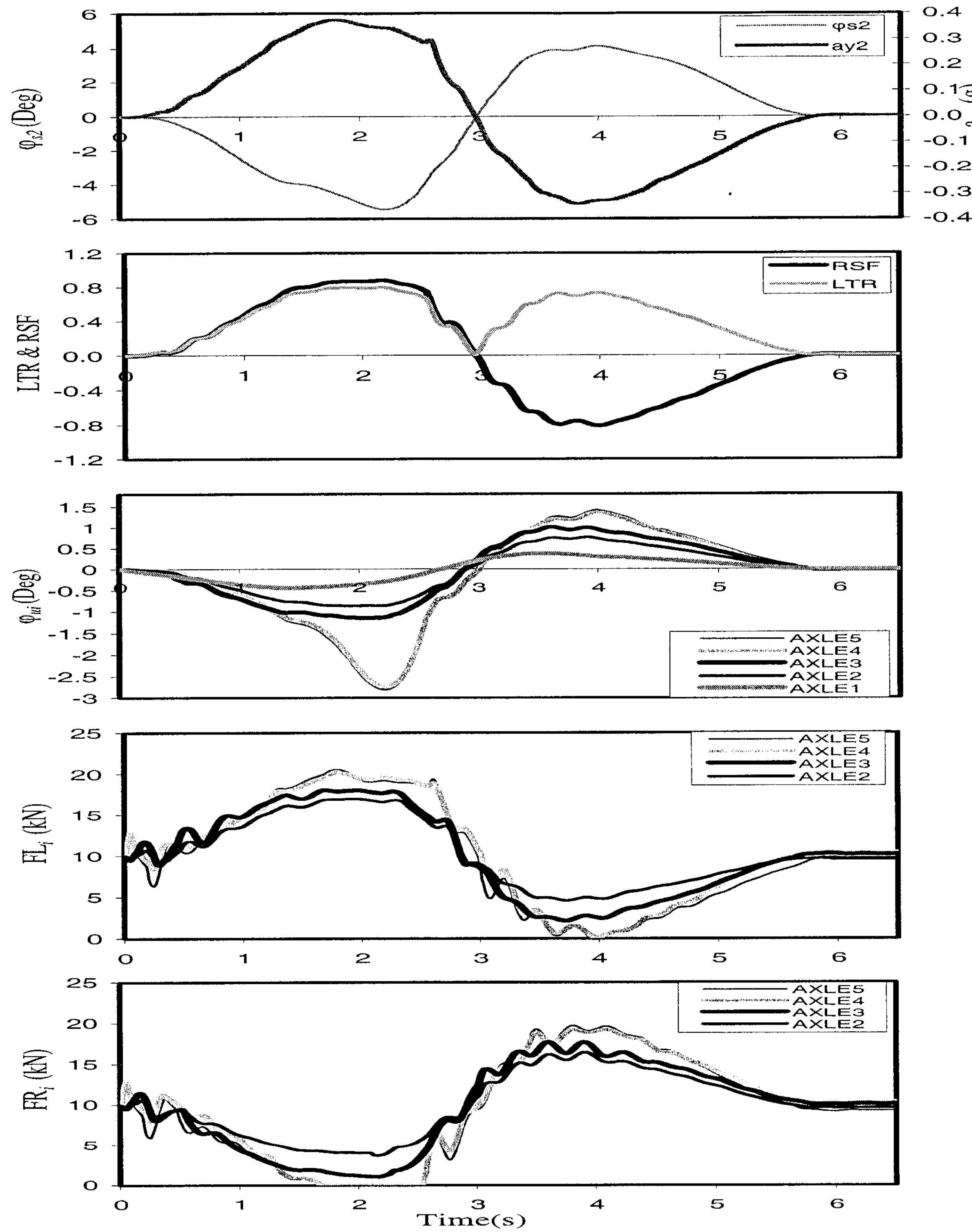


Figure 4.8: Dynamic response of the baseline vehicle to a sinusoidal steering input of 0.25Hz, (cg=2.0m, Operating velocity < Critical rollover velocity).

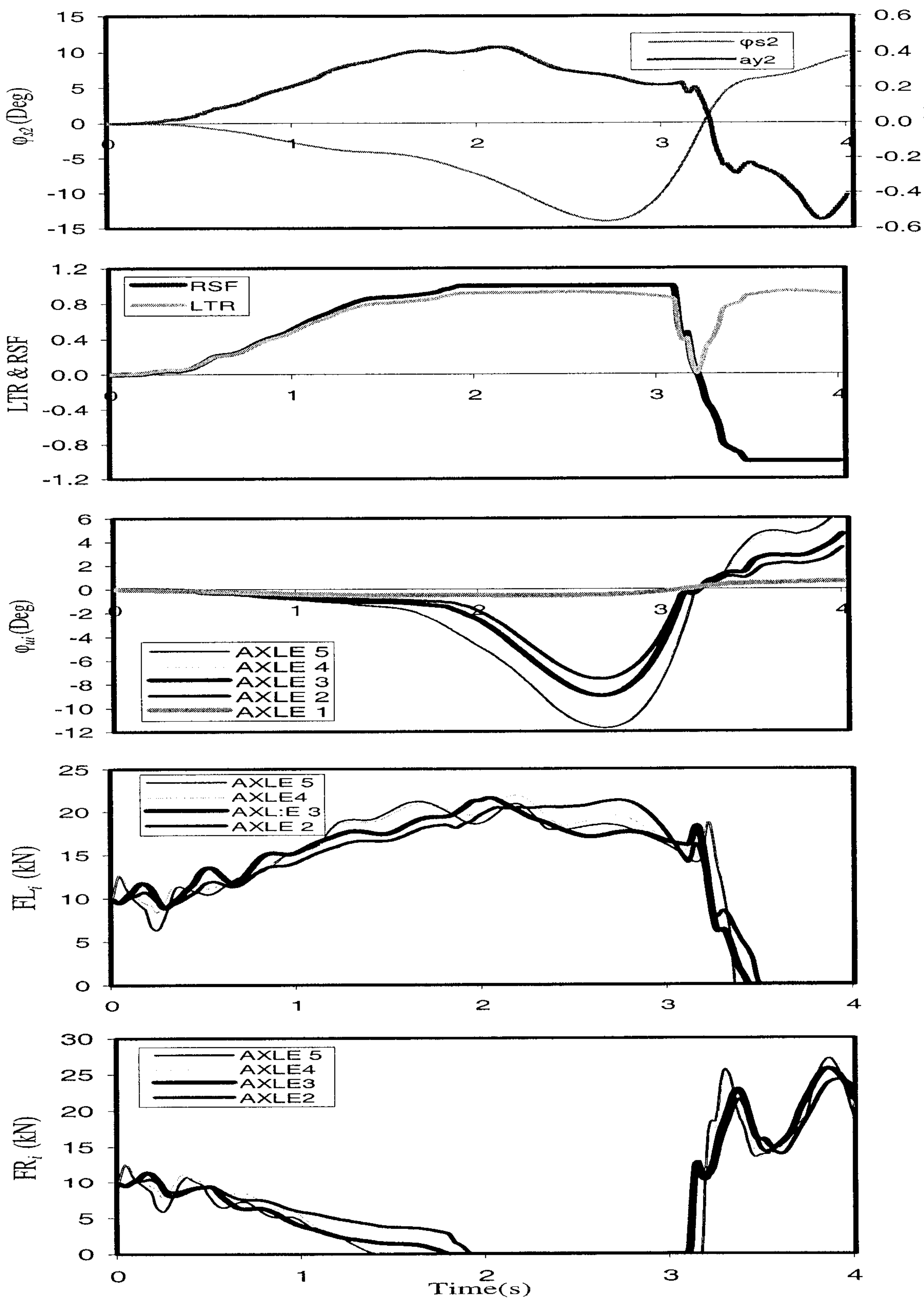


Figure 4.9: Dynamic response of the baseline vehicle to a sinusoidal steering input of 0.25Hz, (cg=2.0m, Operating velocity > Critical rollover velocity).

Table 4.8 summarizes the effects of variations in the selected design and operating conditions on different measures of dynamic rollover, namely, a_{y2}, φ_{s2} and *LTR*., derived from the results attained through the simulation matrix detailed in section 4.3.1. For each variation in design and operating conditions, the cg heights and the steering frequencies are also varied to understand the effect of all these parameters on the roll stability of the vehicle. Considering the baseline vehicle, the lateral acceleration values range from 0.379g (*cg* =2.2m, freq=0.25Hz) to 0.578g (*cg*=1.8m, freq=0.50Hz), while the corresponding roll angle values range from 6.335 to 6.795 degrees. The results suggest higher dynamic roll stability of the vehicle, in terms of the lateral acceleration response, under a higher steering frequency. The roll angle response normally lies within a narrow band under variations in most design variables. The variations in the operating load, however, cause significant variations in the roll angle response as can be seen in Table 4.8(b). At lower frequencies, the change in cg heights result in wider variations in the roll angle than in case of higher steering frequencies. The *LTR* at relative rollover condition is seen to vary in the narrow range of 0.916 to 0.924, considering the baseline vehicle.

The simulation results show that the rollover metrics are highly sensitive to variations in cg heights and steering frequencies as can be seen from Table 4.8. In addition to these parameters, the variations in most design and operating conditions significantly affect the dynamic roll stability limits of the vehicle. The reliability of the indicator would strongly rely upon the sensitivity of the threshold value with variations in the operating parameters and maneuvers. High sensitivity of the threshold to variations in design, operating and environmental parameters could cause either false or lack of warning.

Table 4.8 (a): Effect of variations in geometric and design parameters on a_{y2}, φ_{s2} and LTR when subjected to dynamic maneuvers.

Rollover Metrics →			a_{y2} (g)			φ_{s2} (Deg)			LTR		
Parameter	**Variation**	**Freq(Hz) →**	**0.25**	**0.33**	**0.50**	**0.25**	**0.33**	**0.50**	**0.25**	**0.33**	**0.50**
Baseline Vehicle	-	cg=1.8m	0.481	0.51	0.578	6.335	6.379	6.56	0.924	0.923	0.919
		cg=2.0m	0.403	0.441	0.478	6.635	6.693	6.661	0.916	0.92	0.918
		cg=2.2m	0.379	0.409	0.425	6.795	6.775	6.658	0.916	0.919	0.914
Trackwidth	2.59m	cg=1.8m	0.535	0.541	0.596	6.478	6.416	6.717	0.922	0.922	0.913
		cg=2.0m	0.432	0.465	0.537	6.695	6.763	6.711	0.917	0.918	0.919
		cg=2.2m	0.406	0.442	0.477	6.852	6.929	6.731	0.915	0.917	0.916
Axle Tandem Spread	1.22m	cg=1.8m	0.48	0.507	0.58	6.287	6.319	6.514	0.923	0.922	0.919
		cg=2.0m	0.404	0.441	0.472	6.615	6.731	6.587	0.916	0.919	0.918
		cg=2.2m	0.38	0.409	0.42	6.747	6.778	6.584	0.916	0.918	0.919
	1.83 m	cg=1.8m	0.484	0.514	0.576	6.38	6.436	5.596	0.925	0.924	0.918
		cg=2.0m	0.403	0.445	0.478	6.616	6.79	6.627	0.918	0.92	0.919
		cg=2.2m	0.39	0.41	0.43	6.81	6.813	6.737	0.921	0.919	0.915
Articulation Point Roll Stiffness	4800 kNm/rad	cg=1.8m	0.483	0.51	0.578	6.508	6.441	6.738	0.924	0.923	0.918
		cg=2.0m	0.402	0.441	0.447	6.825	6.91	6.752	0.916	0.919	0.918
		cg=2.2m	0.377	0.409	0.428	6.987	6.991	6.884	0.916	0.918	0.918
	7200 kNm/rad	cg=1.8m	0.482	0.51	0.577	6.272	6.335	6.51	0.924	0.923	0.919
		cg=2.0m	0.407	0.442	0.471	6.52	6.627	6.471	0.918	0.92	0.918
		cg=2.2m	0.38	0.411	0.426	6.641	6.705	6.593	0.916	0.919	0.914

Table 4.8 (b): Effect of variations in operating conditions on a_{y2}, φ_{s2} and LTR when subjected to dynamic maneuvers.

Rollover Metrics →			a_{y2} (g)			φ_{s2} (Deg)			LTR		
Parameter	**Variation**	**Freq(Hz) →**	**0.25**	**0.33**	**0.50**	**0.25**	**0.33**	**0.50**	**0.25**	**0.33**	**0.50**
Baseline Vehicle	-	cg=1.8m	0.481	0.51	0.578	6.335	6.379	6.56	0.924	0.923	0.919
		cg=2.0m	0.403	0.441	0.478	6.635	6.693	6.661	0.916	0.92	0.918
		cg=2.2m	0.379	0.409	0.425	6.795	6.775	6.658	0.916	0.919	0.914
Operating Load	50% Payload	cg=1.8m	0.514	0.524	0.569	4.072	3.958	3.946	0.88	0.885	0.882
		cg=2.0m	0.475	0.483	0.519	4.223	4.14	3.921	0.885	0.884	0.878
		cg=2.2m	0.448	0.459	0.49	4.268	4.176	3.99	0.885	0.886	0.878
Operating Velocity	Critical Speed+20km/h	cg=1.8m	0.526	0.552	0.602	6.368	6.328	6.494	0.923	0.928	0.924
		cg=2.0m	0.456	0.478	0.48	6.658	6.644	6.559	0.931	0.932	0.921
		cg=2.2m	0.42	0.43	0.429	6.728	6.628	6.609	0.931	0.931	0.918

Table 4.8 (c): Effect of variations in tire properties on a_{y2}, φ_{s2} and LTR when subjected to dynamic maneuvers.

Rollover Metrics →			a_{y2} (g)			φ_{s2} (Deg)			LTR		
Parameter	**Variation**	**Freq(Hz)→**	**0.25**	**0.33**	**0.50**	**0.25**	**0.33**	**0.50**	**0.25**	**0.33**	**0.50**
Baseline Vehicle	-	**cg=1.8m**	0.481	0.51	0.578	6.335	6.379	6.56	0.924	0.923	0.919
		cg=2.0m	0.403	0.441	0.478	6.635	6.693	6.661	0.916	0.92	0.918
		cg=2.2m	0.379	0.409	0.425	6.795	6.775	6.658	0.916	0.919	0.914
Tire Vertical Stiffness	**972 kN/m**	**cg=1.8m**	0.478	0.507	0.577	6.524	6.503	6.711	0.924	0.923	0.919
		cg=2.0m	0.404	0.442	0.465	6.79	6.922	6.716	0.917	0.92	0.917
		cg=2.2m	0.375	0.408	0.421	7.012	6.977	6.845	0.915	0.919	0.914
	875 kN/m	**cg=1.8m**	0.478	0.507	0.577	6.524	6.503	6.711	0.924	0.923	0.919
		cg=2.0m	0.404	0.442	0.465	6.79	6.922	6.716	0.917	0.92	0.917
		cg=2.2m	0.375	0.408	0.421	7.012	6.977	6.845	0.915	0.919	0.914
	787 kN/m	**cg=1.8m**	0.471	0.504	0.568	6.844	7.088	7.015	0.923	0.922	0.92
		cg=2.0m	0.407	0.443	0.455	7.275	7.305	7.233	0.919	0.922	0.917
		cg=2.2m	0.374	0.407	0.406	7.488	7.432	7.144	0.916	0.921	0.913
Tire Cornering Stiffness	**Radial-Dry**	**cg=1.8m**	0.489	0.529	0.609	6.481	6.468	6.436	0.923	0.922	0.922
		cg=2.0m	0.404	0.444	0.484	6.627	6.681	6.592	0.917	0.92	0.918
		cg=2.2m	0.375	0.409	0.431	6.778	6.854	6.652	0.916	0.919	0.915
	Radial-Dry- ½ Tread	**cg=1.8m**	0.488	0.531	0.607	6.648	6.412	6.409	0.923	0.923	0.923
		cg=2.0m	0.409	0.443	0.475	6.717	6.715	6.498	0.918	0.92	0.919
		cg=2.2m	0.383	0.412	0.426	6.847	6.799	6.599	0.918	0.92	0.916
	Radial-Dry- 1/3Tread	**cg=1.8m**	0.479	0.486	0.48	6.504	5.404	6.46	0.926	0.914	0.917
		cg=2.0m	0.423	0.438	0.452	6.682	6.786	6.735	0.923	0.922	0.915
		cg=2.2m	0.378	0.409	0.404	6.897	6.815	6.614	0.916	0.921	0.915

Table 4.8 (d): Effect of variations in suspension parameters on a_{y2}, φ_{s2} and LTR when subjected to dynamic maneuvers.

Rollover Metrics →			a_{y2} (g)			φ_{s2} (Deg)			LTR		
Parameter	**Variation**	**Freq(Hz)→**	**0.25**	**0.33**	**0.50**	**0.25**	**0.33**	**0.50**	**0.25**	**0.33**	**0.50**
Baseline Vehicle	-	**cg=1.8m**	0.481	0.51	0.578	6.335	6.379	6.56	0.924	0.923	0.919
		cg=2.0m	0.403	0.441	0.478	6.635	6.693	6.661	0.916	0.92	0.918
		cg=2.2m	0.379	0.409	0.425	6.795	6.775	6.658	0.916	0.919	0.914
Suspension Lateral Spread	Baseline+20%	**cg=1.8m**	0.492	0.514	0.582	5.606	5.588	5.703	0.929	0.927	0.926
		cg=2.0m	0.413	0.452	0.489	5.87	5.96	5.83	0.923	0.927	0.924
		cg=2.2m	0.383	0.42	0.44	5.96	6.052	5.9	0.92	0.924	0.921
Damping Ratio	0.1	**cg=1.8m**	0.481	0.504	0.581	6.359	6.358	6.577	0.924	0.924	0.921
		cg=2.0m	0.4	0.439	0.473	6.599	6.734	6.598	0.917	0.919	0.917
		cg=2.2m	0.378	0.411	0.429	6.765	6.847	6.768	0.917	0.919	0.915
	0.05	**cg=1.8m**	0.476	0.48	0.601	6.376	6.184	5.539	0.924	0.918	0.924
		cg=2.0m	0.401	0.434	0.482	6.62	6.813	6.808	0.915	0.918	0.917
		cg=2.2m	0.378	0.404	0.429	6.856	6.988	6.916	0.915	0.916	0.912
	0.025	**cg=1.8m**	0.464	0.462	0.605	6.337	6.234	6.393	0.921	0.911	0.926
		cg=2.0m	0.416	0.435	0.472	6.717	6.887	6.828	0.918	0.918	0.914
		cg=2.2m	0.409	0.414	0.434	7.163	7.085	7.171	0.922	0.92	0.908
Roll Center Height	Baseline(ST)-20%	**cg=1.8m**	0.479	0.508	0.581	6.367	6.414	6.508	0.924	0.923	0.919
		cg=2.0m	0.402	0.442	0.466	6.624	6.707	6.532	0.917	0.921	0.917
		cg=2.2m	0.374	0.411	0.42	6.767	6.81	6.667	0.915	0.92	0.911
	Baseline(ST)+20%	**cg=1.8m**	0.487	0.519	0.58	6.352	6.437	6.507	0.924	0.923	0.918
		cg=2.0m	0.407	0.444	0.477	6.624	6.768	6.562	0.917	0.92	0.918
		cg=2.2m	0.378	0.411	0.43	6.824	6.765	6.685	0.915	0.919	0.915
	Baseline(TR)-20%	**cg=1.8m**	0.466	0.493	0.573	6.549	6.556	6.726	0.923	0.923	0.924
		cg=2.0m	0.399	0.438	0.462	6.894	6.897	6.814	0.917	0.921	0.918
		cg=2.2m	0.376	0.408	0.413	7.03	7.06	6.85	0.916	0.92	0.913
	Baseline(TR)+20%	**cg=1.8m**	0.502	0.533	0.582	6.146	6.195	6.308	0.922	0.922	0.915
		cg=2.0m	0.414	0.454	0.497	6.484	6.648	6.39	0.916	0.919	0.918
		cg=2.2m	0.382	0.419	0.444	6.575	7.06	6.565	0.914	0.917	0.914

(contd..)

Table 4.8 (d-contd.): Effect of variations in suspension parameters on a_{y2}, φ_{s2} and LTR when subjected to dynamic maneuvers.

Rollover Metrics →			a_{y2} (g)			φ_{s2} (Deg)			LTR		
Parameter	**Variation**	**Freq(Hz)→**	**0.25**	**0.33**	**0.50**	**0.25**	**0.33**	**0.50**	**0.25**	**0.33**	**0.50**
Auxiliary Roll Stiffness	Baseline(ST)+20%	cg=1.8m	0.484	0.513	0.582	6.401	6.370	6.473	0.924	0.923	0.92
		cg=2.0m	0.408	0.443	0.480	6.655	6.664	6.582	0.918	0.92	0.917
		cg=2.2m	0.379	0.414	0.428	6.841	6.849	6.689	0.916	0.919	0.914
	Baseline(ST)-20%	cg=1.8m	0.481	0.507	0.572	6.34	6.34	6.62	0.924	0.923	0.918
		cg=2.0m	0.403	0.403	0.473	6.624	6.624	6.657	0.917	0.917	0.919
		cg=2.2m	0.379	0.405	0.420	6.765	6.807	6.679	0.917	0.919	0.915
	Baseline(TR)+20%	cg=1.8m	0.492	0.523	0.591	6.158	6.169	6.342	0.922	0.921	0.917
		cg=2.0m	0.409	0.459	0.497	6.486	6.539	6.436	0.915	0.917	0.917
		cg=2.2m	0.384	0.420	0.441	6.569	6.691	6.47	0.913	0.916	0.912
	Baseline(TR)-20%	cg=1.8m	0.472	0.5	0.554	6.544	6.554	6.747	0.925	0.924	0.922
		cg=2.0m	0.400	0.435	0.457	6.855	6.855	6.683	0.918	0.921	0.918
		cg=2.2m	0.371	0.387	0.398	7.099	7.105	6.842	0.916	0.921	0.924
Suspension Stiffness	Baseline(ST)-20%	cg=1.8m	0.485	0.497	0.57	6.285	6.299	6.467	0.924	0.924	0.919
		cg=2.0m	0.413	0.441	0.493	6.703	6.675	6.633	0.919	0.921	0.92
		cg=2.2m	0.382	0.41	0.441	6.833	6.817	6.663	0.917	0.919	0.916
	Reyco (ST)	cg=1.8m	0.485	0.506	0.554	6.384	6.426	6.444	0.923	0.923	0.916
		cg=2.0m	0.411	0.434	0.479	6.693	6.742	6.727	0.918	0.919	0.916
		cg=2.2m	0.381	0.403	0.45	6.875	6.83	6.8	0.916	0.917	0.916
	Baseline(ST)+20%	cg=1.8m	0.494	0.522	0.533	6.462	6.487	6.351	0.923	0.926	0.918
		cg=2.0m	0.424	0.432	0.434	6.809	6.587	6.388	0.919	0.921	0.919
		cg=2.2m	0.39	0.392	0.409	6.805	6.584	6.597	0.919	0.917	0.909
	Mack (TR)	cg=1.8m	0.509	0.538	0.594	6.42	6.44	6.578	0.921	0.92	0.915
		cg=2.0m	0.424	0.461	0.502	7.07	6.945	6.699	0.914	0.918	0.915
		cg=2.2m	0.384	0.424	0.442	7.35	7.25	6.771	0.912	0.915	0.912
	Baseline(TR)-20%	cg=1.8m	0.496	0.532	0.59	6.61	6.779	6.853	0.923	0.922	0.916
		cg=2.0m	0.413	0.452	0.495	6.985	7.066	6.878	0.917	0.92	0.92
		cg=2.2m	0.372	0.415	0.438	7.13	7.127	7.039	0.913	0.919	0.925

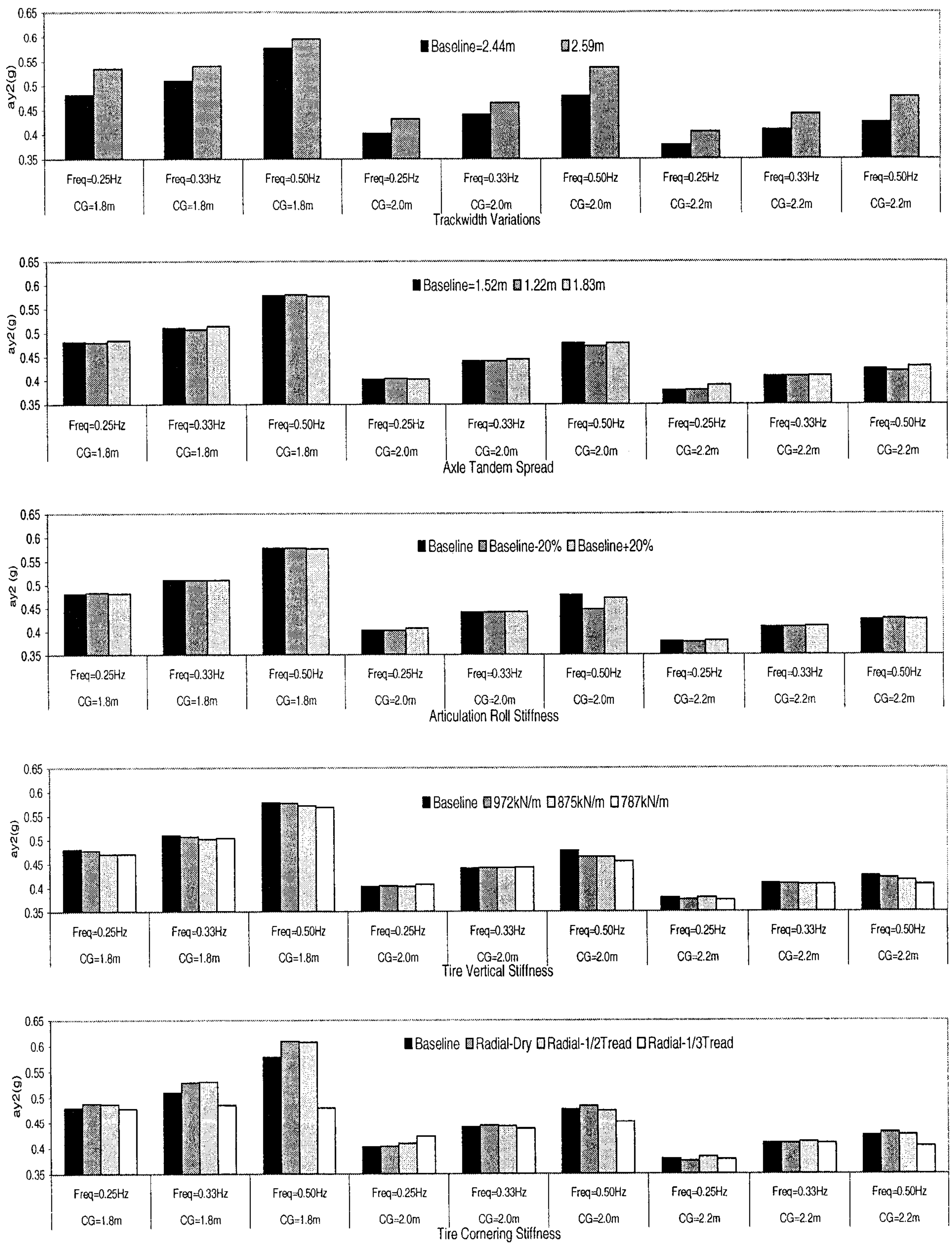


Figure 4.10: The response of a_{y2} to variations in trackwidth, axle tandem spread, articulation roll stiffness, tire vertical stiffness and tire cornering stiffness under dynamic maneuvers.

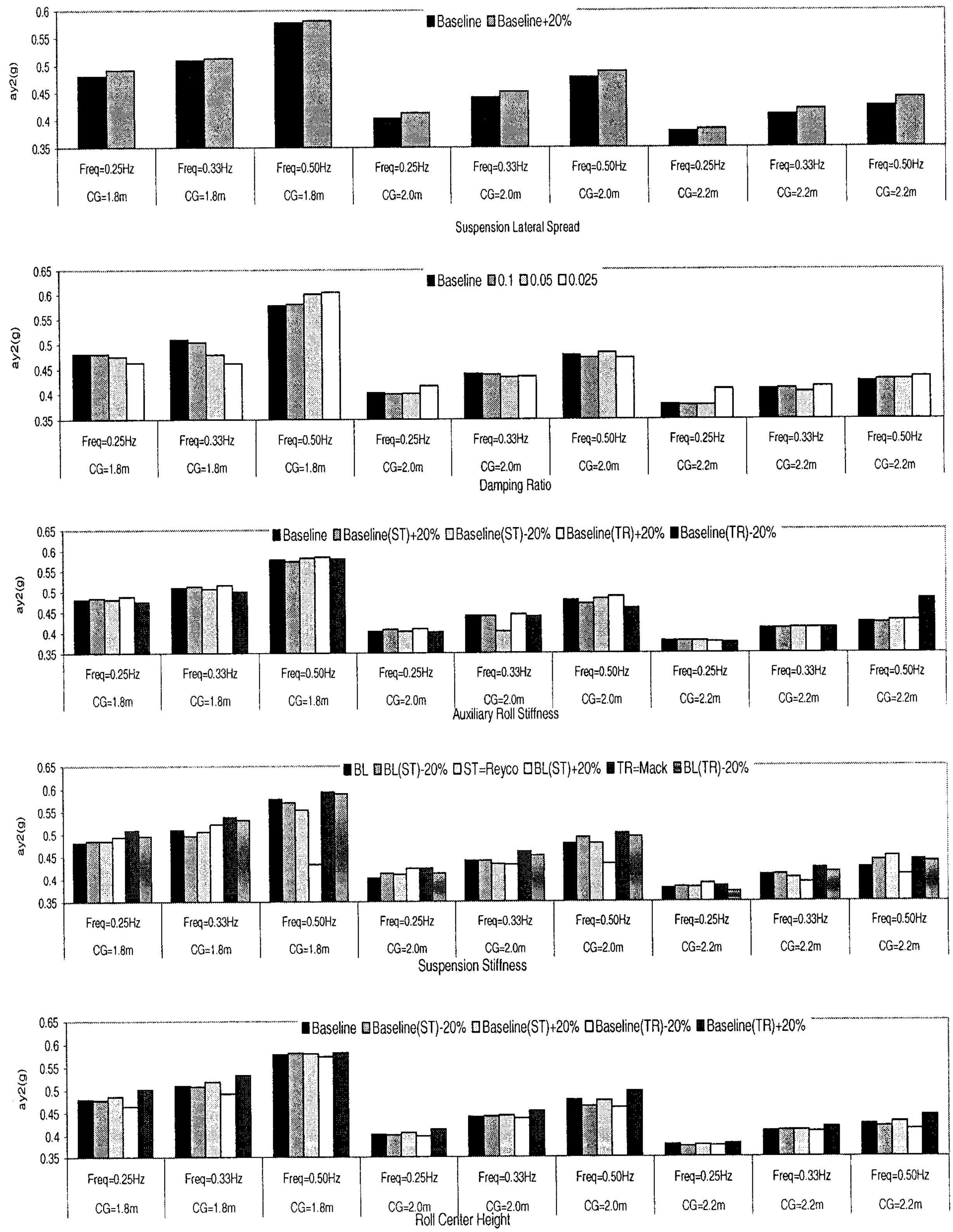


Figure 4.11: The response of a_{y2} to variations in suspension lateral spread, damping ratio, auxiliary roll stiffness, suspension vertical stiffness and roll center height under dynamic maneuvers.

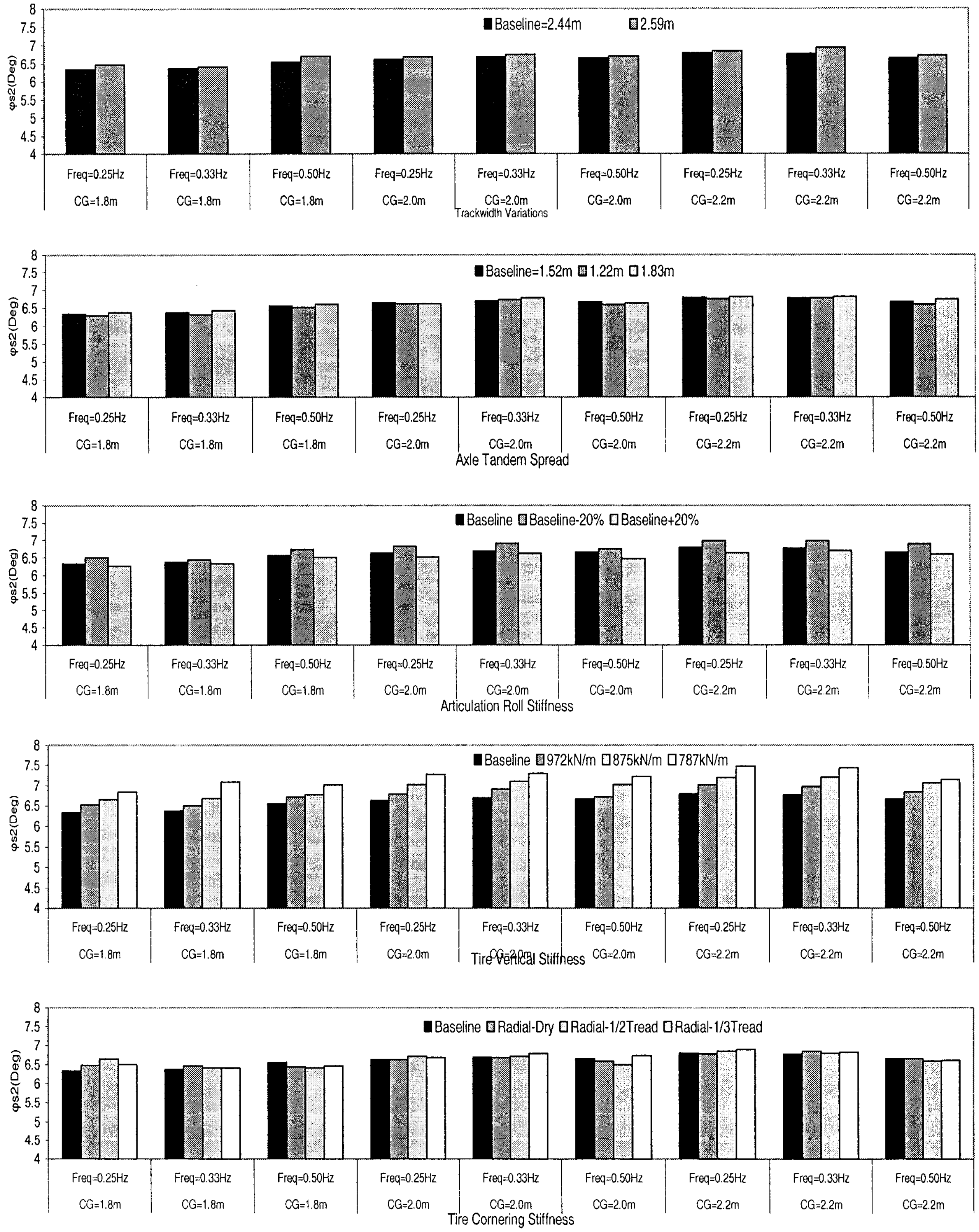


Figure 4.12: The response of φ_{s2} to variations in trackwidth, axle tandem spread, articulation point roll stiffness, tire vertical stiffness and tire cornering stiffness properties under dynamic maneuvers.

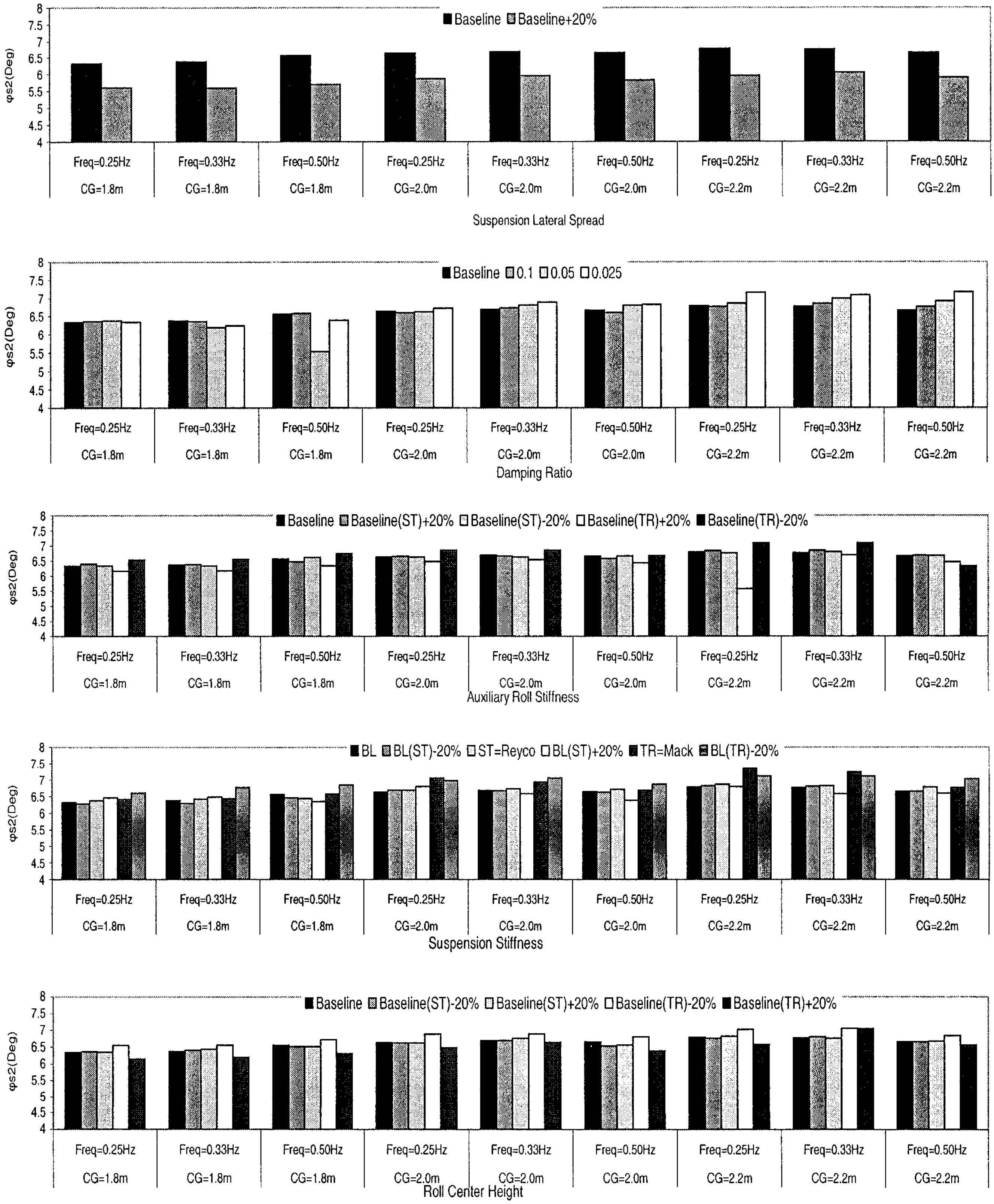


Figure 4.13: The response of a_{y2} and φ_{s2} to variations in suspension lateral spread, damping ratio, auxiliary roll stiffness, suspension vertical stiffness and roll center height under dynamic maneuvers.

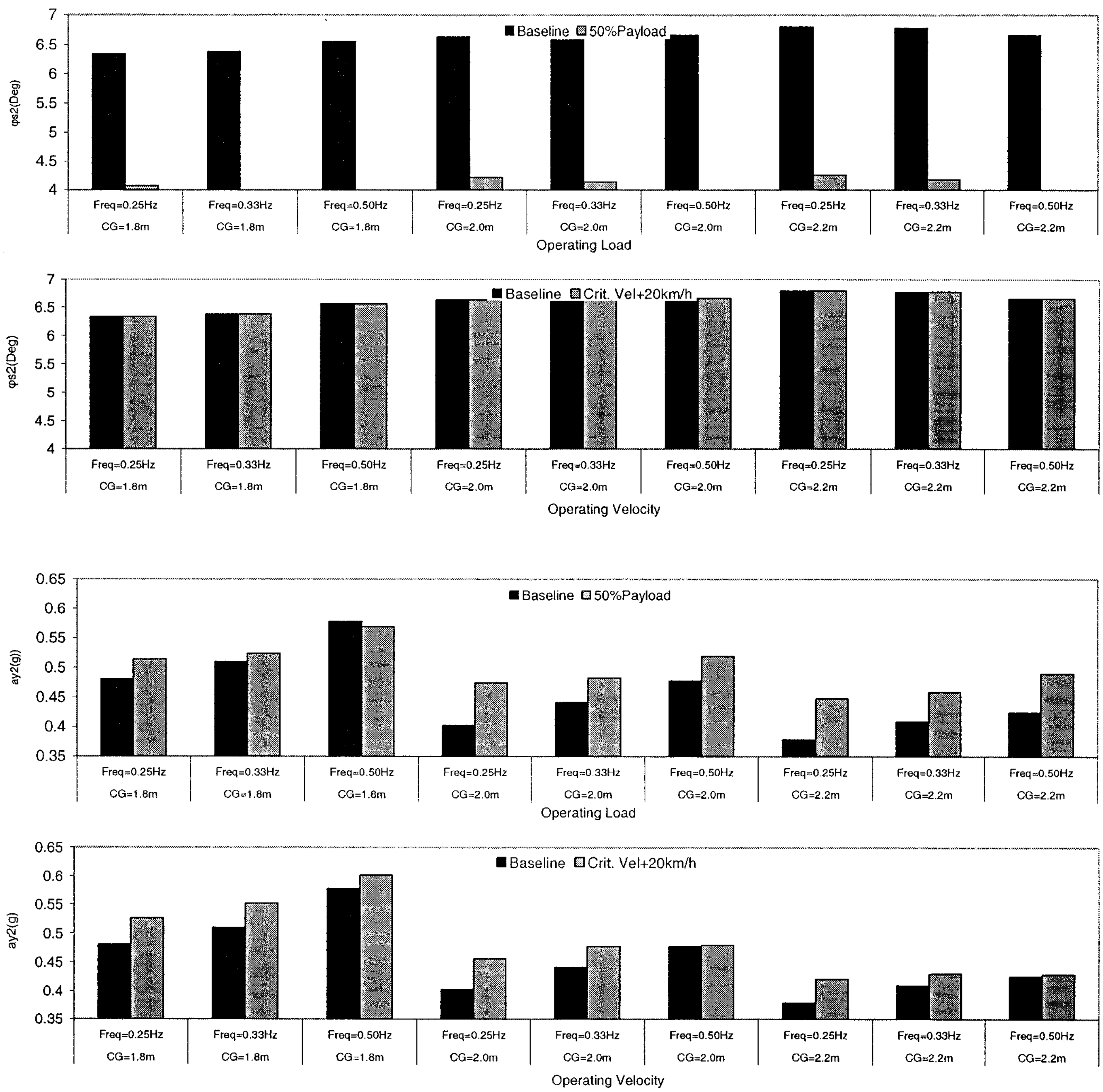


Figure 4.14: Variations in a_{y2} and φ_{s2} to variations in operating load and operating speed.

A rollover indicator can be considered reliable, if its threshold value can be predicted with little error under different operating conditions, and it does not vary significantly in daily operations of the vehicle. The values assumed by different metrics are obtained at the stage when *RSF* reaches a unity value or when the forward velocity approaches the critical rollover velocity. The *LTR* approaches a unity value considerably later than the *RSF*, which would relate more or less with an absolute rollover. The use of *LTR* measure is thus unlikely to provide an early warning of the roll instability.

The relative comparisons of the effects of different variations, as outlined in the simulation matrix, on the rollover metrics are presented in Figures 4.10 to 4.14. These figures illustrate the level of sensitivity of the rollover metrics to variations in different design and operating conditions under sinusoidal steering of varying frequencies, and *cg* heights.

4.4.4 Sensitivity Analysis Based on the Simulation Results on Dynamic Rollover Thresholds

Influence of Vehicle Trackwidth: The variations in the trackwidth from 2.44m (baseline) to 2.59m shows little increase in the roll angle of the semi-trailer roll angle. However the increase in trackwidth by 0.15m increases the lateral acceleration corresponding to the relative rollover condition (*DRT*) by approximately 10% or more. Thus it can be concluded that the dynamic roll stability of an articulated vehicle is sensitive to variations in the trackwidth. The variation in trackwidth is seen to cause variations in the roll angle (φ_{s2}) not more than 5% and hence it can be inferred that the variation in the trackwidth has little to moderate influence on the roll angle at relative roll instability condition. At higher steering frequencies, the dynamic rollover threshold is generally more than the static rollover threshold, and approaches a smaller value with decrease in the excitation frequency approaching a value lower or equal to the *SRT*.

Influence of Articulation Roll Stiffness: The variations in the articulation roll stiffness for the fifth wheel cause only slight variations in the dynamic rollover threshold in terms of lateral acceleration as well as the roll angle of the semi-trailer sprung mass corresponding to relative rollover condition. The variation in the lateral acceleration threshold is mostly less than 2%, while the variation in roll angle is below 3%. It can thus be concluded that the articulation roll stiffness has little influence on the dynamic roll stability of the vehicle, as observed for the static roll stability limit.

Influence of Axle Tandem Spread: An increase in the axle spread is seen to marginally increase the dynamic roll stability of the vehicle, when measured in terms of a_{y2} and φ_{s2}. A reduction in axle spread from 1.52 m to 1.22 m yields only minimal changes in the a_{y2} and φ_{s2}. However when the axle spread is increased from 1.52 m to 1.82 m, the dynamic rollover threshold is seen to increase marginally.

Influence of Tire Vertical Stiffness: The tire stiffness based on the tire pressure resulted in choice of four different stiffness values for the simulations: 1050 kN/m, 962.5 kN/m, 875 kN/m and 787.5 kN/m [80]. A decrease in the stiffness yields lower lateral acceleration threshold and higher roll angle at relative roll instability condition. The variation in the a_{y2} is observed to be less than 5%, when the tire stiffness is reduced from 1050 kN/m (110psi) to 787.5 kN/m (80psi), while the variation in the roll angle is seen to be approximately 10% for the same variation. It can thus be concluded that the variations in the tire stiffness in the considered range do not adversely affect the dynamic rollover threshold, but affects the roll angle moderately. An increase in the stiffness although can be considered as a mean to increase the roll stability by enhancing the effective roll stiffness.

Influence of Tire Cornering Stiffness: The effect of variations in the cornering stiffness of the tires is seen to affect the roll stability of the vehicle more significantly. The effect is more evident under higher steering frequencies while the effect on the lateral acceleration values is more pronounced than the roll angle values. A decrease in the tread condition of the tire decreases the dynamic rollover threshold of the vehicle. At higher frequencies the decrease is almost up to 17% in the lateral acceleration value for the case involving 1.8m cg height and 0.50Hz steer frequency. It is thus concluded that at higher steering frequencies the roll stability of the vehicle is seen to deteriorate with the reduced tire tread, while at lower frequencies the effect of variations in the cornering stiffness is relatively less.

Influence of Suspension Lateral Spread: The effect of variations in the suspension lateral spread is seen to have significant effect on the roll angle response corresponding to the relative roll instability condition, while the effect on the lateral acceleration of the semi-trailer sprung mass is seen to be moderate. A decrease in the suspension lateral spread by 20% in all the axles is seen to decrease the lateral acceleration threshold value by less than 3%, while it is seen to increase the threshold value of the roll angle by 16% (0.25Hz steering frequency and 2.2m cg height). A reduction in the suspension spread is, however, not a feasible option in real practice. This particular variation is thus ignored for the parametric sensitivity analysis. An increase in the suspension spread by 20% in the rear axles of the tractor and the semi-trailer is seen to increase the lateral acceleration threshold in the range of 0.5% to 3.5%, while it results in decrease in the roll angle by up to 12.5%.

Influence of Damping Ratio: The variations in the damping ratio for all the axles suspensions of the vehicle cause moderate to significant changes in the lateral acceleration threshold as well as the roll angle. The variations in lateral acceleration threshold value are less than 10% for variations in damping ratio considered at lower frequencies and low cg heights, while at higher frequencies and high cg heights the variations are below 5%. Lower suspension damping, however, yields higher dynamic rollover threshold values at higher steering frequencies but lower values at lower steering frequencies. The effect on the variations in the roll angle is also seen to be less than 10%. Hence, the suspension damping is seen to have moderate to significant influence on the roll stability of the vehicle under dynamic maneuvers while the effect is dependent upon the rate of steering.

Influence of Auxiliary Roll Stiffness: The auxiliary roll stiffness of different composite axle sets is varied by varying the auxiliary roll stiffness of the semi-trailer axles and the tractor rear axles. The results obtained from simulations indicate that the variations in the auxiliary roll stiffness of the tractor rear axles influence the dynamic roll stability of the vehicle more than the variations in the semi-trailer axles. The increase or decrease in auxiliary roll stiffness in the semi-trailer axles causes marginal increase or decrease in the lateral acceleration or the roll angle of the vehicle at relative roll instability condition. The variations in the auxiliary roll stiffness of the tractor rear axles cause a variation of around 4-7% in a_{y2} for the cg height of 2.2m and steering frequency of 0.50Hz, while the overall variations are below 5% in all other cases. The influence of such variations in the semitrailer roll angle response is also found to be moderate. The roll angle decreases marginally with increase in auxiliary roll stiffness and increases with a decrease in the auxiliary roll stiffness.

Influence of Roll Center Height: The variations in the heights of the roll centers above the ground for different axles show moderate effect on the dynamic rollover measures. The variations in the roll center height of the tractor axles, however, show greater influence on roll stability limit of the vehicle than variations in the semitrailer axles. The variations in the lateral acceleration as well as the roll angle corresponding to the relative rollover condition vary within 5% for all the variations considered. Thus we can conclude that the roll center height above the ground has a relatively moderate of influence on the dynamic roll stability limit of the vehicle. The distance between the roll center and the center of gravity of the sprung mass, however, affects the roll stability of the vehicle significantly as can be seen from, Table 4.8. An increase in the cg height for a given roll center height above ground tends to significantly decrease the roll stability of the vehicle. A higher location of the roll center above the ground may thus be considered desirable parametric in view of the dynamic roll response of the sprung mass. The shifting of the roll center to the tire-ground contact point when a particular axle lifts off brings in errors in the simulation results as the constant velocity yaw/roll model considers a fixed roll center height while computing the response parameters.

Influence of Suspension Vertical Stiffness: Stiffer suspensions are known to increase the roll stability of tractor semi-trailers by enhancing the effective roll stiffness, while the ride quality tends to deteriorate. The results show that the considered variations in the selected suspensions for a given axle moderately affect the dynamic roll stability of the vehicle and the roll angle at relative rollover condition. The variations in the tractor rear axles suspension rate are seen to influence the roll stability more than the variations in the semi-trailer axles suspension rate. The range of variations in the suspension rates of the tractor rear or the semi-trailer axles, considered in the study, yields changes in the stability measures within 5%. However, variations in the vertical stiffness of all the axle suspensions reveal significant influences on the roll stability of the vehicle. The results thus suggest that increasing the auxiliary roll stiffness and choice of suspension, could contribute to enhanced roll stability limits.

Influence of Operating Speed: The simulation results are derived corresponding to the minimum velocity at which the vehicle approaches the relative rollover condition ($|RSF = 1|$). This velocity is referred to as the critical roll velocity of the vehicle. An increase in the forward speed yields insignificant effect on the roll angle, while the lateral acceleration experienced by the semitrailer sprung mass tends to increase. The results suggest that the rollover metric should be based on the semitrailer lateral acceleration at critical rollover velocity, rather than the roll angle.

Influence of Operating Load: A reduction in the payload by 50% yields higher lateral acceleration but lower roll angle response corresponding to the at relative rollover condition. This decrease in the payload by 50% is seen to decrease the roll angle by up to 40% and increase the dynamic rollover threshold by up to 18% over the entire range of the cg heights and the steering frequencies considered in this study. However, for a given cg height, the lateral acceleration response is seen to be moderately influenced by the variations in the operating load. A decrease in the payload leads to a decrease in the axle loads and the yaw, pitch and roll mass moments of inertia of the vehicle, thereby significantly decreasing the roll angle at the relative roll instability condition.

The above results on the dynamic roll analysis suggest that the semitrailer roll angle is less sensitive to variations in the cg height, maneuver variations and variations in many design parameters, but it is extremely sensitive to the loading condition. The *LTR* is most insensitive but its measurability remains the major challenge, while the lateral acceleration varies considerably, although within a range, when the design and operating conditions are varied. This range of variations could provide significant information on the minimum stability limits for developing a warning and control strategy. It is also a salient observation that the parametric variations on the tractor rear axles show greater effect on roll stability of the vehicle as compared to that on the semitrailer axles. Higher cg heights and lower steering frequencies result in low threshold limits in terms of lateral acceleration response.

The operating load is one of the most crucial parameter, which significantly affects the sensitivity of the considered rollover metrics. The suspension parameters are also seen to influence the roll stability of articulated vehicles. The reliability of the considered metrics compare unfavorably, which emphasizes the need for alternate measures of roll instability that are independent of variations in design as well as operating conditions.

Table 4.9 summarizes the sensitivity of a_{y2}, φ_{s2} and *LTR* measures to variations in different design and operating parameters of a tractor-semitrailer combination with cg height in the 1.8m-2.2m range and subject to sinusoidal steering maneuvers of 0.25Hz, 0.33Hz and 0.50Hz frequencies.

Table 4.9: Summary of sensitivity of a_{y2}, φ_{s2} and *LTR* to variations in design and operating parameters under dynamic directional maneuvers.

VARIATIONS	EFFECT ON a_{y2}	EFFECT ON φ_{s2}	EFFECT ON LTR
Trackwidth	Significant	Moderate	Insignificant
Roll Center Height	Moderate	Moderate	Insignificant
Suspension Lateral Spread	Moderate	Significant	Insignificant
Tire stiffness	Moderate	Significant	Insignificant
Cornering Stiffness	Significant	Moderate	Insignificant
Suspension Vertical Stiffness	Moderate	Moderate	Insignificant
Damping	Moderate	Moderate	Insignificant
Axle Spread	Insignificant	Insignificant	Insignificant
Auxiliary Roll Stiffness	Moderate	Moderate	Insignificant
Art. Point Roll Stiffness	Insignificant	Insignificant	Insignificant
Operating Speed = Crit.Vel+20 km/h	Significant	Insignificant	Insignificant
Loading Condition= 50% Payload	Significant	Significant	Moderate

4.5 SUMMARY

The simulation parameters for a baseline 5-axle tractor-semitrailer vehicle are formulated on the basis of known geometric parameters, suspension and tire properties, steering and handling components, and mass distribution properties. A comprehensive simulation matrix comprising wide variations in the component properties and weights and dimensions is formulated to study the sensitivity of the vehicles response parameters to variations in design and operating conditions. The simulations are conducted for each element of the simulation matrix and the response parameters obtained from the simulations are analyzed to derive the rollover metrics corresponding to the relative rollover condition characterized by the unity value of *RSF*. The simulations are performed to assess static and dynamic rollover measures, under steady and sinusoidal steering inputs. The most important response parameters like the lateral acceleration of the semi-trailer sprung mass and the roll angle are obtained at the relative roll instability condition. The results show that the trackwidth, the center of gravity height, suspension properties and the operating load have significant effect on the roll stability limits of the vehicle under both static and dynamic conditions. The damping, tire stiffness, lateral suspension spread, roll center height above the ground and the auxiliary roll stiffness have moderate influence, while the articulation point roll stiffness and the axle spread have little influence on the roll stability. The effect of parametric design variations incorporated to the tractor's rear axles have more effect on the roll stability of the vehicle as compared to the variations incorporated into the semitrailer axles. The objective of these simulations was to obtain the measures of rollover that are least sensitive to design and operating condition variations. Although the lateral acceleration and roll angles vary within certain limits when the vehicle is subjected to various design and operating condition variations, the most significant variation is seen when the loading condition is changed. Thus in an endeavor to obtain a rollover metric that is independent of design and operating condition variations, the reported measures cannot be employed singularly to predict the onset of a rollover in a reliable manner. Alternate measures are thus desired to predict an impending rollover with greater reliability. These metrics should be robust and least sensitive to all the variations considered, while being easily measurable and capable of providing reasonable lead-time.

CHAPTER 5

IDENTIFICATION OF A NEW RELIABLE ROLLOVER METRIC

5.1 INTRODUCTION

The parametric sensitivity analyses of the existing measures of rollover performed in the previous chapter show that the vast majority of these measures tend to deviate considerably with variations in design and operating conditions. The lateral acceleration, which is accurately and easily measurable, is seen to vary with the mentioned variations and it is not possible to have a generalized value of lateral acceleration that can be considered as the rollover threshold for articulated vehicles, irrespective of the design and operating condition variations. Piché [26] suggested that the roll angles of the axles can provide an indication of impending rollover instability with greater reliability. The measures based upon unsprung mass roll angle are observed to be less sensitive to variations in maneuvers and sprung mass cg height as well as most of the design variations, while it shows extreme sensitivity to variations in the loading conditions.

The results attained on the relative sensitivity of various measures suggest that the reported measures could not be reliably applied for predicting the onset of rollover of articulated vehicles, when potential variations in design and operating conditions are considered. A need to identify alternate measures that predict the onset of rollover with greater reliability over a wide range of variations in design and operating conditions thus exists. These alternate metrics then need to be thoroughly analyzed for their reliability and lead-time performance in order to identify the most effective metric(s) that can be incorporated in an early warning and control strategy.

In this chapter, an alternate rollover metric based upon the combination of roll deflections of unsprung masses of the tractor and the semitrailer is synthesized. The proposed rollover metric based upon the normalized roll dynamic responses of sprung and unsprung masses of both the units is analyzed to predict its reliability through sensitivity analyses to variations in design and operating conditions, in accordance with the simulation matrix discussed in the previous chapter. The simulation results are used to quantify the range of values assumed by the metric, when subjected to the selected ranges of given variations. The lead-time performance of the proposed measure is further investigated over the range of variations. The merits of the proposed rollover metrics are discussed in relation to the reported measures in terms of their sensitivity to design and operating conditions variations, measurability and lead-time using the performance measures described in the previous chapter. Based on the relative assessment, the most reliable and feasible measures are identified for implementation in an early warning algorithm.

5.2 IDENTIFICATION OF POTENTIAL ROLLOVER METRICS

The response vectors obtained from a vehicle in motion are mostly linear or angular in nature. The lateral acceleration and the roll angles of the sprung and the unsprung masses, dynamic wheel loads and the load transfer form the response vectors that provide the primary motion cues, when a vehicle experiences potentially critical roll motions. The simulation results have shown that both the lateral acceleration and the semitrailer sprung mass roll angle provide an actual measure of the rollover propensity. Owing to their sensitivity to changes in design configurations and operating conditions, these measures are considered to offer poor reliability in predicting a potential rollover. It thus becomes imperative to identify alternate measures of roll instability, which are in phase with these measures but are superior in terms of reliability and lead-time issues.

As reported by Piché [26] and as evident from the results obtained from the simulations, the axle roll angles provide information regarding impending roll instability with fair amount of reliability. The results revealed only small variations in roll angle responses of the tractor front and the semi-trailer rear axles over the entire range of variations considered. The roll angle responses of the tractor rear axles, however, were observed to be sensitive to variations in the load carried by the vehicle. The ratios of the roll angle of the rearmost axle of the semi-trailer to that the tractor front axle, and to that of the tractor rear axle, represent the rearward roll amplification of the articulated vehicle and could possibly negate the influence of variations in the design and operating conditions, particularly the loading conditions. Considering that a measure based upon the semitrailer roll angle could offer superior lead-time performance, a measure incorporating the roll response of both the semitrailer sprung mass and the tractor front and rear axles could be more desirable. These alternate rollover metrics are synthesized as possible measures of the relative rollover condition and the *RSF*. The synthesis is realized upon systematic formulation of the response measures that are judged to relate to the roll dynamics of the vehicle. The measures mentioned above are discussed in detail as follows:

Rearward Amplification in Roll (RAR_φ):

The ratio of the sprung mass roll angle response of the semi-trailer to that of the tractor, the Rearward Amplification in Roll (RAR_φ) response of the sprung masses, is defined as:

$$RAR_\varphi = \frac{\phi_{s2}}{\phi_{s1}} \tag{5.1}$$

where ϕ_{s2} is the semitrailer sprung mass roll angle and ϕ_{s1} is the tractor sprung mass roll angle. In both the static and dynamic maneuvers, this ratio is seen to be close to unity before the vehicle attains the relative rollover condition, as a result the two sprung mass roll angles exhibit good phase response. The suitability of this ratio as an indicator of the relative roll instability necessitates that the ratio should either attain a distinctive value or vary within a narrow range, when variations in design and operating conditions are considered. The preliminary simulation results suggest that the reliability of this measure is an area of concern, while the lead-time performance is unappealing, since the measure does not incorporate the roll angle responses of the unsprung masses, which could yield superior lead-time performance.

Unsprung Mass Roll Amplification Ratio ($URAR_1$):

Considering that the semitrailer axle roll response is relatively insensitive to many design and operating conditions, an alternate measure defining rearward unsprung mass roll tendency is defined to reliably predict the onset of a relative roll condition. This unsprung mass roll amplification is defined as the ratio of the semitrailer rearmost axle roll angle response to that of the tractor front axle and termed as unsprung mass roll amplification ratio ($URAR_1$), such that:

$$URAR_1 = \Phi_{u5/1} = \frac{\phi_{u5}}{\phi_{u1}} \tag{5.2}$$

This ratio incorporates the axle roll response of the fifth axle (ϕ_{u5}), which experiences the largest roll response, and the first axle (ϕ_{u1}), which experiences the least roll response. In both the static and dynamic maneuvers, the roll response of the rearmost axle relates to a possible lift off of the rear axles and thus an impending roll instability. An increase in the rearmost axle response could thus be considered as the first indication of a roll instability. Such a measure, however could yield frequent false warnings, specifically under dynamic maneuvers and when the vehicle encounters bumps or experiences a lateral force or impulse not large enough to cause a rollover. The reliability of this measure could be enhanced through consideration of the relative change in the roll angle of the front axle, which does not lift off until the loss of contact of the tire with the road occurs over a sustained period of time. The reliability of this measure under varying design and operating conditions, however, needs to be assessed to judge its suitability as a predictor of a roll instability.

Unsprung Mass Roll Amplification Ratio ($URAR_2$):

Most articulated vehicles exhibit relatively small roll angle response of the tractor front axle, which may cause relatively large variations in the unsprung mass roll amplification ratio ($URAR_1$). Alternatively, the amplification of the rearmost axle roll response with respect to that of the tractor rear axle may be considered as a better measure. The unsprung mass roll amplification ratio with respect to the tractor rear axle ($URAR_2$), is thus defined as:

$$URAR_2 = \mathbf{\Phi}_{u5/2} = \frac{\phi_{u5}}{\phi_{u2}} \tag{5.3}$$

The suitability of this ratio as a reliable measure for implementation in an early warning system requires further investigations on its sensitivity to variations in the design and operating conditions. It is anticipated that the tractor rear axle roll response offers a greater potential for early warning as compared to the tractor front axle roll response, which tends to be relatively low in magnitude. Furthermore, the semitrailer axles generally exhibit the largest roll angle and provide the earliest indication of roll instability. The measure is thus expected to yield reasonably good lead-time. A preliminary investigation involving limited simulations revealed that the unsprung mass roll amplification ratio containing the tractor rear axle response yields relatively less variations in its magnitude.

Normalized Roll-response of Semi-Trailer Sprung Mass (*NRSSM*):

The roll angle responses of the semi-trailer sprung mass(ϕ_{s2}) and the unsprung masses (ϕ_{ui}) maybe considered for predicting onset of a relative rollover instability under static and dynamic maneuvers. These roll angles provide higher lead-time and characterize the roll instability of the vehicle subjected to directional maneuvers at highway speeds [6,26]. An alternate measure, based upon roll responses of the semitrailer sprung mass, and the tractor front and rear axles is thus formulated to identify a more reliable rollover metric that may also yield reasonably good lead time. The measure, expressed as *Normalized Roll-response of Semitrailer Sprung Mass* (*NRSSM*), incorporates the rearward amplification tendency of the combination and the lead unit in roll, and roll deflection transmission characteristics of the lead unit, such that:

$$\lambda_{s2} = \Phi_{s1} . RAR_{\Phi} \tag{5.4}$$

where λ_{s2} is the normalized roll response of the semitrailer sprung mass, and RAR_{Φ} is the instantaneous rearward roll amplification of the sprung masses of the combination, as described in Equation (5.1). The first term in Equation (5.4), Φ_{s1}, can be related to the roll deflection transmission of the tractor and the rearward roll progression of the tractor, which would depend upon the effective roll stiffness of the tractor axle suspension and chassis, and is expressed as:

$$\Phi_{s1} = {\phi_{s1}}/{\Phi_{u1}} \text{ ; where } \Phi_{u1} = \frac{\phi_{u2}}{\phi_{u1}} \text{ and } \phi_{u2} \neq 0 \tag{5.5}$$

where Φ_{s1} is used to express a measure of the roll deflection transmission of the lead unit, and is directly related to the suspension properties of the unit, while Φ_{u1} represents the rearward roll amplification property of the lead unit alone.

The defined measure *NRSSM* may thus be simplified to:

$$NRSSM = \lambda_{s2} = \frac{\phi_{s2}}{\phi_{u2}/\phi_{u1}} \text{ for } \phi_{u2} \neq 0 \tag{5.6}$$

This measure, however, can assume a very high value, when ϕ_{u2} approaches zero, as it can be seen from equation 5.6. The *NRSSM* value could be taken as 0, when the magnitude of ϕ_{u2} is in the vicinity of a limiting value around zero (e.g. $|\phi_{u2}| \leq 0.20$ degrees), as this would help to prevent sudden surge in the value of *NRSSM* as ϕ_{u2} approaches a very small value thereby preventing the generation of false warnings. Figure 5.1 illustrates the variations in *NRSSM* for the nominal parameter articulated vehicle combination under steady-turning and sinusoidal directional maneuver at 0.25Hz. The results obtained from the simulations indicate that the magnitude of *NRSSM* could vary within ±3 degrees under typical directional maneuvers. The variations in *NRSSM* are further compared with those of the lateral acceleration (a_{y2}), semitrailer sprung mass roll angle (φ_{s2}) and *RSF* in Figure 5.2. The figures illustrate the responses to a 0.25Hz sinusoidal maneuver corresponding to forward speeds below and above critical rollover velocity. The critical rollover velocity of the baseline vehicle is defined as the forward speed, which causes the *RSF* to approach a unity value. The results show that the variations in the *NRSSM* follows the trends similar to those of the a_{y2} and φ_{s2}, which are considered as reasonably good measures of a relative roll instability. Moreover, *NRSSM* response leads the φ_{s2} as well as a_{y2}. The *NRSSM* may thus be considered as a potential measure of the relative rollover. Its reliability under a wide range of variations in design and operating conditions is thus investigated through parametric sensitivity analysis.

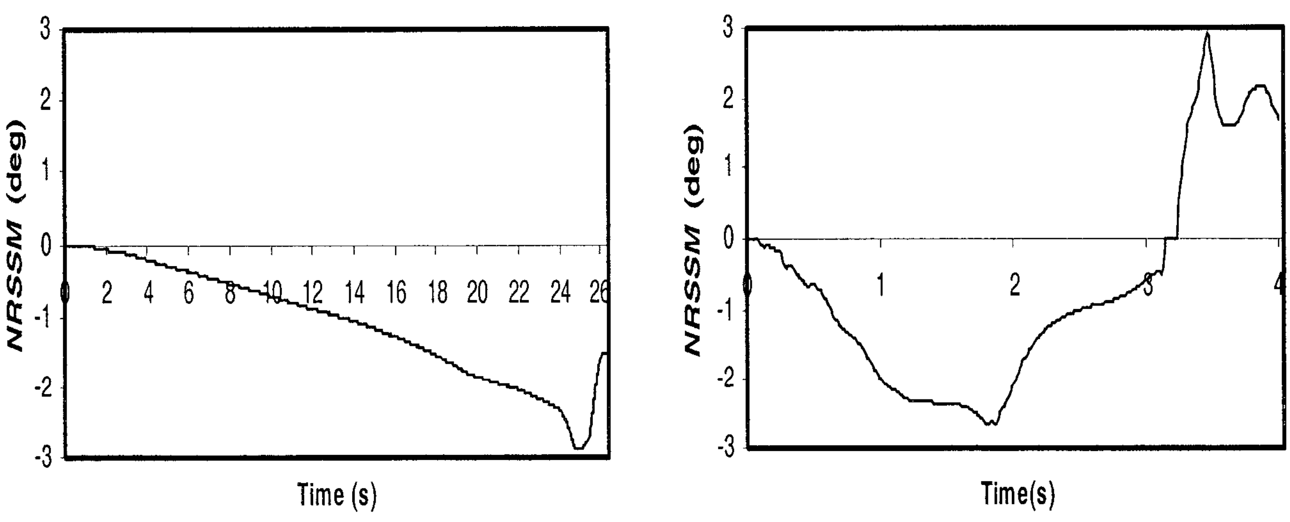
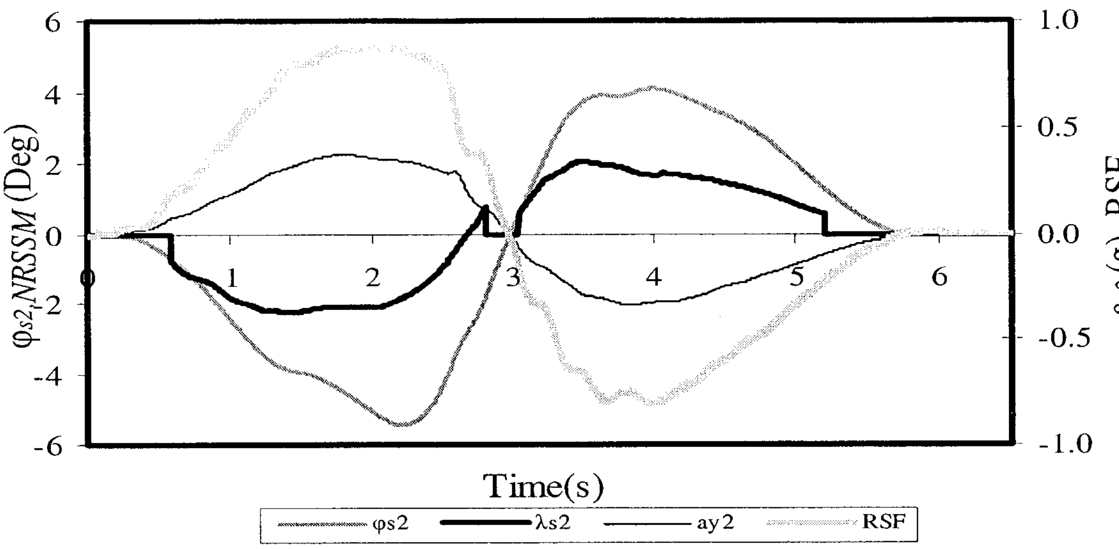
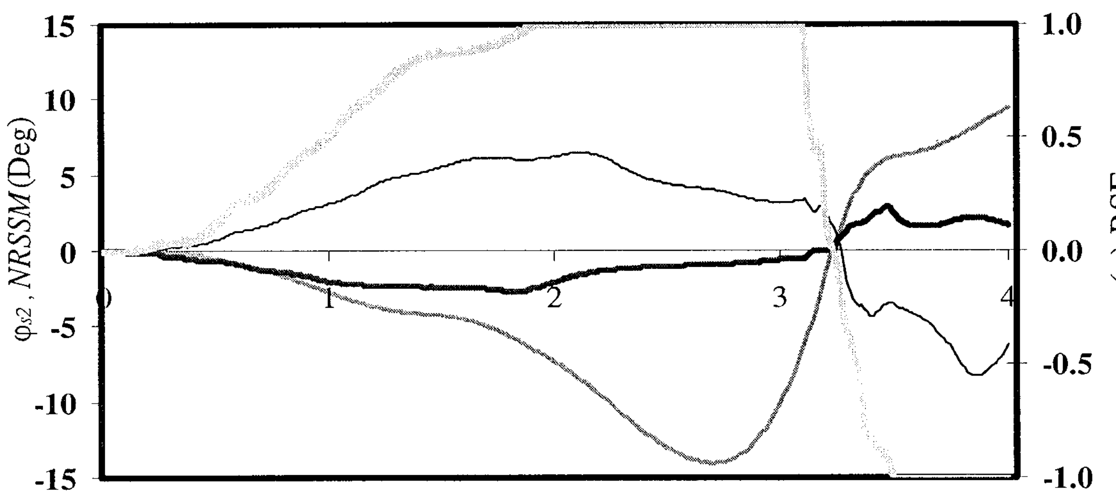


Fig 5.1: Performance of *NRSSM* at steady turning and dynamic maneuvers.



(a): Baseline Vehicle; Forward Speed < Critical Rollover Velocity



(b):Baseline Vehicle; Forward Speed > Critical Rollover Velocity

Figure 5.2: Comparison of variations in *NRSSM* response with a_{y2}, *RSF* and φ_{s2} responses

5.3 PARAMETRIC ANALYSES OF PROPOSED MEASURES OF ROLL INSTABILITY

Comprehensive parametric sensitivity analyses are performed to study the relative sensitivity of the proposed measures ($URAR_1$, $URAR_2$ and *NRSSM*) to variations in design and operating conditions. The results of the analyses are used to identify a more reliable rollover measure that exhibits least variations or sensitivity to vehicle design and operating parameters. The parametric sensitivity analyses are performed using the variations and simulation runs described in Section 4.3.2. The response variables attained corresponding to the relative rollover condition are manipulated in accordance with Equations (5.2) to (5.6) to compute the $URAR_1$, $URAR_2$ and *NRSSM.* The minimum, maximum and mean values of the proposed measures are then computed corresponding to each design and operating variable involving several simulation runs, as described in section 4.3.2.

Table 5.1 summarizes the mean, minimum and maximum values of $URAR_1$, $URAR_2$ and *NRSSM* corresponding to the relative rollover condition, characterized by *RSF* reaching unity value. It should be noted that each such variation involves three levels of cg height and three levels of steering frequencies. The simulation cases indicated in the table refer to simulation runs described in Table 4.5.

The results obtained from the simulations reveal that of the three proposed measures of impending roll instability, the unsprung mass roll amplification ratio considering the first axle ($URAR_1$) is most sensitive to variations in design as well as operating conditions. The range of values within which this ratio varies is between 5.63 and 10.66. Such high variations in the $URAR_1$ can be mostly attributed to relatively small first axle roll angle response leading to relatively high magnification in the ratio.

The results further show that this ratio is less sensitive to variations in the operating load but most sensitive to suspension parameters, specifically the damping ratio, roll center height, suspension stiffness and tire vertical stiffness. This measure is judged to be less reliable for application in an early warning roll control system.

Table 5.1: Effect of parametric variations on the $URAR_1$, $URAR_2$ and *NRSSM*

Potential Measures →	***$URAR_1$***			***$URAR_2$***			***NRSSM*** (Deg)		
Parametric Variations↓	**Min.**	**Max.**	**Avg.**	**Min.**	**Max.**	**Avg.**	**Min.**	**Max.**	**Avg.**
Baseline Vehicle (*Simulation Cases: 1-9*)	6.78	9.01	8.04	2.65	2.98	2.80	2.09	2.59	2.32
Trackwidth (*Simulation Cases: 10-18*)	7.60	9.73	8.81	2.93	3.30	3.08	2.10	2.58	2.35
Axle Tandem Spread (*Simulation Cases: 19-36*)	6.67	9.23	7.99	2.64	2.97	2.80	2.05	2.60	2.33
Tire Vertical Stiffness (*Simulation Cases: 37-63*)	5.63	8.60	7.33	2.22	2.85	2.52	2.12	2.83	2.43
Articulation Stiffness (*Simulation Cases: 64-81*)	6.66	9.59	8.12	2.60	3.12	2.82	2.06	2.63	2.33
Suspension Lateral Spread (*Simulation Cases: 82-90*)	5.88	7.58	6.77	2.34	2.67	2.50	1.99	2.30	2.17
Damping Ratio (*Simulation Cases: 91-117*)	6.55	10.66	8.18	2.50	3.01	2.84	2.02	2.61	2.34
Roll Center Height (*Simulation Cases: 118-153*)	6.20	9.34	8.02	2.51	3.16	2.83	1.97	2.69	2.32
Aux. Roll Stiffness (*Simulation Cases: 154-189*)	6.49	9.22	8.03	2.52	3.18	2.80	2.05	2.63	2.34
Suspension Stiffness (*Simulation Cases: 190-234*)	6.17	9.96	8.40	2.46	3.62	2.99	2.07	2.8	2.41
Tire Cornering Stiffness (*Simulation Cases: 235-261*)	6.69	8.86	7.85	2.48	3.01	2.79	2.10	2.76	2.38
50% Loaded Vehicle (*Simulation Cases: 261-270*)	7.02	7.78	7.43	3.66	4.73	4.31	2.08	2.58	2.36

The unsprung mass roll amplification ratio considering the second axle ($URAR_2$) for the fully loaded vehicle shows very little variation with respect to variations in the design parameters, when compared to those observed for the $URAR_1$. For a fully loaded vehicle, the peak variations in $URAR_2$ occur from 2.22 to 3.62. The $URAR_2$ measure, however, exhibits extreme sensitivity to the loading condition. The peak value of the measure approaches as high 4.73 corresponding to 50% vehicle load. This large variation is attributed to relatively lower roll angle response of the tractor rear axle under reduced load. The extreme sensitivity of this metric to variations in loading conditions suggests that the proposed measure cannot be considered as a reliable measure of the relative roll instability.

The normalized roll-response of the semitrailer sprung mass (*NRSSM*) is observed to be least sensitive to variations in the design and operating conditions. This measure can thus be considered to be most reliable amongst the measures of impending rollover. Considering all the simulations encompassing different cg heights, steering frequencies, design variations and operating condition variations, the *NRSSM* is seen to vary in the narrow range of 1.97 degrees and 2.83 degrees with an average value of 2.35 degrees. The *NRSSM* values corresponding to the relative rollover condition ($|RSF| = 1$), attained for all combinations of parameters, is shown in Figure 5.3. The results suggest that this proposed measure is least sensitive to variations in design and operating conditions considered in this study. The extreme value of 2.83 degrees arises from the variations in the vertical stiffness of the tires. A highly stiff tire yields small variations in axle roll angles and thus a larger value of the *NRSSM*. It should be noted that the simulations consider the tire vertical stiffness ranging from 788.25 kN/m to 1050 kN/m. The peak value of *NRSSM* reduces from 2.83 degrees to 2.76 degrees when the stiffness variations of the tires is limited to 963.5 kN/m. The range of variations in the *NRSSM* can thus be further reduced from 1.97 degrees to 2.76 degrees, when the variations in the tire pressure and thus the stiffness are limited.

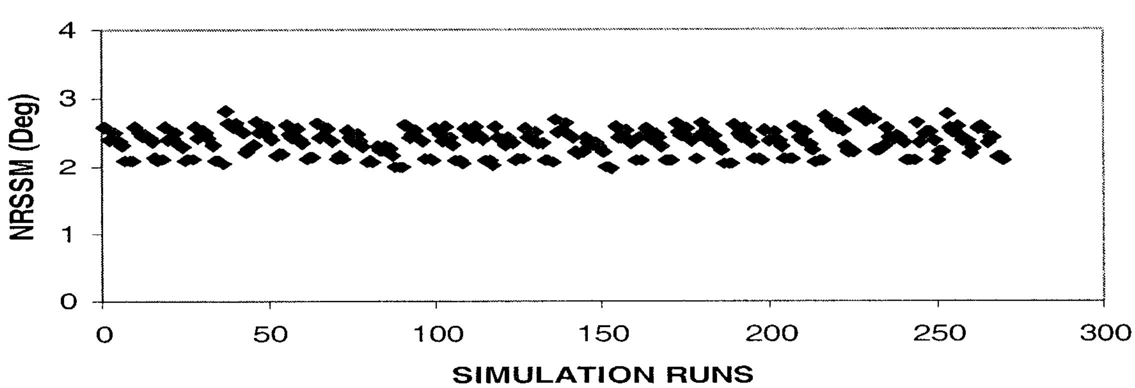


Fig. 5.3: Range of variations in *NRSSM* values over the entire range of design and operating conditions.

The results show that the Normalized Roll-response of the Semitrailer Sprung Mass is the most reliable measure of a relative roll instability of a five-axle tractor semi-trailer combination. The high degree of correlation of the *NRSSM* with *RSF*, its low sensitivity to design and operating condition variations ascertains its potential to be used as a measure of relative roll instability. The proposed measure may also be used in conjunction with an existing measure of roll instability, such as the lateral acceleration response of the semitrailer, to enhance the prediction reliability. A two stage early warning and control system may thus be developed that utilizes an existing measure of roll instability in the first stage and the proposed *NRSSM* in the second stage to detect the onset of a potential rollover.

5.4 RELATIVE ANALYSES OF THE PROPOSED AND REPORTED MEASURES OF ROLL INSTABILITY

The proposed *NRSSM* and various reported measures are further assessed to demonstrate their relative reliability in predicting the onset of a potential rollover, the lead-time performance and measurability. It has been established that the load transfer ratios (*LTR* and *RSF*) can serve as most reliable indicators of a roll instability, with the *RSF*=1 characterizing the attainment of the relative instability condition. The measurement of *LTR* and *RSF*, however, involves on-line acquisition of wheel loads of the moving vehicle, which is a formidable task and requires expensive sensors. The *RSF* and *LTR* although provide good indications of rollover propensity of the vehicle under both steady and dynamic maneuvers, their poor measurability necessitates the identification of alternate measures that are comparatively easier to measure and correlate well with the *RSF*. The semi- trailer lateral acceleration and roll angle, and roll angles of the first and second axles of the tractor, show good correlation with the *RSF* [6]. These measures may thus be considered as potential alternate measures of the relative rollover condition. These measures, however, show extreme sensitivity to variations in various design and operating parameters. While the *NRSSM* is less sensitive to such variations, its correlation with *RSF* needs to be investigated in order to deem it as an effective measure of relative roll instability.

The effectiveness of a rollover metric strongly relies upon its ability to predict the onset of roll instability in a timely manner over a range of variations in the vehicle design and operating conditions. The prediction and control strategy would further necessitate direct measurement of the metric using low-cost and reliable sensors. Three different performance measures are thus formulated to assess the proposed rollover metrics, in order to identify most feasible measures. These include the *sensitivity* and *reliability*, *measurability*, and the *lead-time*. The sensitivity and reliability of a measure is derived in terms of its variation over the range of simulation parameters considered. The lead-time performance of the measure is addressed with reference to a unit value of *RSF*, while the measurability is discussed in view of the readily available sensors for on-line acquisition/computation of the measure.

Reliability and Sensitivity of the Measures

The detection of an impending roll instability is a two step process, which involves online measurement of an indicator and its processing with reference to a preset threshold value. The reliability of the indicator would strongly rely upon the sensitivity of the threshold value with variations in design and operating parameters, and the maneuvers. High sensitivity of a threshold or index to variations in design, operating and environmental parameters could cause either false or lack of warning. A rollover indicator can be considered reliable, if its threshold value can be predicted with little error under different operating conditions, and it does not vary significantly in daily operations of the vehicle.

A few studies conducted on static and dynamic roll performance of heavy vehicles have concluded that the lateral acceleration, and sprung and unsprung mass roll angle responses vary considerably with variations in maneuvers undertaken, loading, suspension properties, roll center heights, etc. [25,36,48,70]. Moreover, a poor correlation between the tractor lateral acceleration response, and the *LTR* and *RSF* has been observed under transient maneuvers [6]. These measures would thus be considered as less reliable for detection of roll instability. It has been suggested that the semitrailer lateral acceleration correlates well with *RSF*, which has been attributed to their in-phase behavior [13]. While the *DWS* measure offers monitoring with far greater ease, its correlation with *RSF* or onset of relative rollover has not yet been established. In a two-stage warning and control process, the semitrailer lateral acceleration could be considered as a partial condition only, which when satisfied, may activate the warning on the basis of a decision made by a more reliable secondary measure. A generally applicable boundary of lateral acceleration, however, would need to be defined for initiating the warning process. The high degree of correlation of *NRSSM* with *RSF* and its in-phase relationship with the semitrailer lateral acceleration in addition to its low sensitivity to design and operating condition variations, as seen in Figures 5.2, ascertains its high potential to serve as a reliable measure for predicting onset of a relative roll instability.

Measurability

The measures based on static rollover analyses, such as lateral acceleration (*SRT*) and *SSF* offer superior measurability using inexpensive accelerometers, while those based on tire-road forces (*LTR* and *RSF*) are most difficult to measure. The measurement of *ELA* requires prior knowledge of *cg* heights of the sprung and unsprung masses, and the measurement of wheel loads, which translates into poor measurability. Although a number of dynamometers have been developed for measurement of wheel forces [64], their high costs and unproven reliability under a wide range of operating conditions remain the primary deterring factors at the present time. The *DWS* measure, on the other hand, offers superior measurability as it utilizes the electronic braking system of the vehicle and eliminates the requirement of vertical load sensors.

Table 5.2: Comparative analysis of rollover metrics in terms of measurability

METRIC	DIFFICULTY LEVEL			COMMENTS
	Low	Medium	High	
a_{y2}	X	-	-	Easily measurable using accelerometers
ELA	-	-	X	Requires measurement of vertical tire forces
LTR	-	-	X	Requires measurement of vertical tire forces
RSF	-	-	X	Requires measurement of vertical tire forces
DWS	X	-	-	Utilizes the ABS platform
NRSSM	-	X	-	Measurable using angular position sensors and micro-machined gyroscopes

The *NRSSM*, which is based on the roll angle deflections of the sprung and unsprung masses, provides relatively better measurability than the *LTR/RSF*. This measure could be acquired using angular position sensors or micro-machined position gyroscopes [58,62]. Further investigations into the silicon micro-machined micro gyroscopes, however, are required to assess their reliability in such applications. Further studies would also be needed to study the effect of road banking on the measures based on roll angles. A comparative analysis regarding measurability of various metrics is presented in Table 5.2.

Lead Time Performance

The lead-time performance of a selected indicator is highly critical for the driver to undertake a corrective action in a timely manner, which must account for delays due to human driver's perception, reaction and neuro-muscular response. The lead-time performance of the proposed measures has been addressed only in a few studies. Through analyses of directional dynamics responses of articulated vehicles, it has been concluded that different phase relationships exist among different response measures [6]. The study suggested that the semitrailer lateral acceleration and the roll angle lag the *RSF*, while the tractor axle roll angles lead the *RSF*. The early warning capability of the reported and identified response parameters under an array of variations in design and operating conditions reveal that the axle roll angles of the tractor front and rear axles (φ_{u1} and φ_{u2}) could provide considerable lead-time, when compared to that of the roll angle of the semitrailer sprung mass, as can be seen in Figure 5.4.

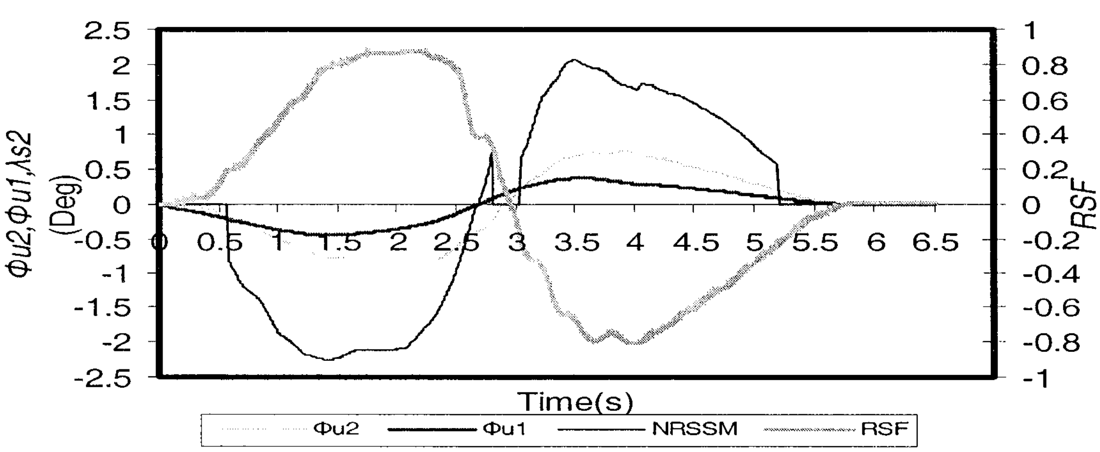


Figure 5.4: Relative lead-time performance of different measures (0.25 Hz sinusoidal steering).

The figure shows the time histories of the tractor axles roll angles, *NRSSM* and *RSF* responses of the baseline vehicle subjected to a 0.25Hz sinusoidal steering at a forward speed below the critical rollover speed. The results show that the magnitude of *NRSSM* with the magnitude of *RSF*, although opposite in direction. Moreover, the *NRSSM* leads the *RSF* considerably, and could thus provide reasonably good lead-time.

The online monitoring of semitrailer lateral acceleration also provides appreciable lead time, when compared to the semitrailer roll angle, as is evident from Figure 5.5. The *RSF,* however, normally leads the semitrailer lateral acceleration marginally. The *RSF* was found to lead all the measures of angular roll deflection except for the tractor front axle roll angle. The *NRSSM* response was generally found to lead the *RSF* and the semitrailer lateral acceleration as the vehicle approaches a relative roll instability condition as shown in Figure 5.6 (forward speed > critical rollover speed). It can thus be inferred that *NRSSM* provides superior lead-time, when compared to most of the measures of roll instability [81].

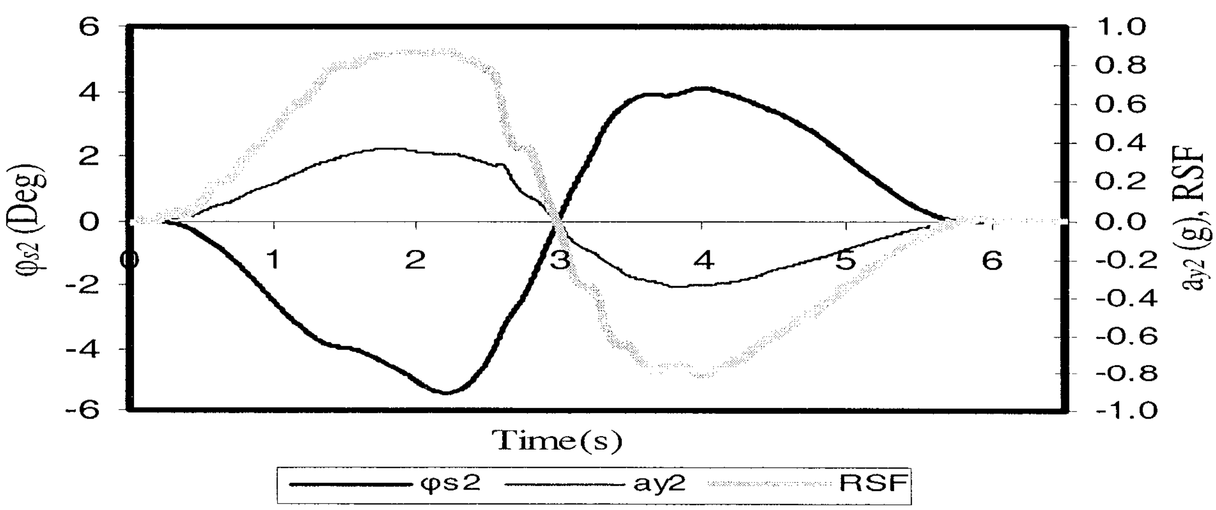


Figure 5.5:Phase relationship *of* a_{y2}, φ_{s2} and *RSF* responses (speed = 75 km/h; steering frequency = 0.25Hz).

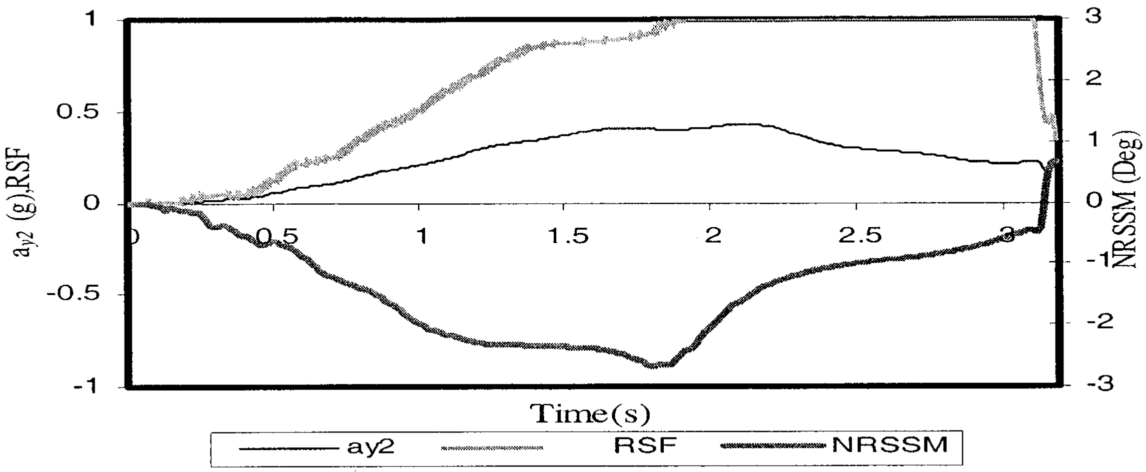


Fig. 5.6: Comparison of *NRSSM*, a_{y2} and *RSF* response of the baseline vehicle (0.25 Hz steering input; forward speed > critical rollover speed).

The relative lead-time performance of different measures as observed from the wide-range of simulations performed in this study is further summarized in Table 5.3, in a qualitative manner. The lag/lead of a measure in a given row is listed with respect to the measures in different columns while that in relation to *RSF* is presented in the highlighted column. From the results obtained from the simulations it can be inferred that the tractor rear axle roll angle and semitrailer sprung mass roll angle normally lag the *RSF*. The tractor front axle roll angle and *NRSSM* lead the *RSF* and provide appreciable lead-time for early warning, while the lateral acceleration response of the semitrailer sprung mass lags the *RSF*, albeit marginally.

Table 5.3: A comparison of relative lag/lead performance of different rollover metrics.

Rollover Measures	a_{y2}	*RSF*	φ_{s2}	φ_{u1}	φ_{u2}	λ_{s2}
a_{y2} →	x	lag	lead	lag	lead	lag
RSF →	lead	x	lead	lag	lead	lag
φ_{s2} →	lag	lag	x	lag	lag	lag
φ_{u1} →	lead	lead	lead	x	lead	lead
φ_{u2} →	lag	lag	lead	lag	x	lag
λ_{s2} →	lead	lead	lead	lag	lead	x

5.5 IDENTIFICATION OF FEASIBLE ROLLOVER METRICS

The measurability, reliability and lead-time analysis of the measures of roll instability show that each measure of roll instability has its merits and limitations. However in the endeavor of developing a comprehensive and effective early warning and control system, a feasibility analysis needs to be performed to obtain a set of measures that are most effective and can be employed in a early warning and control strategy.

The simulation results suggest that *RSF* directly relates to the relative roll instability condition of a vehicle. Owing to its poor measurability, this measure cannot be applied for on-line detection. The semitrailer lateral acceleration response yields good correlation with *RSF* and *NRSSM* (Figures 5.2, 5.4, 5.5 and 5.6), and can be measured with greater ease. Its reliability, however, is considered to be poor due to its high sensitivity to variations in cg height and rate of steering input, and moderate sensitivity to variations in various design and operating conditions. The semitrailer sprung mass and axles' roll angle responses also revealed reasonably good correlation with *RSF* (Figures 5.2, 5.4 and 5.5), and relatively low sensitivity to variations in design and operating conditions under rated axle loads. These measures, however, revealed greater sensitivity to variations in the axle loads.

The proposed *NRSSM* measure correlates very well with *RSF* and is seen to be in phase with it, while it is relatively insensitive to wide range of variations in design and operating conditions as seen in Figure 5.3, 5.4 and 5.6. Each simulation run in Figure 5.3 refers to a specific combination of parameters considered within the simulation matrix comprising 279 different combinations. The results indicate that the variations in *NRSSM* over the entire range of parameters lie within a constricted range. This measure could thus be considered most reliable due to its least sensitivity to design and operating conditions. The figure shows that the variations in *NRSSM* over the entire range of design and operating parameters considered, lie within a narrow band, from 1.97 to 2.83 degrees with mean and median values being 2.35 and 2.39 degrees, respectively (Table 5.4). The standard deviation of the measure is obtained as 0.195 degrees with a low coefficient of variation of 0.027.

Table 5.4: Mean, mode and range of *NRSSM* values

Salient Values	NRSSM (Degrees)
Minimum Value	1.97
Maximum Value	2.83
Mean Value	2.35
Median Value	2.39
Standard Deviation	0.195
Coefficient of Variation	0.027

The *NRSSM* offers not only better measurability than the measures based upon the measurement of tire loads but also enhanced lead-time performance with respect to *LTR*, *RSF* and roll angle responses of the individual sprung and unsprung units, as evident from Figures 5.2, 5.4, 5.5 and 5.6. The results also show that the front axle roll angle and the *NRSSM* lead the lateral acceleration response normally by approximately 0.35s and 0.3s, respectively, while the roll angle of the tractor rear axle lags the acceleration response by 0.1s on an average.

The high degree of correlation between *NRSSM* and *RSF,* and its low sensitivity to variations in design and operating condition variations ascertains its high potential to serve as a reliable measure for predicting onset of a relative roll instability condition. The semitrailer lateral acceleration response, which although lags the *RSF* marginally, shows good correlation with the *RSF* and thus the onset of roll instability of the vehicle. It provides superior measurability by means of robust and inexpensive accelerometers that present accurate measurement of the lateral acceleration experienced by the vehicle. The reliability, measurability, lead-time and feasibility of the prediction may be thus further enhanced by formulating a two-stage strategy incorporating both these measures i.e. *NRSSM* and a_{y2} [81].

5.6 SUMMARY

A set of potential indicators of roll instability is identified based on the results obtained from the simulation matrix described in Chapter 4. The Rearward Amplification in Roll of the sprung masses (RAR_{φ}), Unsprung Mass Rearward Amplification Ratio considering the tractor front axle ($URAR_1$), the Unsprung mass Rearward Amplification Ratio considering the tractor rear axle ($URAR_2$) and the Normalized Roll-Response of the Semitrailer Sprung Mass (*NRSSM*), are proposed as potential measures of impending roll instability. The reliability of these measures in predicting onset of a rollover is assessed through extensive parametric sensitivity analyses under varying design and operating conditions. The analysis showed that the *NRSSM* could serve as the most reliable indicator of roll instability. The proposed normalized roll-response of the semi-trailer sprung mass is further assessed in terms of the performance measures, such as measurability, reliability, lead-time and feasibility. On the basis of the relative analysis, it is inferred that the semi-trailer sprung mass lateral acceleration (a_{y2}) and the normalized roll-response of the semi-trailer sprung mass (*NRSSM*) are the two most comprehensive measures of impending roll instability, which provide superior lead-time, and can be considered measurable. A two-stage prediction and warning algorithm may thus be realized on the basis of on-line monitoring of a_{y2} and *NRSSM*. The concept of this two-stage monitor together with the threshold values is realized in the following chapter.

CHAPTER 6

THE ROLLOVER EARLY WARNING AND CONTROL STRATEGY AND RELATED HUMAN FACTOR ISSUES

6.1 INTRODUCTION

The prevention of rollover of heavy vehicles in an open-loop manner could be realized in three systematic stages: (i) detection of onset of a potential rollover through online monitoring of a selected rollover metric; (ii) signal processing and generation of a warning for the driver; (iii) a corrective action to be undertaken by the driver. The first stage involves the selection of a rollover metric that could reliably predict the onset of a potential instability with reasonable lead-time for the driver. The second stage forms the most important interface between the dynamic state of the vehicle and the driver's perception of the warning, while the final stage is limited to the driver's actions in the form of steering and/or braking.

The generation of an early warning for the driver to induce a corrective action requires electronic systems that utilize sensors, signal processors and warning devices [10,57]. It has been suggested that around forty percent of the rollover related accidents involving commercial vehicles could be prevented using rollover warning systems [40]. The roll advisory system suggested by Ehlbeck [10] can be considered as a reference for developing a comprehensive rollover early warning and control system that would utilize either driver-in-the-loop or an active control system. In either case, the driver's perception of the impending roll instability of the vehicle through the mode of warning forms an integral part of the control strategy.

A warning system must predict the onset of a rollover with greater reliability, with minimal false occurrences. The prediction of an impending rollover on the basis of the relative rollover condition may be considered to be somewhat conservative, since the restoring moments developed by the front axle suspension and tires is ignored. The design of a fail-safe system would also require the identification of threshold values corresponding to a *RSF* value slightly below unity. This may cause false warnings under some transient maneuvers, which tend to restore the tire road contact, when the steering direction is reversed. The preservation of adequate margin of safety and minimizing the frequency of false warnings thus poses a complex challenge.

The results presented in the previous chapters suggest that the normalized roll-response of the semitrailer sprung mass (*NRSSM*) and the semitrailer lateral acceleration could predict the potential roll instability with greater reliability and lead-time. The assessment of the reliability of these measures would require further analyses to establish the ranges of their variations. This would help identify the respective threshold values corresponding to acceptable safety margin and tolerance for possible false warnings. In this regard, the human factors issues associated with an effective delivery of the warning signal in both audio as well as visual means need to be explored.

In this chapter the variations in the *NRSSM* and the semitrailer lateral acceleration responses are investigated corresponding to different values of safety margin expressed in terms of various percentages of *RSF*. The results are analyzed to identify the lower limits and the average values of these parameters at various stages of relative roll instability. The lead-time capability of these measures at varying conditions of relative roll instability is also analyzed. An early warning and control algorithm is proposed that would issue a two stage audio-visual warning to the driver and would also be capable of activating a control system to arrest the roll instability.

6.2 IDENTIFICATION OF THE ROLLOVER THRESHOLD VALUES

The threshold values of the selected metrics must be chosen to ensure high reliability with acceptable safety margin and to reduce the frequency of false occurrences. Considering that the *RSF* directly relates to the relative rollover condition and thus a potential roll instability, the margin of safety can be selected by considering different values of *RSF*. The threshold values, however, must be selected at *RSF* values slightly below unity in order to avoid the false warnings. For this purpose, the variations in the selected metrics, semitrailer lateral acceleration (a_{y2}) and *NRSSM*, are evaluated corresponding to 85%, 90%, 95% and 100% *RSF* values in order to identify the ranges of respective threshold values. The results are further evaluated to assess the lead-time, when these measures approach the respective stages of relative roll instability indicated by the percentage of *RSF*. The simulations are performed under sinusoidal steering inputs at three different frequencies of *0.25Hz, 0.33Hz* and *0.50Hz*. The a_{y2} and *NRSSM* responses corresponding to the selected values of *RSF* are derived for the baseline vehicle. The variations in the measures are further evaluated under variations in the design and operating conditions described in Chapter 4. The results are evaluated to identify the lead-time determined with respect to *RSF=1*.

6.2.1 Baseline Vehicle

The simulation results attained for the baseline vehicle subjected to sinusoidal steering inputs are investigated to derive the semitrailer lateral acceleration (a_{y2}) and *NRSSM* responses corresponding to different values of *RSF*. Table 6.1 illustrates the variations in the values of a_{y2} and *NRSSM*, and the lead-time corresponding to different values of *RSF*. The variations are summarized in terms of respective minimum, maximum and mean values together with the standard deviation and coefficient of variation *(COV)*. The results represent the variations in the respective measures for the baseline vehicle subject to sinusoidal steering at different frequencies *(0.25Hz,0.33Hz* and *0.50Hz)*, while three different cg heights (1.8m,2.0m and 2.2m) are also considered, as described in Chapter 4. It should be noted that the forward speed is varied until the rollover condition is realized (*RSF=1*).

Table 6.1: Variations in a_{y2}, *NRSSM* and lead time corresponding to different stages of *RSF* limits (Baseline Vehicle)

Measure→	a_{y2}(g)				λ_{s2} (Deg)				*Lead-Time* (s)			
RSF →	85%	90%	95%	100%	85%	90%	95%	100%	85%	90%	95%	100%
Minimum	0.312	0.347	0.378	0.379	2.207	2.130	2.164	2.09	0.343	0.217	0.082	0
Maximum	0.416	0.475	0.550	0.578	2.430	2.546	2.723	2.59	0.582	0.334	0.117	0
Average	0.356	0.409	0.440	0.464	2.318	2.394	2.512	2.32	0.474	0.275	0.093	0
Std. Deviation	0.038	0.045	0.059	0.059	0.076	0.134	0.189	0.189	0.078	0.051	0.010	0
COV	0.107	0.111	0.133	0.127	0.032	0.056	0.075	0.081	0.166	0.189	0.106	0

The results suggest considerable variations in the a_{y2} response, ranging from 0.379g to 0.578g, corresponding to *RSF*=1, with *COV* of 0.127. The *COV* value reduces to 0.107, when *RSF*=0.85 is considered. The variations in the *NRSSM* are observed to be relatively lower, where a *COV* of 0.081 is attained at *RSF*=1. The *COV* value reduces to 0.032 at *RSF*=0.85. The lead-time is also observed to be maximum at 85%*RSF* and decreases with increase in the percentage of *RSF*. These results therefore show that the variations in these measures could be considerably reduced when a lower value of *RSF* (0.85) is chosen. This *RSF* value thus could enhance the prediction strategy based upon these measures and further provide a reasonably good margin of safety coupled with low coefficient of variation. The results further reveal that the lead-time performance of the measures could be significantly enhanced by selecting *RSF* of 0.85. The results summarized in Table 6.1 could be applied to identify the threshold values of the selected measures, which would provide acceptable safety margin and lead-time. For the baseline vehicle, the results corresponding to 85%*RSF* reveal that the a_{y2} and *NRSSM* responses could range from 0.312g to 0.416g, and 2.207 degrees to 2.43 degrees, respectively, under variations in cg height and steering rates considered. The performance of the identified measures corresponding to variations in design and operating conditions would yield further insight into the values attained by these measures at different levels of relative roll instability.

6.2.2 Influence of Variations in the Design Parameters

The acceptance of threshold values as potential indicators strongly relies upon their sensitivities to variations in design and operating conditions. The sensitivity of the identified measures and lead-time values to variations in design parameters, described in Chapter 4, are evaluated at different stages of relative roll instability in order to identify more reliable threshold values. The results attained are described in the following sections:

Trackwidth

Table 6.2 illustrates the variations in a_{y2} and *NRSSM* when the trackwidth is increased from 2.44m to 2.59m. It is observed that the coefficient of variation is least for *NRSSM* at 85%*RSF* and at the same time the lead-time is the highest. The range of variations in a_{y2} spans from 0.344g to 0.48g with an average of 0.398g. The *COV* at 90% *RSF* is seen to be least while considering the semitrailer's lateral acceleration response. The standard deviation at 85% *RSF* and 90% *RSF* are seen to be almost similar. The *NRSSM* ranges from 2.124 degrees to 2.382 degrees with an average of 2.295 degrees at 85% *RSF*. The variations in *NRSSM* considering different stages of relative roll instability, when the trackwidth is varied in addition to variations in cg height and steering frequencies, are found to be low, thereby reinforcing its potential to be considered as a reliable rollover metric.

Table 6.2: Variations in a_{y2}, *NRSSM* and lead time corresponding to different stages of *RSF* limits with variations in trackwidth.

Measure→	a_{y2} (g)				λ_{s2} (Deg)				*Lead-Time* (s)			
RSF →	85%	90%	95%	100%	85%	90%	95%	100%	85%	90%	95%	100%
Minimum	0.344	0.409	0.403	0.406	2.124	1.981	2.243	2.110	0.38	0.16	0.08	0
Maximum	0.48	0.544	0.608	0.596	2.382	2.713	2.770	2.580	0.517	0.263	0.14	0
Average	0.398	0.462	0.477	0.493	2.295	2.433	2.547	2.357	0.443	0.188	0.098	0
Std. Deviation	0.051	0.047	0.064	0.062	0.097	0.233	0.206	0.185	0.042	0.032	0.019	0
COV	0.130	0.101	0.135	0.127	0.042	0.096	0.081	0.078	0.095	0.172	0.198	0

Suspension Stiffness

The choice of suspension springs is seen to have noteworthy influence on a_{y2} and *NRSSM* at different values of *RSF*. The *NRSSM* values range from 1.952 degrees to 2.475 degrees with average value of 2.242 at 85%*RSF* with standard deviation of 0.120. The coefficient of variation is seen to be minimum at 85%*RSF* in case of *NRSSM*, and at

90%*RSF* for the a_{y2}, as seen in Table 6.3. The standard deviation of variations in a_{y2} appear to be similar at 85% *RSF*.

Table 6.3: Variations in a_{y2}, *NRSSM* and lead time corresponding to different stages of *RSF* limits with variations in suspension stiffness

Measure→	a_{y2} (g)				λ_{s2} (Deg)				*Lead-Time* (s)			
SF →	85%	90%	95%	100%	85%	90%	95%	100%	85%	90%	95%	100%
Minimum	0.319	0.344	0.368	0.372	1.952	2.004	1.983	2.075	0.273	0.163	0.060	0
Maximum	0.447	0.485	0.554	0.594	2.475	2.624	2.831	2.818	0.675	0.480	0.243	0
Average	0.369	0.406	0.440	0.459	2.242	2.317	2.495	2.414	0.453	0.305	0.121	0
Std. Deviation	0.040	0.040	0.050	0.560	0.120	0.155	0.201	0.204	0.096	0.092	0.050	0
COV	0.110	0.099	0.14	0.121	0.053	0.067	0.082	0.090	0.211	0.304	0.411	0

Articulation Roll Stiffness

Considering variations in articulated roll stiffness, it can be observed from Table 6.4 that the *NRSSM* as well as a_{y2} show the least variation in their values at 85%*RSF*. The *NRSSM* ranges from 2.221 degrees to 2.440 degrees with an average of 2.304 degrees at 85%*RSF*, while the a_{y2} response ranges from 0.312g to 0.419g with an average of 0.365 g. The average lead-time available is also seen to be 0.476s and the standard deviation in *NRSSM* as well as a_{y2} is seen to be minimum in case of *RSF* value of 0.85.

Table 6.4: Variations in a_{y2}, *NRSSM* and lead time corresponding to different stages of *RSF* limits with variations in articulation roll stiffness.

Measure→	a_{y2} (g)				λ_{s2} (Deg)				*Lead-Time* (s)			
RSF →	85%	90%	95%	100%	85%	90%	95%	100%	85%	90%	95%	100%
Minimum	0.312	0.348	0.377	0.377	2.221	2.126	2.147	2.067	0.345	0.191	0.060	0
Maximum	0.419	0.476	0.549	0.578	2.440	2.583	2.771	2.635	0.644	0.320	0.124	0
Average	0.356	0.409	0.441	0.455	2.304	2.398	2.523	2.334	0.476	0.262	0.094	0
Std.Deviation	0.037	0.043	0.056	0.059	0.075	0.135	0.194	0.183	0.082	0.049	0.015	0
COV	0.106	0.107	0.127	0.131	0.032	0.564	0.077	0.078	0.174	0.187	0.167	0

Damping Ratio

The relative variations in the lateral acceleration and *NRSSM* responses of the combination with variations in axle suspension damping ratio are observed to be similar to those obtained under variations in many other design variables, as summarized in Table 6.5. While the *NRSSM* is seen to possess average values of 2.334 degrees and 2.345 degrees at 85%*RSF* and 100%*RSF*, respectively, the lateral acceleration response is seen to vary in a broader range. The results, however, yield lower coefficient of variation in *NRSSM* and higher lead-time at 85% *RSF*, as compared to the other stages of relative roll instability.

Table 6.5: Variations in a_{y2}, *NRSSM* and lead time corresponding to different stages of *RSF* limits with variations in damping ratio.

Measure→	a_{y2} (g)				λ_{s2} (Deg)				*Lead-Time* (s)			
RSF →	85%	90%	95%	100%	85%	90%	95%	100%	85%	90%	95%	100%
Minimum	0.281	0.333	0.379	0.378	2.093	2.154	2.237	2.022	0.376	0.102	0.061	0
Maximum	0.450	0.522	0.559	0.605	2.682	2.834	2.83	2.610	0.625	0.373	0.247	0
Average	0.350	0.415	0.441	0.458	2.334	2.443	2.563	2.345	0.465	0.254	0.115	0
Std.Deviation	0.049	0.050	0.052	0.554	0.158	0.195	0.183	0.197	0.069	0.057	0.042	0
COV	0.143	0.121	0.119	0.132	0.067	0.079	0.071	0.084	0.147	0.223	0.379	0

Auxiliary Roll Stiffness

The effect of variations in auxiliary roll stiffness on the identified measures and lead time at different levels of *RSF* is shown in Table 6.6. It can be seen that at 85%*RSF*, the coefficient of variation is the lower than those for the 95%*RSF* and 100%*RSF*. Selection of a relative rollover condition near 90%*RSF* yields slightly lower *COV* value than that attained for 85%*RSF*, while the corresponding standard deviation is higher. The results summarized in Table 6.6 suggest that the variations in auxiliary roll stiffness do not significantly affect the performance of the identified measures at different values of

RSF. The average values of a_{y2} and *NRSSM* are seen to vary from 0.360g to 0.456g, and 2.311 degrees to 2.341 degrees, respectively, when the relative roll instability condition is varied from *RSF*=0.85 to 1.0. The average lead-time is seen to be maximum in case of 85%*RSF* at 0.471 seconds.

Table 6.6: Variations in a_{y2}, *NRSSM* and lead time corresponding to different stages of *RSF* limits with variations in auxiliary roll stiffness.

Measure→	a_{y2} (g)				λ_{s2} (Deg)				*Lead-Time* (s)			
RSF →	85%	90%	95%	100%	85%	90%	95%	100%	85%	90%	95%	100%
Minimum	0.309	0.348	0.365	0.375	2.121	2.120	2.042	2.051	0.342	0.131	0.040	0
Maximum	0.437	0.492	0.558	0.584	2.601	2.770	2.821	2.632	0.643	0.422	0.151	0
Average	0.360	0.411	0.442	0.456	2.311	2.402	2.542	2.341	0.471	0.263	0.090	0
Std.Deviation	0.039	0.043	0.055	0.061	0.121	0.185	0.202	0.179	0.070	0.070	0.021	0
COV	0.108	0.106	0.124	0.134	0.052	0.077	0.079	0.076	0.153	0.260	0.221	0

Roll Center Height

The height of the roll center above the ground surface, when varied, shows reasonable influence on the ranges of values of the identified measures at every stage of relative roll instability considered. Variations in the roll center height result in *NRSSM* values ranging from 1.960 degrees to 2.731 degrees at 85%*RSF*, while the a_{y2} values range from 0.304 to 0.433g at the same level of relative roll instability, as illustrated in Table 6.7. The average lead-time corresponding to 85%RSF is seen to be 0.463 s.

Table 6.7: Variations in a_{y2}, *NRSSM* and lead time corresponding to different stages of *RSF* limits with variations in roll center height.

Measure→	a_{y2} (g)				λ_{s2} (Deg)				*Lead-Time* (s)			
RSF →	85%	90%	95%	100%	85%	90%	95%	100%	85%	90%	95%	100%
Minimum	0.304	0.351	0.362	0.373	1.960	1.962	1.861	1.9722	0.347	0.164	0.071	0
Maximum	0.433	0.494	0.541	0.580	2.731	2.862	2.894	2.691	0.695	0.461	0.213	0
Average	0.352	0.401	0.432	0.453	2.312	2.411	2.501	2.325	0.463	0.272	0.113	0
Std.Deviation	0.038	0.043	0.051	0.064	0.194	0.25	0.246	0.192	0.082	0.083	0.031	0
COV	0.107	0.105	0.118	0.131	0.083	0.105	0.098	0.082	0.175	0.299	0.286	0

Suspension Lateral Spread

The increase in suspension lateral spread above the baseline value yields moderate effect on the magnitudes of *NRSSM*, irrespective to the relative rollover condition chosen. The effect is, however, relatively greater on the semitrailer's lateral acceleration response (Table 6.8). The *COV* of the *NRSSM* values is seen to be lower than a_{y2} thereby translating to higher reliability of *NRSSM*. At higher levels of *RSF*, greater variation in the *COV* values is generally observed for both measures, thereby affirming 85%*RSF* as a suitable stage for generating the early warning .

Table 6.8: Variations in a_{y2}, *NRSSM* and lead time corresponding to different stages of *RSF* limits with variations in suspension lateral spread.

Measure→	a_{y2} (g)				λ_{s2} (Deg)				*Lead-Time* (s)			
RSF →	85%	90%	95%	100%	85%	90%	95%	100%	85%	90%	95%	100%
Minimum	0.312	0.347	0.378	0.379	1.43	1.94	2.07	1.99	0.34	0.19	0.08	0
Maximum	0.435	0.478	0.55	0.582	2.178	2.54	2.72	2.59	0.60	0.33	0.10	0
Average	0.361	0.418	0.444	0.463	1.94	2.276	2.424	2.246	0.461	0.261	0.097	0
Std.Deviation	0.038	0.041	0.053	0.058	0.154	0.178	0.194	0.178	0.075	0.075	0.010	0
COV	0.106	0.098	0.120	0.125	0.070	0.075	0.080	0.079	0.164	0.164	0.110	0

Axle Spread

Table 6.9 illustrates the performance of the identified measures when subjected to variations in the axle tandem spread, which would be applicable for the tractor rear and semitrailer axles. The range of variations in the identified measures, when subjected to variations in axle tandem spread (1.2m, 1.5m & 1.8m), is seen to be moderate. The minimum and maximum values of *NRSSM* are attained as 2.192 and 2.460, while those for a_{y2} are 0.313g and 0.422g at 85%*RSF*. The effect of variations in axle tandem spread is seen to be more profound in case of a_{y2} and relatively small in case of *NRSSM*. At

85%*RSF*, the *COV* value of the *NRSSM* is merely 0.032 corresponding to 85%*RSF* and increases to 0.082 for 100%*RSF*. The average lead-time at 85%*RSF* is found to be 0.464s.

Table 6.9: Variations in a_{y2}, *NRSSM* and lead time corresponding to different stages of *RSF* limits with variations in axle tandem spread.

Measure→	a_{y2}(g)				λ_{s2} (Deg)				*Lead-Time* (s)			
RSF →	85%	90%	95%	100%	85%	90%	95%	100%	85%	90%	95%	100%
Minimum	0.313	0.301	0.372	0.383	2.192	2.111	2.141	2.052	0.340	0.193	0.082	0
Maximum	0.422	0.475	0.550	0.583	2.460	2.611	2.75	2.613	0.602	0.334	0.112	0
Average	0.351	0.401	0.442	0.450	2.301	2.392	2.522	2.331	0.464	0.269	0.095	0
Std.Deviation	0.039	0.049	0.056	0.059	0.076	0.138	0.197	0.191	0.069	0.047	0.009	0
COV	0.110	0.123	0.127	0.130	0.032	0.057	0.078	0.082	0.149	0.175	0.102	0

Tire Vertical Stiffness and Cornering Stiffness

The variations in the tire vertical stiffness and cornering stiffness also yield relatively small changes in the magnitudes of *NRSSM*. Tables 6.10 and 6.11 illustrate the variations in *NRSSM*, a_{y2} and the lead-time, when these design parameters are varied. The results show considerable variations in a_{y2} value at different stages of *RSF*, while the *NRSSM* values remain relatively less sensitive. The standard deviation of the variations in a_{y2}, however, is reasonably small ranging from 0.034g to 0.050g for 85%*RSF* to 100%*RSF*. At 85%*RSF*, the maximum lead-time available is seen to be quite high at 0.813 seconds when the tire cornering stiffness is varied. The value of *NRSSM* varies in a narrow band in both cases at different percentages of *RSF* as compared to the value of a_{y2}. The *COV* values in both cases are seen to increase with increase in the percentage of *RSF*, and least variations could be attained near 85%*RSF*.

Table 6.10: Variations in a_{y2}, *NRSSM* and lead time corresponding to different stages of *RSF* limits with variations in tire vertical stiffness.

Measure→	a_{y2}(g)				λ_{s2} (Deg)				*Lead-Time* (s)			
RSF →	85%	90%	95%	100%	85%	90%	95%	100%	85%	90%	95%	100%
Minimum	0.312	0.354	0.348	0.374	2.271	2.223	2.214	2.122	0.310	0.221	0.093	0
Maximum	0.416	0.475	0.537	0.577	2.712	2.823	2.984	2.831	0.620	0.412	0.151	0
Average	0.353	0.401	0.430	0.450	2.451	2.556	2.661	2.430	0.471	0.294	0.121	0
Std.Deviation	0.034	0.041	0.057	0.058	0.123	0.147	0.194	0.194	0.084	0.50	0.017	0
COV	0.098	0.102	0.132	0.129	0.050	0.057	0.073	0.079	0.179	0.172	0.147	0

Table 6.11: Variations in a_{y2}, *NRSSM* and lead time corresponding to different stages of *RSF* limits with variations in tire cornering stiffness.

Measure→	a_{y2}(g)				λ_{s2} (Deg)				*Lead-Time* (s)			
RSF →	85%	90%	95%	100%	85%	90%	95%	100%	85%	90%	95%	100%
Minimum	0.313	0.356	0.369	0.375	2.192	1.971	2.150	2.113	0.322	0.214	0.081	0
Maximum	0.422	0.474	0.529	0.609	2.550	2.712	2.842	2.761	0.813	0.472	0.201	0
Average	0.356	0.405	0.434	0.455	2.365	2.425	2.558	2.385	0.484	0.296	0.117	0
Std.Deviation	0.034	0.041	0.050	0.061	0.104	0.153	0.187	0.175	0.111	0.065	0.033	0
COV	0.096	0.102	0.115	0.134	0.044	0.063	0.073	0.073	0.230	0.219	0.286	0

The results summarized in tables 6.2 to 6.11 suggest that the selection of a relative rollover condition near 85%*RSF* would help reduce the variations in the a_{y2} and *NRSSM* measures caused by variations in various design and operating parameters. Moreover, the 85%*RSF* condition generally yields higher lead-time performance. The selection of 85%*RSF* would thus be most appropriate for generation of early warning on the onset of a potential rollover. The added safety margin, however, could yield higher risk of false warnings, which maybe addressed through a two-stage warning algorithm, discussed in the following sections.

6.2.3 Influence of Variations in the Operating Parameters

The influences of variations in the operating parameters, such as forward speed and operating load on the relative changes in both measures, a_{y2} and *NRSSM*, are further investigated to asses their prediction reliability and lead-time performance. The results are attained under variations in the operating load, identical to those described in section 4.3.2. The variations in the forward speed include the critical rollover speed and a speed of 20kmph above the critical speed. Table 6.12 summarizes the ranges of variations in the measures and the lead-time due to variations in the operating speed. The results reveal that an increase in the operating velocity by 20kmph does not significantly affect the values attained by these parameters or cause a significant change in the available lead-time when compared to the baseline vehicle operating at critical rollover velocity.

Table 6.12: Variations in a_{y2}, *NRSSM* and lead time corresponding to different stages of *RSF* limits with variations in operating velocity.

Measure→	a_{y2}(g)				λ_{s2} (Deg)				*Lead-Time* (s)			
RSF →	85%	90%	95%	100%	85%	90%	95%	100%	85%	90%	95%	100%
Minimum	0.321	0.358	0.391	0.421	2.230	2.184	2.293	2.190	0.331	0.212	0.080	0
Maximum	0.419	0.455	0.539	0.602	2.511	2.712	2.980	2.842	0.513	0.322	0.120	0
Average	0.357	0.400	0.454	0.485	2.361	2.498	2.723	2.584	0.413	0.275	0.093	0
Std.Deviation	0.038	0.041	0.060	0.062	0.108	0.169	0.238	0.252	0.060	0.047	0.012	0
COV	0.106	0.102	0.133	0.128	0.046	0.067	0.087	0.097	0.146	0.173	0.131	0

Table 6.13 illustrates the effect of variations in the operating load on the ranges of the identified measures and the effect on the lead-time. The effect of load variations on the *NRSSM* at relative roll instability condition characterized by *RSF=1,* has already been established in the previous chapter. The peak magnitude of *NRSSM* remains above 2.0 degrees in all cases, with the exception of the 85%*RSF* case. The minimum values of *NRSSM* at fractional relative roll stability conditions, however, are generally lower than 2 degrees. A reduction in the load carried by the vehicle to 50% of the rated payload is seen to influence the lateral acceleration and *NRSSM* values significantly. The magnitudes of *NRSSM* and a_{y2} range from 0.306g to 0.506g and 1.471 degrees to 1.901 degrees,

respectively at 85%*RSF*. Higher variations in the *NRSSM* values, however, are obtained under 100% *RSF* conditions, while the corresponding variations in a_{y2} are relatively smaller. These results further suggest the use of both measures for developing a reliable predictor of the onset of a rollover, and a safety margin corresponding to 85%*RSF*.

Table 6.13: Variations in a_{y2}, *NRSSM* and lead time corresponding to different stages of *RSF* limits with variations in operating load.

Measure→	a_{y2}(g)				λ_{s2} (Deg)				*Lead-Time* (s)			
RSF →	85%	90%	95%	100%	85%	90%	95%	100%	85%	90%	95%	100%
Minimum	0.306	0.413	0.432	0.448	1.471	1.634	1.752	2.080	0.213	0.142	0.091	0
Maximum	0.506	0.523	0.536	0.569	1.901	2.214	2.491	2.582	0.561	0.312	0.181	0
Average	0.394	0.468	0.484	0.497	1.692	1.993	2.237	2.236	0.384	0.203	0.117	0
Std.Deviation	0.067	0.037	0.038	0.037	0.155	0.191	0.243	0.196	0.097	0.051	0.030	0
COV	0.172	0.080	0.078	0.075	0.091	0.096	0.108	0.083	0.253	0.254	0.260	0

The simulation results show that the lead-time performance of a early warning and control algorithm could be enhanced by selecting a relatively lower relative rollover criterion near 85%*RSF*. The coefficient of variation of the *NRSSM* response is seen to be the least at 85%*RSF* in most cases. The minimum values of the lateral acceleration responses of the vehicle at 85%*RSF* is almost always seen to be around 0.30g, irrespective of the variations in the design and operating conditions considered in the study. It should be noted that identical variations in the cg heights of 1.8m, 2.0m and 2.2m, were retained even when the load carried by the baseline vehicle was reduced to 50% of the rated load.

A lower rollover criterion (85%*RSF*) would be expected to cause frequent false warnings of a potential rollover. A compromise design is desired to utilize the benefits of the associated superior lead-time and reliability, and reduce the risk of false warnings. The results clearly show that the rollover criterion based upon unity value of *RSF* provide insignificant lead-time for the driver to induce a corrective maneuver. Moreover, the responses of both measures corresponding to *RSF=1*, show greater variations when design and operating parameters are varied. The lead-time performance of the prediction based upon a_{y2} and *NRSSM* generally improves as a more conservative rollover criterion is chosen. The selection of the criterion near 85%*RSF* yields higher lead-time and greater reliability in terms of coefficient of variation of the identified metrics, i.e *NRSSM* and a_{y2}.

6.2.4 Threshold values of a_{y2} and *NRSSM*

The development of an early warning rollover prevention mechanism necessitates identification of reliable values of thresholds of the selected indices, namely *NRSSM* and a_{y2}. A warning on the onset of a potential rollover would be generated when the instantaneous values of these metrics exceed their threshold values. The simulation results attained under wide variations in the design and operating parameters could be effectively used to identify the threshold values of the measures corresponding to *RSF*=0.85.

Minimum and mean values of both measures are thus evaluated corresponding to variations in each of the design and operating parameters considered in the study. The mean values of the minima and the means are further evaluated to identify threshold limits of *NRSSM* and a_{y2} for the two-stage warning strategy. The means of the minima may be implemented to initiate a first-stage visual warning, which would be followed by an auditory second-stage warning, based on the mean of the means for *NRSSM* and a_{y2}.

Table 6.14 summarizes the means of the minima and the means of the two measures, respectively, together with the values incorporating a further 5% safety factor to arrive at the primary and secondary threshold values respectively. The safety factor yields the lower limit of the thresholds of 2.0 degrees and 0.30g, respectively, for *NRSSM* and a_{y2}, for generation of the primary warning. The corresponding mean values for the second-stage warning are 2.15 degrees and 0.35g.

Table 6.14: Primary and secondary threshold limits

Rollover Measure	Primary Threshold Value	Primary Threshold Value (5% Safety factor)	Secondary Threshold Value	Secondary Threshold Value (5% Safety factor)
NRSSM	2.12 Degrees	2.0 Degrees	2.28 Degrees	2.15 Degrees
a_{y2}	0.312 g	0.30g	0.363 g	0.35g

The threshold values of *NRSSM* and a_{y2} based on 85%*RSF* and 5% safety factor could be incorporated into an open-loop rollover warning and control algorithm to provide reliable prediction. A two-stage design of a prediction and warning algorithm is conceived, where the first segment triggers the primary warning of lesser severity, when the identified metrics exceed their primary threshold values, 2.0 degrees and 0.30g for the *NRSSM* and a_{y2}, respectively. The second segment of the algorithm corresponds to the generation of a more severe warning when the measures exceed the secondary threshold limits of 2.15 degrees and 0.35g, respectively.

6.3 EARLY WARNING AND CONTROL ALGORITHM AND OVERVIEW OF THE WARNING SYSTEM

It has been reported that the driver of an articulated vehicle generally remains unaware of an impending rollover instability, which mostly initiates at the rearmost axle under a dynamic directional maneuver [69]. A potential rollover can be averted by informing the driver of the impending roll instability, such that the driver may induce a corrective action. This approach would require the detection of the onset of a roll instability through on-line monitoring of key states of the vehicle that directly relate to the onset of a rollover. Considering that articulated vehicle combinations employ considerable variations in their design and operating variables, it is vital that the identified states be either insensitive to such variations or account for such variations. Moreover, the detection must be timely, such that the driver may undertake the corrective action with sufficient lead-time. The simulation results attained under wide variations in design and operating conditions suggest that the continuous monitoring of semitrailer lateral acceleration and normalized roll response of the combination (*NRSSM*) could predict the onset of a rollover in a reliable and timely manner. The prediction through these measures would require real-time measurement and processing of semitrailer lateral acceleration and roll angle and the roll angles of the tractor front and rear axles.

A low pass filter may be integrated within the algorithm to eliminate sudden surges in its value attributed to zero roll angle response of the tractor rear axle. Alternatively, the value of *NRSSM* could be suppressed to zero, when the magnitude of the tractor rear axle roll angle lies within a narrow band around zero such as $|\phi_{u2}| \leq 0.20$ degrees. The prediction algorithm also integrates the threshold values of the measures, which are relatively insensitive to variations in design and operating conditions, and the rate of steering. The proposed algorithm involves continuous comparison of the measured values of *NRSSM* with its primary threshold value of 2.0 degrees, at the first stage. It is recognized that the *NRSSM* value may approach a higher value when the front axle encounters a sudden bump. The use of *NRSSM* alone may thus cause false predictions of a rollover. The preliminary prediction and warning thus utilizes simultaneous monitoring and consideration of the semitrailer lateral acceleration response, which is also compared with its primary threshold value of 0.30g. A primary visual warning is generated for the driver only when both the measures i.e a_{y2} and *NRSSM*, exceed their respective primary threshold values. The continuous monitoring of the *NRSSM* and a_{y2}, however, is continued, when their respective magnitudes are observed to be below or above their primary threshold values. The warning process is continued until the *NRSSM* and a_{y2} magnitudes approach acceptable values below their thresholds.

In the event when either the *NRSSM* or a_{y2} exceed their respective secondary threshold values (2.15 degrees or 0.35g), a severe audio warning is generated for the driver to undertake a corrective action in an urgent manner. In an open-loop control, this secondary warning serves as a more severe warning for the driver on the onset of a rollover. This warning signal may also be employed in a closed-loop rollover control mechanism with the aid of engine braking or vehicle brake system. In this case, the warning must be issued to the driver to ensure that the driver remains aware of the actions of the active controller. A sampling rate of 100Hz would be considered appropriate for monitoring of the two measures, a_{y2} and *NRSSM*, when the vehicle is in motion. A microprocessor may be employed for digitizing the measured signals, computing the *NRSSM* and generating the warning signals. The processor may also store the important events where one or more measures approach their threshold values. This information could be further applied to assess the design features of a particular road segment, the driver performance and vehicle behavior.

The warning in the initial stage is thus generated only when both measures exceed the respective primary threshold limits. This would reduce the potential for false warnings at the early stage. The second-stage warning, however, is generated when either of the two measures exceeds the final threshold limits, irrespective of the value of the other measure. This approach would permit the formulation of a fail-safe strategy, while it may also lead to a possible false warning in the final stage. Figure 6.1 presents the early warning algorithm involving the two-stage warning based upon on-line monitoring of the *NRSSM* and a_{y2}.

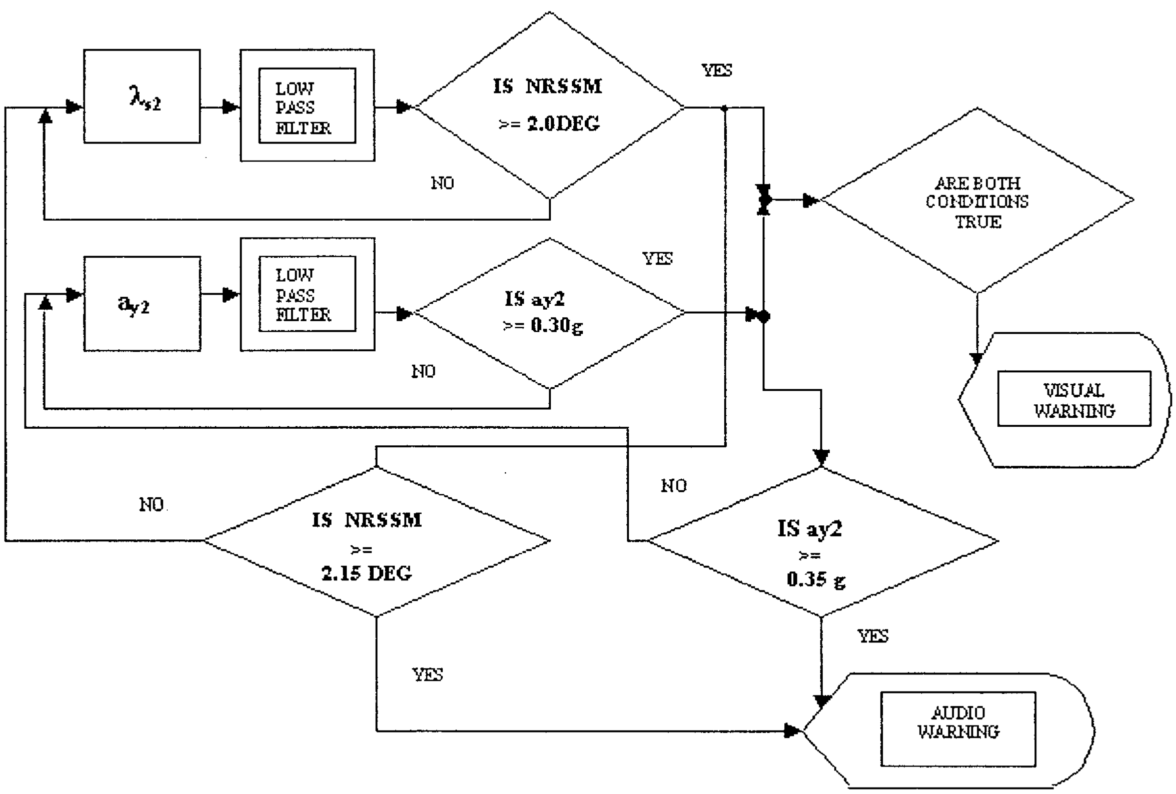


Figure 6.1: Flowchart depicting the early warning and control loop for prevention of roll instability based on *NRSSM* and a_{y2}.

6.3.1 Sensors and Signal Filters

The continuous monitoring of the responses of a vehicle in motion can be achieved using lateral acceleration and roll angle sensors. The lateral acceleration of the semi-trailer sprung mass can be easily measured using inexpensive and accurate micromachined accelerometers, which are considered superior for measurement of low frequency signals. Accelerometers with input range of ± 1g and capable of withstanding temperature variations and sudden shocks would be considered appropriate for the application.

The determination of *NRSSM* requires the measurement of the semi-trailer sprung mass roll angle and the roll angles of the tractor front and rear axles. Compact and rugged micromachined inertial sensors can be used to measure the roll angles with reasonably high accuracy [62]. Such gyroscopes or inertial sensors should be capable of withstanding high shocks experienced during loading and unloading. The variations in the roll angles of the sprung and unsprung masses due to steering inputs occur at low frequencies. The sensors, therefore, must be capable of measuring dc signals. Under a dynamic directional maneuver, such as path change and obstacle avoidance, the ratio of the roll angle response of the rear axle to that of the first axle may approach a very low value, which may cause a high value of *NRSSM* and thus a false warning. This effect can be eliminated by suppressing the *NRSSM* value when the magnitude of the tractor rear axle is in the vicinity of a threshold value around zero (e.g. ±0.20 degrees). Alternatively, a low pass filter may be employed to suppress the sudden surge in *NRSSM* as the tractor rear axle roll angle response approaches zero. A low pass filter for the lateral acceleration response would also be desirable to eliminate the high frequency variations induced by the tire-road interactions.

6.3.2 Human Factors Issues

The effectiveness of the early warning and control strategy depends upon the mode of delivery of the warning signal to the driver. The visual warnings can be displayed continuously with a “No Risk” message being displayed in green light as long as the vehicle operates within acceptable roll stability limits. When one or both of the threshold values are exceeded to initiate either visual or audio warnings, the mode of warning requires consideration of factors associated with guaranteed human perception and reaction. The visual warning generated at the primary threshold stage could be an “Alert” message along with the illumination of an arrow suggesting the direction of steering that could prevent the roll instability condition. The direction of the arrow would depend upon whether the value of *NRSSM* and/or lateral acceleration is positive or negative. The visual display unit could also generate a “Warning” with higher intensity of illumination and suggesting the direction of steering to prevent the roll instability when the secondary threshold is breached and the audio signal is issued. Figure 6.2 shows the possible messages that could be displayed using the visual display unit.

The secondary and more intense warning can be provided using the vehicles audio system and upon detection of the secondary thresholds violation through a continuous audio beep in small intervals of time or a buzz at sufficiently large amplitude can be delivered to the driver. The processor should overrun all the other audio signals or any audio entertainment unit, when a rollover threat is detected and issue the signal to generate the audio warning signal.

Figure 6.2: The possible displays in the proposed visual display unit.

The success in the prevention of a roll instability through warning and driver control depends upon the response time of the driver, which is the sum of the perception, and reaction time, and the neuromuscular response time. A greater lead-time also helps in accommodating a greater response time. The human factors issues associated with the visual and audio warning signal delivery are detailed in the following subsections:

6.3.3 Visual Display of Warning Signal

Visual signals are considered to be more comprehensive in terms of delivery of the information, when placed within the cone of vision. The effectiveness of the visual signal delivery depends upon the illumination, color, contrast and duration of the signal, and the distance at which the visual display unit is placed from the driver's eyes. Alphanumeric characters in bright colors need to be selected to display the warning signals using the visual display unit. The warning message should blink consistently with a time interval of 0.01 seconds. The "No Risk" signal should be displayed as soon as the rollover threat is removed. The visual signal should be conspicuous, legible and comprehensive in terms of delivered message. The positioning of the visual display should be such that the driver can easily see it while focusing on his primary visual field, i.e. the road ahead and the dashboard displays.

The visual rollover warning display in the form of a 15cm x 10 cm vacuum fluorescent display can be designed, which would display characters in alpha numeric form and the signals in either green, orange or red colors against a dark background. The display is also desired to have an electro-luminescent property so that it can be seen both in bright daylight as well as in the dark. It should be rugged enough so that it is sheltered against water, dust, and heat and can endure the rigors of commercial truck operation. The illumination of the characters of the display and the size of the characters are decided on the basis of the shortest time required by the driver to perceive the condition of the vehicle. The amplitude of the signal is decided based on the ability of the driver to perceive the warning signal, even if he is not focusing at the visual display. The positioning of the visual display should be within a cone of vision with cone angle equal to 40 degrees, so that by all means it lies in the driver's field of vision. This display screen can thus be placed either on top of the dashboard, away from the circumference of the steering wheel, within the drivers visual field, or inside a slot at the upper half of the dashboard, within a 40 degree cone of vision.

6.3.4 Audio Warning Signal

An audio signal becomes invaluable when the driver does not focus on the visual signal for some reason. An audio warning could be most effective when an urgent attention is required. Audio signals are particularly effective, when the signal is of short duration. An audio warning of a potential roll instability should be of a sufficiently large amplitude, delivered for a relatively short duration such as a 70dB beep or buzz. Furthermore, the processor must override the input from the onboard audio entertainment devices to transmit the warning signal using the speakers. Further efforts would be desirable to investigate ergonomic considerations, minimization of false alarms and adaptability of the driver to the system.

6.4 SUMMARY

The lateral acceleration of the semi-trailer sprung mass and the *NRSSM* are integrated to develop a two-stage rollover warning strategy. The values of these two metrics are examined under several stages of relative roll instability to establish ranges of their values under several design and operating condition variations. The simulation results reveal relatively lower variations in the values of *NRSSM* and a_{y2} corresponds to 85%*RSF*, when the variations in cg height, rate of steering, operating speed and load, and component properties are considered. The lead-time is also found to be greatest in almost all the cases at 85%*RSF*. An early warning and control algorithm is thus proposed based upon 85%*RSF* rollover condition. The algorithm based on the threshold values of the identified rollover metrics is a two-tier warning algorithm that would provide effective and comprehensive rollover warning and control function. The 85%*RSF* condition can be characterized in terms of the *NRSSM* reaching 2.0 degrees or the lateral acceleration reaching 0.30g at the first stage. A second tier of warning threshold values based on the average of the mean values of these two metrics under variations in design and operating conditions is determined at 2.15 degrees and 0.35g to diminish the possibility of failure of the delivery of the warning signal when one of the thresholds is not breached but the other metric's threshold is breached significantly. The system requirements for the proposed warning strategy are outlined and the human factors issues associated with the effective usage of the warning system is discussed.

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

7.1 HIGHLIGHTS OF THE INVESTIGATION

As described in Chapter 1, the overall objectives of this dissertation research is to identify effective and feasible measures of roll instability that would be incorporated in an early warning strategy to predictively warn the driver of an articulated vehicle of an impending rollover. The specific objectives involved a detailed understanding of the rollover mechanics of an articulated vehicle so as to identify the relevant rollover instability condition, development of a commensurate model for analysis of roll dynamic response of articulated vehicles, thorough assessment of known rollover metrics for their suitability for implementation in an early warning mechanism, extensive parametric sensitivity analyses to assess the variations in existing metrics under variations in the design and operating parameters, identification of a potential metric for reliable prediction of impending roll instability, relative assessment of the existing and potential rollover metrics to derive the most suitable metrics to be incorporated in an early warning strategy and finally developing a two-stage early warning algorithm for predicting rollover of articulated vehicles. Some results attained from this investigation have been published and are awaiting publication [81,82,83]. The major contributions of this dissertation research are below:

- The relative rollover condition is identified as the pertinent index of the onset of a roll instability for the articulated vehicles. It is signified by the lift-off of all wheels on a given track, except the tractor front axle, thereby providing allowance for the driver to undertake a corrective action. The Roll Safety Factor (*RSF*) attaining a unity value characterizes the relative roll instability condition, irrespective of the configuration, design or operating conditions.

- The reported rollover metrics, classified into measures of static and dynamic roll instability, are thoroughly reviewed to assess their potential for applications in early warning rollover prevention devices. Three different performance measures are formulated to assess the reported rollover metrics: measurability, sensitivity and reliability, and the lead-time.
- A comprehensive simulation matrix is formulated to study the reliability of different measures in predicting the onset of a rollover. The simulation matrix comprised a total of 90 and 279 combinations of design and operating condition parameters for relative assessment of sensitivity and thus reliability as well as lead-time of each metric under steady state and dynamic roll analyses respectively. The results are analyzed to identify most significant design and operating conditions affecting roll stability.
- An alternate measure (*NRSSM*), based upon roll responses of the semitrailer sprung mass, and the tractor front and rear axles, is proposed as a more reliable rollover metric that would also yield reasonably good lead time. The reliability of the proposed measure in predicting an impending roll instability is investigated together with its correlation with *RSF*.
- Threshold values of the proposed *NRSSM* measure and semitrailer acceleration are established from the ranges of the responses under wide variations in design and operating conditions.

- A two-stage detection and warning strategy is proposed incorporating both. *NRSSM* and a_{y2}, to enhance the prediction reliability.

7.2 CONCLUSIONS

The present research work has led to the development of a new and potentially more reliable measure for predicting an impending rollover. A comprehensive parametric sensitivity analysis of the influence of design and operating parameters on the roll stability of articulated vehicles is performed and the most feasible rollover metrics for open-loop rollover warning and control are identified based on relative assessments of the reported and proposed rollover metrics. The major conclusions drawn from the results of the study are summarized below: on the studies carried out in this thesis work, the following conclusions can be drawn:

- An open-loop rollover prevention strategy based upon early warning must detect the onset of potential rollover on the premise of the relative roll instability condition, which would not only provide superior lead-time but also reduce the effort to restore the vehicle to the stable domain.
- The dimensionless *RSF* measure attains a value of ± 1, as the vehicle approaches its relative rollover condition, irrespective of the vehicle configuration, which makes it superior in terms of reliability. However, its complex measurability necessitates the identification of alternate metrics that correlate with the *RSF*.
- The comprehensive parametric sensitivity analyses revealed that the trackwidth, the center of gravity height, suspension and the operating load

have significant effect on the roll stability of the vehicle under both static and dynamic conditions.

- The parametric sensitivity analyses of the existing rollover metrics also revealed that the existing measures could not be employed singularly to predict onset of roll instability in a reliable manner.
- Measures based on *RSF* and *LTR* could predict the onset of a rollover most reliably. These measures, however, are considered to be suited for an early warning monitor due to the complexities associated with their measurements.
- The measures based on lateral acceleration and sprung mass roll angles are quite sensitive to variations in design and operating conditions.
- The measures based upon the axle roll angles are relatively insensitive to variations in component design properties but quite sensitive to variations in operating load.
- An alternate measurable metric based on rearward amplification tendencies of the lead unit and the combination, termed as normalized roll-response of the semitrailer sprung mass (*NRSSM*), correlates very well with *RSF* and yields superior lead-time for undertaking a corrective action.
- The *NRSSM* varies within a narrow band under wide variations in vehicle design and operating parameters, suggesting its superior reliability.
- The *NRSSM* yields reasonably good lead-time performance, which can be further enhanced by setting a conservative relative rollover condition (*RSF*=0.85).

- Formulating a two-stage warning based upon *NRSSM* and a_{y2} could reduce the potential of false warnings, associated with the chosen conservative limit.
- The threshold limits of 0.30g and 2.0 degrees for a_{y2} and *NRSSM*, respectively, could effectively serve the basis for the primary warning.
- A more severe second-stage warning must be generated when either *NRSSM* exceeds its threshold value of 2.15 degrees or a_{y2} exceeds 0.35g.
- A warning strategy based upon *NRSSM* together with the semitrailer lateral acceleration response is proposed to enhance the reliability and lead-time, while reducing the possibilities of false warnings. An algorithm for the two-stage audio/visual warning is finally proposed.

7.3 RECOMMENDATIONS FOR FUTURE WORK

The thesis research is carried out to develop a concept in effective rollover warning strategy based on a reliable and measurable metric that would predictively warn the driver of an articulated vehicle of an impending rollover in a timely manner. Based on this study, it is recommended to undertake the following future works to further develop an effective and fail-safe warning and control system for prevention of rollover induced accidents involving articulated vehicles:

- In order to compliment and reinforce the simulation model, real world tractor-trailer rollover testing should be pursued.

- Measuring the axle roll angle of tractor rear axles is precarious and further investigation into the online monitoring of the unsprung mass roll angles is required.
- Further studies, however, would be needed to assess the applicability of micro gyroscopes for on-line monitoring of *NRSSM* and the effect of road banking on the selected measures.
- The response of *NRSSM* to various other steering maneuvers other than those applied in this investigation needs to be analyzed.
- The improvement of roll stability of articulated vehicles through changes in the vehicles design parameters should be explored further so as to enhance the roll stability of such vehicles.
- Rollover prevention through active control systems based upon the identified metrics as inputs for the control system needs to be investigated to eliminate the driver from the control loop thereby addressing the issue of failure of the warning system or perception disability of the driver.
- The effectiveness of a two-stage early warning system with particular emphasis on the driver-warning system interface and related human factors issues needs to be further analyzed.

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