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Date : 4/5/07

MODELLING AND SIMULATION OF A HEAVY VEHICLE

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A THESIS SUBMITTED AS PARTIAL REQUIREMENTS FOR THE DEGREE OF THE BACHELOR OF MECHANICAL ENGINEERING (AUTOMOTIVE)

FACULTY OF MECHANICAL ENGINEERING
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2007

"I hereby declare that the thesis is based on my original work except for quotations and citations which have been duly acknowledged"

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ABSTRACT

This paper discusses the benefits of heavy tractor trailer truck simulation and the advantages of an active suspension. Some time was taken to learn the simulation software (Arcsim) as well as some basic vehicle dynamics. The model was used in two simulations, a lane change and WC steer cornering analysis at 38 km/h and 90 km/h. Both the simulations showed the disadvantages of the passive suspension system. An increase in roll angle was observed in the cornering analysis and the lane change when a passive suspension was used. The vehicle speeds, as well as the results of changes to inner, outer, or both springs were plotted to help establish relevant variables in future models.

ABSTRAK

Kertas kerja ini membincangkan kebaikan simulasi kenderaan berat dan bagaimana keputusan pasif suspensi beraksi. Selain itu, daripada kajian ini dapat juga mendalami serba sedikit tentang perisian simulasi dan sekaligus mengetahui tentang dinamik kenderaan. Model yang dibentuk digunakan dalam dua simulasi, iaitu penukaran jalan dan belokan *WC steer* pada 38 km/h dan 90 km/h halaju kenderaan. Kedua-dua simulasi ini menunjukkan kelemahan sistem suspensi pasif. Daripada kajian yang dibuat, sudut gulingan akan meningkat apabila system suspensi pasif digunakan. Halaju jkenderaan dan juga keputusan perubahan dalaman dan luaran spring telah diplot untuk membantu mencari pembolehubah yang releven untuk model yang akan datang.

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LIST OF SYMBOL

SYMBOL	DEFINITION
F_b	Lateral shear force in vehicle frame
F_y	Lateral tire force
F_z	Vertical tire force
h	Height centre of sprung mass, measured upwards from roll centre
h_b	Height of frame twist axis, measured upwards from ground
h_{cm}	Height of total centre of mass, measured upwards from ground
h_s	Height of centre of sprung mass, measured upwards from ground
h_u	Height of centre of unsprung mass, measured upwards from ground
k	Suspension roll stiffness
k_b	Vehicle frame torsional stiffness
k_t	Tire roll stiffness
l	Suspension roll damping rate
l_b	Vehicle frame torsional damping rate
m	Total mass
m_s	Sprung mass
m_u	Unsprung mass
U	Forward speed
u	Active roll torque
r	Front
f	Rear

GREEK LETTER	DEFINITION
β	Sideslip angle
δ	Steer angle
ϕ	Absolute roll angle of sprung mass
ϕ_t	Absolute roll angle of unsprung mass
$\dot{\psi}$	Yaw rate

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CHAPTER ONE:
INTRODUCTION

CHAPTER ONE

INTRODUCTION

1.1 Overview

Roll-over of heavy vehicles is a serious worldwide safety problem. Over 15,000 rollover accidents per year involving commercial heavy vehicles occur in United States, including 9,400 involving tractor semi-trailer combinations (Blower D. 1997). The average cost of each of these accidents to vehicle operators has been estimated at between £75,000 and £100,000 (Harris R. 1995). These costs include recovery and repair of the vehicle, product loss, and road repair and re-surfacing. There are also costs attributable to expenditure on hospitals and emergency services, and to social security benefits paid as a result of these accidents. These high costs are the primary motivation for research into active roll control of heavy vehicles.

Roll-over accidents mostly involve articulated vehicles and occur on highways (Kusters L.J.J. 1995). Three major causes of rollover have been identified:

a) Sudden course deviation (often combined with heavy braking and high initial speed)
b) Excessive speed on curves
c) Shifting load

It is usually difficult for the driver to sense the roll-over behavior of a tractor semi-trailer combination as his perception is mainly based on the response of the tractor, and the trailer is generally the first unit to rollover.

The problems with heavy vehicle roll-overs have been well documented, for example, roll-overs occur when the overturning moment on the vehicle, due to lateral acceleration and displacement of center of gravity (CoG), exceed the restoring forces that the tires can supply. A static balance is no longer possible and the CoG must accelerate outwards. Statistics indicate that the less stable the vehicle, the more likely it will be involved in a roll-over accident. This indicates that it is difficult for a driver to anticipate the roll-over limit of the vehicle. Studies have shown that avoidance of roll-over can only be achieved in most cases by improving the stability of the vehicle.

Braking can provide an effective means of reducing roll due in part to the coupling between roll, lateral, and yaw dynamics. This coupling allows the use of yaw moment generation through differential braking to influence roll. A second effect is provided by the nonlinear nature of pneumatic tires. Since roll is directly caused by the lateral acceleration of the center of mass, a reduction in lateral acceleration can also reduce the roll angle. When tires are made to generate a longitudinal force through braking, their reserves of grip for lateral force generation are correspondingly reduced. Thus, actively brakes the axles can reduce lateral acceleration and in turn reduce the roll angle. A third beneficial effect of braking is that vehicle speed is reduced. Rollover often occurs when the vehicle is traveling at a rate too high for a given curve. Braking slows the vehicle increasing the chance that it will negotiate the curve without rolling over.

It was shown that only a minority of rollover accidents could have been avoided with a warning device, potentially more with a skilled driver, but half of the rollover accidents they investigated were not preventable by driver action alone. This highlights the need for an active safety system for heavy vehicles.

1.2 Problem Statement

1.2.1 To know the stability of standard heavy vehicle while accelerates, lane change and WC cornering

1.2.2 To find the effectiveness of longitudinal, lateral and vertical model

1.3 Objectives of The Project

1.3.1 To develop vehicle dynamics model of a heavy vehicle

1.3.2 To validate the model with the existing multibody vehicle model namely *ARCSIM*

1.4 Scopes of The Project

1.4.1 Development of vertical dynamics model

1.4.2 Development of logitudinal dynamics model

1.4.3 Development of lateral dynamics model

1.4.4 Combination of three model

1.4.5 Model validation

CHAPTER TWO: *LITERATURE REVIEW*

CHAPTER TWO

LITERATURE REVIEW

2.1 Overview

Simulations are an important part of heavy vehicle development. For most companies it is important to evaluate what a vehicle will do before the prototype is built. Using simulations can save millions in the overall cost of vehicle development. If a company can test a heavy vehicle design with simulation software, the process of building a prototype can be placed much later in the design process where it is made after expensive design changes are completed. In 1998 the importance of simulation was illustrated with an article from PR Newswire. Gary Diaz, the chief technical officer of Navistar International explains the importance of using simulations, "We must reduce development cost while improving product quality in order to exceed the expectations of our customers and achieve our corporate vision of being the best truck engine company in North America." Diaz goes on to say that our engineers and designs need the ability to evaluate design alternatives early in the development process, so we get it right the first time. With the use of software like TruckSim/Arcsim, the Center for Advanced Vehicle Design and Simulation (CAVDS) is developing a full process for vehicle development.

2.2 Type of Heavy Vehicle

2.2.1 Truck and Delivery Van

Figure 2.1 Type of truck and delivery van

This type of truck is used for delivery any item in light weight. So to full fill this function, this truck built follows the ability to mobility, vehicle controller, comfortable and safety. This truck is designed with front mounted engine; front or rear wheel drive, dependent suspension and over 3.5tonne at dual tire at rear axle.

2.2.2 Light and Heavy truck (vehicle)

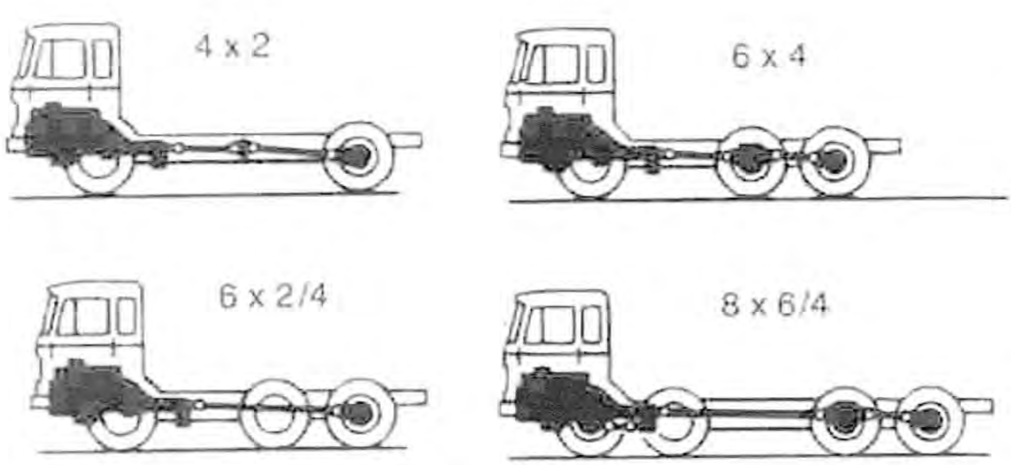

Figure 2.2 Type of light and heavy truck (vehicle)

2.2.3 *'Road Train'*

Figure 2.3 Example for road train vehicle

This type of truck is almost the same with the heavy vehicle but this truck have extra trailer at rear of truck to put an additional item.

2.3 The Basic Factors Affecting Vehicle Behavior

2.3.1 Yaw Plane

Figure 2.4 shows a free-body diagram of a four-wheeled tractor as viewed from the top (the yaw plane). There are just three equations that govern its basic behavior in the yaw plane, the sum of the tire shear forces and the horizontal hitch force must equal the vehicle mass times its acceleration in both the vehicle X and Y directions, and the moment of those forces about the vehicle mass center must be equal to the product of the yaw acceleration and the vehicle yaw moment of inertia. Aligning moments at each wheel (M_{zi}) also contribute to the overall yaw moment. Thus, a main objective of ArcSim models is to accurately predict tire shear forces, tire aligning moments, and hitch forces. Yaw behavior is also influenced by the rotary motion of the vehicle bodies in roll and pitch. Mechanical energy is transferred as the vehicle rolls and pitches, and these motions contribute to the vehicle transient response.

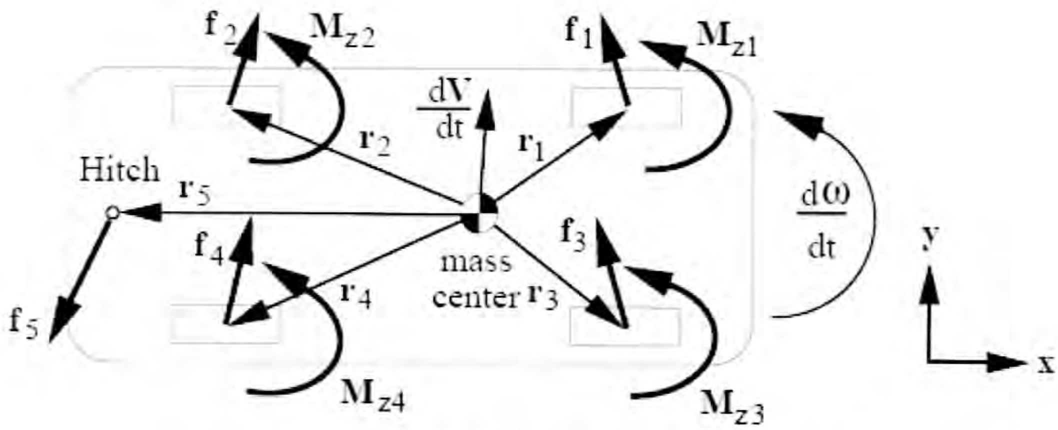

Figure 2.4 Free body diagrams in the yaw plane

2.3.2 Roll Plane

Figure 2.5 shows a free-body diagram in the roll plane for a truck body and one axle as viewed from the rear when making a left-hand turn. (A left turn implies a positive rotation of the vehicle about the Z axis.) The only external forces acting on the vehicle system are due to the tires and any hitches connecting it to other vehicle units. The vehicle is also subjected to roll moments ($\mathbf{M}_h$) from the hitch (es). The vertical components of the tire forces, which do not appear explicitly in the yaw-plane equations, are basic factors in the roll-plane equations. The overall roll stability is characterized by the first equation in Figure 2.5, involving the balance of moments about the vehicle mass center. When tires lift-off of the ground, the forces are zero on one side of the axle. Lift off usually occurs first for tires on the axle furthest to the rear of all vehicle units in the system, on the side that is on the inside of the turn (the left side in Figure 2.5). When liftoff has occurred for more than a critical number of axles, roll stability cannot be maintained. This condition is considered to be the onset of rollover. Thus, the basic factors for predicting roll stability are the vertical tire forces, and their points of application relative to the vehicle mass center, which change as the vehicle rolls.

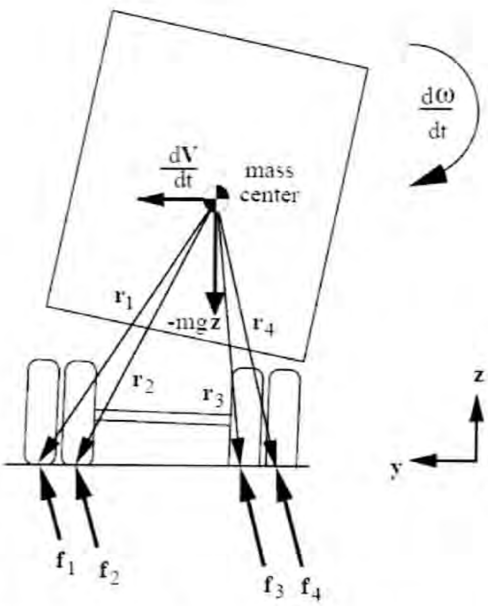

Figure 2.5 Free body diagrams in the roll plane

For simplicity in Figure 2.5, the separate motion of the axle is not shown, and the moment balance in the X direction is written as if all of the roll inertia were lumped into the sprung mass. The independent motions of the axles relative to the main bodies are a secondary factor that must be included in the model to obtain realistic transient effects.

2.4 Rigid Body Kinematics

The tractor-semitrailer model consists of rigid bodies for the sprung mass of the lead unit, the sprung mass of the trailing unit (if there is one), and the axles. The kinematics of these rigid bodies is described below.

2.4.1 Sprung Masses

The sprung mass of the lead unit, S_1, (with body fixed unit vectors $\mathbf{s}_{1x}$, $\mathbf{s}_{1y}$ and $\mathbf{s}_{1z}$) is given six kinematical DOF, such that it can attain any position and/or any orientation in space.

The sprung mass of the trailing unit, S_2 (with body fixed unit vectors $\mathbf{s}_{2x}$, $\mathbf{s}_{2y}$ and $\mathbf{s}_{2z}$), is a child of S_1. The hitch connection between S_1 and S_2 is modeled kinematically as a ball joint. It is given three rotational DOF relative to S_1, but no translational DOF. The motion of S_2 about the hitch is further retarded by a roll torsional spring, described later.

2.4.2 Axles

The movement of an axle relative to the sprung mass to which it is attached can be described by a sequence of two motions:

a) A rotation in roll

b) A vertical translation (Winkler, C.B., *et al*)

Two rigid bodies, A_r and A, are used to model the axle, each with one DOF. Body A_r is an intermediate frame that is given zero mass and rotational inertia. Body A is given the mass and rotational inertia properties of the axle. All external forces and moments that act on the axle (i.e., suspension and tire forces and moments) are applied to body A.

The bodies Ar and A, respectively, have body fixed unit vectors: $\mathbf{ar}_x$, $\mathbf{ar}_y$, $\mathbf{ar}_z$ and $\mathbf{a}_x$, $\mathbf{a}_y$, $\mathbf{a}_z$. Figure 2.6 shows the directions of the unit vectors in the nominal configuration for a positive roll-steer coefficient. Body A_r is a child of S that rotates in roll about an axis aligned with $\mathbf{ar}_x$.

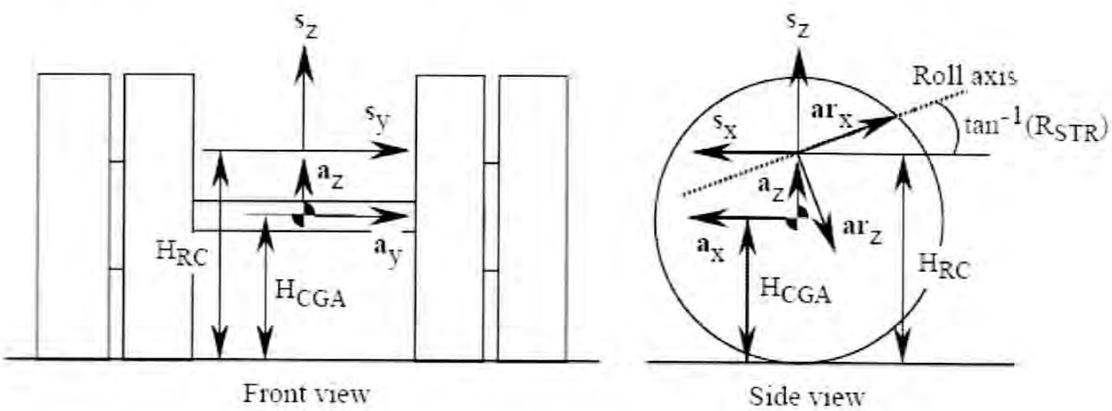

Figure 2.6 Axle axes and directions

The roll axis ($\mathbf{ar}_x$) is inclined from $\mathbf{S}_x$ by the arc-tangent of R_{STR}, the roll-steer coefficient. The axis direction is set such that a positive roll of the sprung mass (leaning to the right, relative to the axle) also corresponds to a positive rotation of the axle about $\mathbf{ar}_x$, relative to the sprung mass. Positive roll angle causes the axle to steer to the left for a positive value of R_{STR}. This sign convention for roll-steer is consistent with the ISO axis system where $\mathbf{s}_z$ is up and steer to the left is positive. However, for the SAE coordinate system, in which the $\mathbf{s}_z$ axis is pointed down, the sign of the roll-steer coefficient, would be reversed because the left-hand steer would be negative. Body A translates vertically in the direction $\mathbf{a}_z$ and it is a child of A_r.

Nominally, body A is aligned with body S. (Thus, A is tilted with respect to A_r) The origin of the coordinate system of body a body is located at the axle mass center, which is assumed to be in line with the spin axis of the wheels on the axle. The axle roll and mass centers are assumed to lie at the center of the axle laterally and longitudinally. The heights of the roll and mass centers are specified by the user with the parameters H_{RC} and H_{CGA}.

2.5 Mass and Rotational Inertia

The lateral locations of all mass centers (sprung masses and axles) are given Y coordinates of zero, placing them in the center pitch plane of the vehicle. The locations of the axle mass centers are also at the centers of the axles. The heights of the mass centers are provided by the user for the axles individually, and for the entire vehicle units. The heights of the mass centers for the sprung masses are calculated by ArcSim from other user input parameters.

The masses of the sprung masses are described with equations involving the axle loads. Equations are also used to locate the mass centers longitudinally. This is done to eliminate the need for the user to calculate mass properties that are based on axle loads axle loads are instead used directly as inputs. It is usually easier to measure the inertial properties for an entire unladen unit than for the laden (or unladen) sprung mass alone. ArcSim calculates the mass and rotational inertia properties of the sprung mass from measurements made for the entire unladen unit, the difference between the laden and unladen axle loads, and parameters that the user must set to estimate the size of the load. The load is assumed to be rectangular and of uniform density.

This approach is taken to permit the user to enter data for the complete unladen units. These numbers never change for a particular vehicle, regardless of how it is loaded. Thus, to change the load, the user need only modify the load parameters. The roll, pitch and yaw moments of inertia are about the mass center with respect to the body X, Y, and Z axes. The products of inertia are not zero, but for most loading conditions, they contribute little to the system dynamics. For unusual loading conditions, however, (e.g., when the load mass center is very high, or very far forward or rearward), the X-Z product of inertia can affect the transient response (yaw rate in particular). A parameter for the X-Z product of inertia is included to account for such situations. For the axles, the moments of inertia about the mass center in yaw and roll are assumed to be equal and must be provided by the user. The moment of inertia in pitch and the products of inertia are relatively small. They are neglected by setting them to zero in the ArcSim models.

2.6 Equilibrium

The heights of reference points such as wheel centers and the top side of the load bed are defined for the case of the vehicle resting in equilibrium on a flat level surface. In a real vehicle, these heights depend on the load: the heights decrease slightly as the vehicle settles down when the payload is increased. In general, the sensitivity of these heights to load has a negligible effect on the vehicle behavior. The common practice is to use the same height values for all load conditions. However, if you are concerned with this effect, then the height parameters can be adjusted for the various loads.