HEAVY TRUCK DYNAMICS MODELING USING MULTI-BODY DYNAMICS

A Thesis

Presented in Partial Fulfillment of the Requirements for
the degree Master of Science in the
Graduate School of The Ohio State University

by
Chih-Liang Feng, B.S.M.E.

* * * * *

The Ohio State University
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Master's Examination Committee: Approved by:
Dr. Dennis A. Guenther
Dr. Mardi Hastings

Dennis A. Guenther
Advisor
Department of Mechanical Engineering
To every member of my family
ACKNOWLEDGMENT

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VITA

December 28, 1969 ............................................ Born - Taipei, Taiwan

June, 1992 .......................................................... B.S. Mechanical Engineering
                                            National Taiwan University
                                            Taipei, Taiwan

July, 1992 - May, 1994 ........................................ Republic of China Army
                                           Taichung, Taiwan

September, 1994 - present ................................. Research Associate
                                      Department of Mechanical Engineering
                                      The Ohio State University
                                      Columbus, Ohio

FIELD OF STUDY

Major Field: Mechanical Engineering

    Studies in Vehicle Dynamics - Dr. Dennis A. Guenther
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CHAPTER I
INTRODUCTION

1.1 General

Heavy truck behavior is more complex than passenger cars. Trucks include more bodies and more complex kinematics and dynamics than passenger cars. For example, the springs are stiffer, which may induce numerical problems. With the increasing reports of truck accidents, one can conclude that truck safety in the United States should be studied.
Thus, the study of truck dynamic response is becoming more important in order to increase the safety on highways.

In this research, the heavy-duty truck model is developed using DADS (Dynamic Analysis and Design System) [1], and it is to be transferred to RTRD (Real Time Recursive Dynamic) formulation. RTRD was developed at the University of Iowa and is the core program for NADS (National Advanced Driving Simulator) which is being developed for the U.S. Department of Transportation.

Truck geometry data was obtained from UMTRI (University of Michigan Transportation Research Institute). Inertia and suspension parameters are referenced to the sample data measured previously at UMTRI. Tire parameters are referenced to the sample data from the previous work of a joint NHTSA (National Highway Traffic Safety Administration) /SAE (Society of Automotive Engineers) heavy truck tire testing program.

1.2 Objectives

The objectives of this research are focused on the following points:

1) Introduce the fundamental multi-body dynamics and DADS analysis program theory.
2) Model the GM-VOLVO tractor for handling scenarios using DADS. Future study should include tractor-semitrailer behavior. This research is focused on developing a model of the multi-body dynamics of the tractor, including the detailed geometry of the tractor, and relative motion of each component to the other. The basis is the roll axis
concept. Suspension forces of the leaf springs and the shock absorbers characteristics are included. Tire forces are modeled using a simple linear model. To study the handling scenarios of the tractor, a kinematic steering system is constructed.

1.3 Thesis Overview

The thesis is divided into five chapters. Chapter 1 provides a general information of the applications of computer simulation in this research. The objectives of this thesis are also stated, giving the directions toward modeling the tractor dynamics.

Chapter 2 is the literature review. In this chapter the review of several past studies using multi-body computer simulation programs to study vehicle dynamics is made. Other papers focused on the study of heavy truck dynamics using multi-body dynamic simulation programs or mathematical simulations. The descriptions and the methodology of each model and the conclusions of each study are compared.

Chapter 3 introduces the fundamental multi-body dynamics and DADS analysis program. The flow of modeling in DADS is illustrated. Different analysis types of DADS analytical programs are described, and fundamental kinematics and dynamics concepts in DADS are introduced: Jacobian matrix, Lagrange multipliers, Newton equations of motion, and computational procedures in DADS. At the end of the chapter, the numerical methods in DADS are illustrated.

In Chapter 4, a formal multi-body model of the tractor is developed. The experimental data including the geometry, inertia properties, suspension characteristics,
and tire parameters are presented. The dynamic behavior of each component of the tractor is analyzed, and construction of the model with different kinematic joints and forces is shown. The hysteresis of the leaf spring and the properties of the shock absorbers are modeled. The tire forces are modeled using a simple linear model, and a kinematic steering system for handling analysis is presented.

In Chapter 5, the simulation results are presented and analyzed. The conclusions of this research and recommendations for future work are made in Chapter 6.
CHAPTER II
LITERATURE REVIEW

Computer simulation is a powerful tool to aid one in the understanding of the detailed behavior of dynamic systems and also can be used to predict system behavior before a design is implemented. With many conditions, it is necessary to choose between several design alternatives. With computer simulation, one can perform parametric trade-off studies to make the design decisions. It could further help in understanding design errors and problems. Most important of all, computer simulation could predict system
behavior, especially when testing is either too expensive or dangerous. There are many computer programs available today to study vehicle handling dynamics. VDANL (Vehicle Dynamics Analysis Non-Linear) and IDSFC (Improved Digital Simulation, Fully Comprehensive) [2] are two of the so called lumped parameter model simulations that are dedicated to vehicle handling dynamics.

With the advance of computer hardware, many dynamic simulations can be performed systematically using multi-body dynamics. Among them are ADAMS (Automatic Dynamic Analysis of Mechanical Systems), developed at Mechanical Dynamics Inc., and DADS (Dynamic Analysis and Design System) [1], developed at CADSI (Computer Aided Design Software Inc.). Multi-body dynamics computer simulation programs have their advantages as well as disadvantages. The main advantage of the programs is that models can easily be changed to modify the design. Such changes typically involve the additional degrees of freedom in the model or the additional sets of constraints equations. For traditional lumped parameter model simulation programs, these changes would be very difficult and time-consuming. On the other hand, because typical multi-body dynamics models include many small masses and stiff springs, which add additional degrees of freedom and enlarge the frequency domain, the computation time and quantity of input parameters required increase.

Numerical multi-body programs eliminate the need for deriving equations for each model manually by using a generic multi-body formulation coded in conventional engineering analysis language such as FORTRAN [2]. Another method for eliminating
the manual derivation of equations uses computer algebra programs such as MACSYMA, REDUCE, MAPLE, and MATHEMATICA [2] which have already been developed for mainframes and work stations. Some dynamicists have developed symbolic methods for automatically deriving equations of motion. AUTOSIM [2] is representative of a new simulation technology called symbolic multi-body programs and is used to model many automobiles and trucks. A symbolic multi-body program such as AUTOSIM offers a way to develop simulation models which are customized for an intended purpose. A clear result is the TRUCKSIM [2] software, which is very easy to use and could make detailed models for any engineer with a personal computer.

TRUCKSIM computer software is currently being developed at UMTRI (University of Michigan Transportation Research Institute). It is a PC-based, truck-simulation software package consisting of graphic user interface (GUI), and a collection of heavy truck simulation programs. The interface utilizes the built-in capabilities of off-the-shelf programming graphical database packages such as Toolbook and Supercard. All of the simulation programs in TRUCKSIM were generated at UMTRI using AUTOSIM.

Traditional vehicle dynamic simulations are based on a set of differential equations that describe the behavior of a class of vehicles [1]. On the other hand, computer programs for multi-body dynamics supply the program with detailed description of each mass involved and the joint connections. Multi-body computer programs first appeared in the early 1970s and now are being increasingly used by vehicle dynamicists.
ADAMS is a very powerful multi-body dynamics programs. There are many vehicle dynamics simulations done by using ADAMS. On the other hand, there are not as many simulations done by using DADS. Because of the similarity of these two multi-body dynamics programs, several studies using ADAMS are reviewed:

N. Orlandea [3] described a computer simulation of the front suspension of a 1973 Chevrolet Malibu using ADAMS. His research was focused on evaluating the speed, economy and accuracy of various computer simulations in predicting displacements and loads in a suspension system. N. S. Rai [4] simulated the suspension abuse events, e.g., curb, chuck hole, railroad ties and rough road impacts involving nonlinear, large displacements of suspension components with ADAMS and user written force subroutines. Rai concluded that the peak loads calculated from the simulation models agreed very well with values obtained from the physical testing.

Michael McGuire [5] used ADAMS to examine the effects of longitudinal suspension compliance on both the ride and handling performance of a vehicle in a 1/4 car model. He concluded that the longitudinal compliance caused the increased lateral stiffness of the model.

Herbert Loos [6] used ADAMS to model the handling behavior of a Ford Bronco II. Loos developed a simulation model in each design detail (e.g. each individual suspension bush). A total vehicle model originally had more than 100 degrees of freedom. But in order to reduce computation time for special applications, the degrees of freedom of the model was reduced to 46. He also included standard handling maneuvers
for simulation and compared the results with two handling test: Step input and sinusoidal
input steering. The tire model utilized spline fit data to get the lateral forces.

R. J. Antoun [7] used ADAMS to simulate the handling behavior of a Ford
Ranger. His model was constructed with detailed kinematic representations of the front
suspension, steering and rear suspension systems. The model was contained in an
ADAMS data set and user-written FORTRAN subroutines. The user-written subroutines
were required to model non-standard systems or components such as tire forces. A SAE
Three Links approximation was used to model the leaf spring of the rear Hotchkiss
suspension. The shock absorbers used 17 force-velocity data points and were modeled as
a function of velocity using spline fit method. He also compared the ADAMS simulation
results and experimental data of a J-turn maneuver with a steady-state lateral acceleration
of approximately 0.3g’s. He concluded that the reason why the ADAMS lateral
acceleration response time was shorter was because the ADAMS tire model did not
include the tire’s relaxation length.

With desktop and personal computers increasing in computational power, serious
consideration is being given to better utilize these computers for the simulations of
vehicle dynamics. Multi-body computer programs like DADS and ADAMS are
developed and implemented at workstations because typically they require more input
data and more computation time than traditional vehicle dynamic simulations. Garrick Hu
[8] used a desktop computer to simulate heavy truck ride behavior. The tractor-semi-trailer
model included system excitation from road roughness, nonlinear leaf spring
characteristics, and dynamic spring rate characteristics of air springs. The model was a 9 degrees of freedom pitch plane model. Each axle of the vehicle had a vertical translation degree of freedom. Each sprung mass had both a vertical translation and a pitch rotation degree of freedom. He also included a first order analysis model of frame flexure. Hu was one of the few researchers who took the bending mode of the truck frame into account. But in his research, elastic deformation of the flexible member was assumed to have no significant effect on the gross motion of the structure. The gross motion was then first determined using rigid body analysis, and the resulting reactions were used to determine the elastic member motion.

J. Y. Wong [9] compared the various computer simulation models for predicting the directional responses of articulated vehicles. The models being compared were the linear yaw plane model, the TBS model, the Yaw/Roll model, and the Phase 4 model [10]. These models varied greatly in complexity, in capability, and in the total number of degrees of freedom included. For example, the Phase 4 model included up to 71 degrees of freedom and required up to about 2300 lines of input data. On the other hand, the linear yaw plane model only included the lateral and yaw motions of the tractor and articulation in the horizontal plane of the other sprung masses and only required up to 35 lines of input data. Wong compared these models in predicting steady-state steering response and in a lane-change maneuver. Wong concluded that the linear yaw plane model did not take into account the effects of load transfer and used a linear tire model, so it does not have the capabilities to predict changes in handling behavior with lateral
acceleration. On the other hand, the other three models took into account the effect of load transfer and the nonlinear behavior of tires to varying degrees, so they could predict changes in handling behavior with lateral acceleration.

H. C. Pflug [10] studied the lateral dynamics behavior of truck-trailer combinations due to the influence of the load. The frames and bodies of the truck and the trailer which were nonrigid under torsional strain and rigid under flexural strain were simulated by two rigid bodies linked by a torsion spring aligned with the longitudinal axis of the vehicle. A basic assumption was made that the distribution of mass was symmetrical with respect to the longitudinal axis of the vehicle. He found out that loaded trailers oscillated with higher amplitudes than empty trailers did, and on empty trailers the oscillating motions decayed faster.

R. D. Ervin [11] studied the yaw stability of tractor-semitrailers in steering-only maneuvers. He developed a computer simulation model which included the frame torsional stiffness. From the results, a conclusion was made that while tractor yaw instability could occur well below the rollover threshold of certain vehicles, modified stiffness parameters could eliminate such premature yaw instability.

D. Cebon [12] has done considerable simulation studies of heavy truck dynamics, especially the studies on heavy trucks vibrations. He noticed that very few heavy vehicle simulations had included experimental validation. He developed a vibration simulation program suitable for modeling lumped parameter non-linear vehicle models of up to three dimensions. Two and three-dimensional non-linear models of a three-axle rigid fuel
tanker were validated with data from experimental tests. These models were: a 6 degree of freedom (DOF), two-dimensional trailer suspension model; an 11 DOF, two-dimensional tractor and trailer model; and an 21 DOF, three-dimensional whole vehicle model. One of the important features in these models was the description of the four-spring tandem axles suspension including the radius arms effect. He made the following assumptions of the suspension model: The leaf springs were treated kinematically as rigid beams. Pitch moments applied by the axle tubes and by rotational inertia of the axle assemblies were zero. Longitudinal acceleration of the unsprung masses was zero. Friction forces at the slipper ends are horizontal and normal forces were vertical with small displacements about the equilibrium position. After the validation, he concluded that the main sources of error in the simulation were thought to be the trailer suspension model properties and the assumption of rigid sprung masses. But the measured trailer bending resonance did not significantly affect the measured tire forces.

More researchers have focused on the study of heavy truck behaviors during the past two years:

Chris H. Verheul [13] used another multi-body dynamics computer program BAMS (Bondgraph-Based Algorithm for Modeling Multi-Body System) to simulate the heavy truck dynamics. He used two models to simulate a racing truck, one was simple and the other was complex. He compared the simple model with the complex one. He ran the simulation through several maneuvers: Speed sweep while running straight ahead, spiral curve at fixed speed, and speed sweep at constant path curvature. The numeric
value of the first six vehicle modes were also listed, i.e., rear axle roll, front axle roll, pitch about center of gravity, pith about rear axle, chassis roll, and yaw motion. From the results, he concluded that at large accelerations the handling behavior of the simple and the complex model showed a difference. Both models showed initial understeer changing to oversteer at higher lateral accelerations, and changing the frame stiffness would only influence the handling at larger accelerations.

Matti Huhtala [14] used ADAMS to simulate the road-vehicle dynamic interaction forces of three and four-axle trucks. The original four-axle truck model was created in cooperation with the manufacturer to be used in studying ride comfort. The model consisted of 35 inertia masses connected to each other with nonlinear force elements and kinematic joints. Altogether the model formed a set of 1166 differential algebraic equations. He used four links to model the multi-leaf springs and the shackle. Significant hysteresis of the multi-leaf springs was taken into account through the definition of the equivalent torsional springs. The frame of the truck was assumed to be rigid. Another simplification was that the engine and the gear box were connected rigidly to the frame. However, this assumption seemed to be inadequate because in reality they were connected with rubber mountings to each other. The steering system was not included in the model because the handling behavior was not examined in the research. The nonlinear, asymmetrical dampers were modeled with a spline function.
CHAPTER III

INTRODUCTION TO MULTI-BODY DYNAMICS AND DADS ANALYSIS PROGRAM

3.1 Multi-body Dynamics

A multi-body mechanical system is defined as a collection of interconnected rigid bodies that can move relative to one another, connected with joints that limit relative motion of pairs of bodies [15]. Multi-body dynamics is the study of the dynamic behavior of such systems undergoing large displacements caused by the effect of external forces that act on the systems. If we define the time history of the position or relative
position of the bodies, we can prescribe the motion of mechanical systems. Algebraic
kinematic relations, differential equations of motion, and external forces then determine
the behavior of the system.

When dealing with multi-body dynamics, vector and matrix algebra play an
important role because basic mathematical foundation for kinematics and dynamics is
formed by them. Matrix notation makes it easy to represent a system of equations, and
matrix operation makes it simple to develop and organize the solutions of systems of
equations. Vectors represent the positions, velocities, and accelerations of points of
bodies. In analyzing the equations of motion or implementing the equations in computer
formulation, vector and matrix algebra are the essential tools in solving multi-body
dynamic problems.

3.2 DADS Analysis Program

A large-scale kinematics and dynamics computer code, DADS (Dynamic Analysis
and Design System) [16], has been developed to implement the analysis for both
kinematics and dynamics of multi-body mechanical systems. It is a very powerful
computer simulation tool used to predict the behavior of single or multi-body systems.

3.2.1 Flow of Modeling in DADS

When we started the DADS program, the first thing we did is to input the data.
The model data is entered and organized through a user-friendly graphical interface. The
data includes bodies, joints, forces, and geometry of the model. Then, we run the DADS simulation. Mathematical equations are formed and solved using algorithms which will be described later. The positions, velocities, and accelerations of each part of the model are predicted. Last, the analysis results can be shown by making plots or by animating the total system behavior.

One thing that is very important is the information file. The information file contains general messages, warnings, and errors from the analysis program. Warnings are usually considered informative, but error messages are problems in the model that make the simulation unsuccessful. The information file is very helpful in getting the general information and debugging the model. A list of independent coordinate names and bodies is also written out to the information file. Initial velocities of all the generalized coordinates are reported.

The flow of modeling is as follows [16]:

Figure 3.1 Flow of modeling
3.2.2 Analysis Types in DADS

There are six types of analysis available in DADS [16]: assembly, kinematic, inverse dynamic, static, linearization of the dynamic equations of motion, and dynamic analysis. Among them, assembly analysis and dynamic analysis are used in this research. The brief description is as follows:

**Assembly analysis**

Unless particularly skipped, this type of analysis is prior to any other type of analysis. The algorithm solves a set of linear or nonlinear equations to find a position that minimizes the “constraint error”, which refers to all the constraints in the model.

**Dynamic analysis**

Dynamic analysis solves a system with a positive number of degrees of freedom and integrates numerically to determine the position, velocity, acceleration of all the bodies. Reaction force of joints is also determined.

3.2.3 Fundamental Kinematics in DADS

**Generalized coordinate**

The basic concept in kinematic analysis is to find position, velocity, and acceleration of specified bodies. Any set of the variables that specifies the position and orientation of all the bodies in a system is a set of generalized coordinates [15]. In DADS, the essential generalized coordinates are Euler parameter orientation generalized coordinates. Here is the brief description:
Euler's Theorem [17]: If the origins of two right-hand Cartesian reference frames coincide, then they may be brought into coincidence by a single rotation about some axis.

The orientation of the new triad system \( (X_n,Y_n,Z_n) \) is obtained by rotating the new triad system through an angle \( \chi \) about a unit vector \( U \) (defined relative to the base coordinate system) as shown in the following figure [16]:

![Euler parameter method](image)

Figure 3.2 Euler parameter method

Euler parameter, along with the location of point \( P \), can be used to orient and position a new triad relative to a base triad. The origin of the new triad (point \( P \)) can be located by three Cartesian coordinates. The Euler parameter \( e_0, e_1, e_2 \) and \( e_3 \) are related to \( \chi \) and \( U \) through the following relationships:

\[
\begin{align*}
  e_0 &= \cos(\chi/2) \\
  e_1 &= U_x \ast \sin(\chi/2) \\
  e_2 &= U_y \ast \sin(\chi/2) \\
  e_3 &= U_z \ast \sin(\chi/2)
\end{align*}
\] (3.1)
They must satisfy the Euler parameter normalization constraint:

$$e_0^2 + e_1^2 + e_2^2 + e_3^2 = 1$$  \hspace{1cm} (3.2)

**Kinematic constraint equations**

For system in motion, generalized coordinates are designated by a column vector,

$$q = [q_1, q_2, \ldots, q_{nc}]^T$$

where \( nc \) is the total number of generalized coordinates. If a mechanical system has \( nb \) rigid bodies, the vector of generalized coordinates for the system is given by

$$q = [q_1^T, q_2^T, \ldots, q_{nb}^T]^T.$$  \hspace{1cm} (3.3)

A system of \( nh \) holonomic kinematic constraint equations which does not depend explicitly on time can be expressed as:

$$\Phi^K(q) = [\Phi^K_1(q), \Phi^K_2(q), \ldots, \Phi^K_{nh}(q)]^T = 0$$

If the time appears explicitly:

$$\Phi^K(q, t) = 0$$  \hspace{1cm} (3.4)

If a system has the generalized coordinates and the consistent and independent constraint equations described above, then the system is said to have \( nc - nh \) degrees of freedom, i.e., abbreviated \( \text{DOF} = nc - nh \). The number of degrees of freedom now can be easily calculated by counting the number of generalized coordinates and subtracting the number of constraint equations.

In DADS, each body has seven generalized coordinates (3D element type), and each joint has one or more constraint equations which determine the mathematical
relationship between the bodies. Here is a table of the relationship between the constraints and generalized coordinates [16]:

Table 3.1 Table of 3D constraints and generalized coordinates

<table>
<thead>
<tr>
<th>3D Element Type</th>
<th>DOF removed</th>
<th>DOF added</th>
</tr>
</thead>
<tbody>
<tr>
<td>body</td>
<td>1</td>
<td>7</td>
</tr>
<tr>
<td>revolute</td>
<td>5</td>
<td></td>
</tr>
<tr>
<td>translational</td>
<td>5</td>
<td></td>
</tr>
<tr>
<td>cylindrical</td>
<td>4</td>
<td></td>
</tr>
<tr>
<td>spherical</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>point</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>position X, Y, Z</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>position, orientation</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>revolute-revolute, intersecting or perpendicular</td>
<td>4</td>
<td></td>
</tr>
<tr>
<td>revolute-translational</td>
<td>4</td>
<td></td>
</tr>
<tr>
<td>revolute-cylindrical</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>revolute-spherical</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>spherical-spherical</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>distance</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>bracket</td>
<td>6</td>
<td></td>
</tr>
<tr>
<td>difference, X, Y, Z</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>ground</td>
<td>6</td>
<td></td>
</tr>
<tr>
<td>universal</td>
<td>4</td>
<td></td>
</tr>
<tr>
<td>screw</td>
<td>6</td>
<td>1</td>
</tr>
<tr>
<td>driver: X, Y, Z, distance, difference,</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>driver, relative angle</td>
<td>2</td>
<td>1</td>
</tr>
<tr>
<td>planar, standard</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>planar, lock out rotation</td>
<td>4</td>
<td></td>
</tr>
<tr>
<td>gear</td>
<td>3</td>
<td>2</td>
</tr>
<tr>
<td>relative</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>point-curve</td>
<td>3</td>
<td>1</td>
</tr>
<tr>
<td>point-surface</td>
<td>3</td>
<td>2</td>
</tr>
<tr>
<td>slide-curve</td>
<td>3</td>
<td>2</td>
</tr>
</tbody>
</table>
Example: Consider the slider-crank mechanism shown in the following figure. We can see that the slider (body 2) is to translate along the x axis. However, several kinematically equivalent models can be used to calculate the DOF of the mechanism. One of the models is illustrated as follows [15]:

![Slider-crank mechanism diagram]

Figure 3.3 Slider-crank with composite joint

Table 3.2 Multi-body summary of slider-crank

<table>
<thead>
<tr>
<th>3D ELEMENT TYPE</th>
<th>NUMBER</th>
<th>DOF ADDED</th>
<th>DOF REMOVED</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rigid bodies</td>
<td>2</td>
<td>$7 \times 2 = 14$</td>
<td></td>
</tr>
<tr>
<td>Euler normalizations</td>
<td>2</td>
<td></td>
<td>$2 \times 1 = 2$</td>
</tr>
<tr>
<td>Spherical-spherical joint</td>
<td>1</td>
<td></td>
<td>$1 \times 1 = 1$</td>
</tr>
<tr>
<td>Revolute joint</td>
<td>1</td>
<td></td>
<td>$1 \times 5 = 5$</td>
</tr>
<tr>
<td>Absolute constraint</td>
<td>1</td>
<td></td>
<td>$5 \times 1 = 5$</td>
</tr>
<tr>
<td>Total generalized coordinates</td>
<td></td>
<td>14</td>
<td></td>
</tr>
<tr>
<td>Total constraint number</td>
<td></td>
<td>13</td>
<td></td>
</tr>
<tr>
<td>Total degrees of freedom</td>
<td></td>
<td>14 - 13 = 1</td>
<td></td>
</tr>
</tbody>
</table>
If DOF independent driving constraints are specified for kinematic analysis, then

\[ \Phi^D (q, t) = 0 \]  \hspace{2cm} (3.5)

The combined constraints of Eq. 3.4 and Eq. 3.5 become

\[ \Phi (q, t) = \begin{bmatrix} \Phi^k (q, t) \\ \Phi^D (q, t) \end{bmatrix} = 0 \]  \hspace{2cm} (3.6)

Next, we use the chain rule of differentiation to evaluate derivatives of both sides of Eq. 3.6 with respect to time and get the velocity equation

\[ \dot{\Phi} = \Phi_q \dot{q} + \Phi_v = 0 \]

or

\[ \Phi_q \dot{q} = -\Phi_v = \nu \]  \hspace{2cm} (3.7)

If we differentiate both sides of Eq. 3.7 with respect to time, using the chain rule again, we can obtain the acceleration equation

\[ \Phi_q \ddot{q} = - \left( \Phi_q \dot{q} \right) \dot{q} - 2 \Phi_v \dot{q} - \Phi_v = \gamma \]  \hspace{2cm} (3.8)

The matrix \( \Phi_q \) that appears in the velocity and acceleration equations is called the Jacobian matrix, or just Jacobian. It is a very import matrix when we deal with the theory or numerical methods of kinematics and dynamics.
3.2.4 Fundamental Dynamics in DADS

We can derive equations of motion of a rigid body and constrained system of rigid bodies from Newton’s laws. However, the virtual work approach is used to develop a self-contained derivation of both variational and differential equation formulations.

The virtual work of a torque \( n \) which acts on the body in the planar system and a force \( F \) which acts at the origin of the \( x' - y' \) frame can be written as [15]:

\[
\delta W = \delta \mathbf{r}^T \mathbf{F} + \delta \phi n \\
= \delta \mathbf{q}^T \mathbf{Q}
\]  

(3.9)

where

\( \delta \mathbf{r} \): virtual displacement, \( \delta \phi \): virtual rotation

\[ \mathbf{q} = [\mathbf{r}^T, \phi]^T, \quad \mathbf{Q} = [\mathbf{F}^T, n]^T \]

The system variation equation of motion in a planar multi-body system (\( nb \) bodies) now can be written:

\[
\sum_{i=1}^{nb} \delta \mathbf{q}_i^T \begin{bmatrix} \mathbf{M}_i \ddot{\mathbf{q}}_i - \mathbf{Q}_i \end{bmatrix} = 0
\]  

(3.10)

If we define a composite state variable vector \( \mathbf{q} \), a composite mass matrix \( \mathbf{M} \), and a composite vector of generalized forces \( \mathbf{Q} \), Eq. 3.10 can be written in more compact form:

\[
\delta \mathbf{q}^T \begin{bmatrix} \mathbf{M} \dddot{\mathbf{q}} - \mathbf{Q} \end{bmatrix} = 0
\]  

(3.11)
where
\[ q = \left[ q_1^T, q_2^T, \ldots, q_{nb}^T \right]^T \] and \( q \) is the virtual displacement vector of body \( i \)

\[ M = \text{diag}(M_1, M_2, \ldots, M_{nb}) \] and \( M_i \) is the mass of body \( i \)

\[ Q = \left[ Q_1^T, Q_2^T, \ldots, Q_{nb}^T \right]^T \] and \( Q_i \) is the generalized force acting on body \( i \)

To reduce the variation equation in Eq. 3.10 to a mixed system of differential-algebraic equations, Lagrange multipliers are introduced. Here is the Lagrange Multiplier Theorem [18]:

Let \( b \) be an \( n \) vector of constraints, \( x \) be an \( n \) vector of variables, and \( A \) be an \( m \times n \) constraint matrix. If

\[ b^T x = 0 \quad (3.12) \]

holds for all \( x \) that satisfy

\[ Ax = 0 \quad (3.13) \]

then there exists an \( m \) vector \( \lambda \) of Lagrange multipliers such that

\[ b^T x + \lambda^T Ax = 0 \quad (3.14) \]

for arbitrary \( x \).

The generalized forces include the applied forces \( Q_i^A \) and constraint forces \( Q_i^C \).

Eq. 3.11 can be written in the form of constrained variational equations of motion:

\[ \delta q^T \begin{bmatrix} M & -Q^A \end{bmatrix} = 0 \quad (3.15) \]
Recall the form of kinematic and driving constraints in Eq. 3.5, the condition for a kinematically admissible virtual displacement $\delta q$ is obtained by differentiate Eq. 3.5 with time as constant; i.e.

$$\Phi_q \delta q = 0$$

(3.16)

From above, we can get the Lagrange form of the equations of motion:

$$M \ddot{q} + \Phi_q^T \lambda = Q^d$$

(3.17)

In addition to the acceleration equations of Eq. 3.8, Eq. 3.17 now may be written in matrix form:

$$
\begin{bmatrix}
M \Phi_q^T \\
\Phi_q \Phi_q^T
\end{bmatrix}
\begin{bmatrix}
\dot{q} \\
\lambda
\end{bmatrix}
= 
\begin{bmatrix}
Q^d \\
\gamma
\end{bmatrix}
$$

(3.18)

This is a mixed system of differential-algebraic equations (DAE).

3.2.5 Analysis Code in DADS

In DADS, the main numerical task is to solve linear and nonlinear equations, and integrate the constrained equations of motion. The following illustrates the key concept used in the above task [15]:

**Linear equation solvers**

LU factorization (described as follows) is the key method used in solving linear equation systems.
Consider a system of \( n \) linear algebraic equations and \( n \) unknowns, the coefficients are real and constants,

\[
\begin{align*}
a_{11} x_1 + a_{12} x_2 + \cdots + a_{1n} x_n &= b_1 \\
a_{21} x_1 + a_{22} x_2 + \cdots + a_{2n} x_n &= b_2 \\
& \vdots \\
a_{n1} x_1 + a_{n2} x_2 + \cdots + a_{nn} x_n &= b_n
\end{align*}
\] (3.19)

which can be written as

\[
Ax = b
\] (3.20)

where

\[
x = [x_1, x_2, \ldots, x_n]^T
\]

\[
b = [b_1, b_2, \ldots, b_n]^T
\]

\[
A = \begin{bmatrix}
a_{11} & a_{12} & \cdots & a_{1n} \\
a_{21} & a_{22} & \cdots & a_{2n} \\
& & \ddots & \vdots \\
a_{n1} & a_{n2} & \cdots & a_{nn}
\end{bmatrix}
\]

Given any nonsingular matrix \( A \), there exits an upper triangular matrix \( U \) with nonzero diagonal elements and a lower triangular matrix \( L \) with unit diagonal elements, such that

\[
A = LU
\] (3.21)

This factorization of \( A \) into \( LU \) is called \( L-U \) factorization. Now the equation

\[
Ax = Lux = b
\] (3.22)
is solved by letting

\[ Ly = b \]  \hspace{1cm} (3.23)

and

\[ Ux = y \]  \hspace{1cm} (3.24)

Eq. 3.23 is first solved and get y, then Eq. 3.24 is solved and get x.

**Nonlinear equation solvers**

To find the solutions of nonlinear equations of the form,

\[ \Phi(q) = 0 \]  \hspace{1cm} (3.25)

Newton-Raphson method is used in DADS.

Let \( q = q^* \) be a solution of Eq. 3.25 and define \( q^{(i)} \) is an approximation of \( q^* \).

From Taylor series expansion of \( \Phi(q) \) at the point \( q = q^{(i)} \), we neglect the higher order terms and rearrange the expression, Newton-Raphson algorithm is obtained:

\[ q^{(i+1)} = q^{(i)} - \frac{\Phi(q^{(i)})}{\Phi_q(q^{(i)})} \]  \hspace{1cm} (3.26)

Whether the sequence of approximate solutions is divergent or convergent is very important. If the algorithm converges and the Jocobian is nonsingular, there is a constant \( c \) such that

\[ |q^{(k+1)} - q^*| < c |q^{(k)} - q^*|^2 \]  \hspace{1cm} (3.27)
The following figures can illustrate the Newton-Raphson method: Fig. 3.4 is the ideal situation when the iteration continues. However, if the root is at the inflection point, the iteration may diverge, as indicated in Fig. 3.5. Another point is the estimate of the solution must be reasonable, i.e. the initial value is very important. As in Fig. 3.6, there are multiple roots. If the initial estimate is not adequate, we may obtain the solution that is not desired. Fig. 3.7 shows if the initial estimate is near a local minimum or maximum, the iteration may diverge.

Here are the figures that describe each situation [15]:

![Figure 3.4 Newton-Raphson method](image)

Figure 3.4 Newton-Raphson method

![Figure 3.5 Root at inflection point](image)

Figure 3.5 Root at inflection point
Numerical integration methods

There are several numerical integration methods used in DADS. In this research, the predictor-corrector approach is used because it is more accurate and error estimates are more reliable. In DADS interface, this method is called PECE method. This method predicts first step and evaluate current function value, then it automatically corrects to next step, and evaluate the function value at that time step. This method gives high accuracy of the solutions of nonlinear equations.
CHAPTER IV

TRUCK MODELING USING DADS

4.1 Background

A complete heavy-duty truck model includes the multi-body formulation of the truck body, suspension characteristics, steering system, tire forces, engine, powertrain, braking, aerodynamic forces, and so on. Since the purpose of this research is to build a model to study the handling behavior of the heavy-duty truck, traction characteristics are not of our interest. Thus, engine and powertrain systems are not included in the model.
The heavy-duty truck is assumed to be driven on a perfectly flat road to study the cornering behavior without braking, so the braking system is not included in the model. To keep the truck at constant speed during cornering, a simple cruise control is used. Aerodynamic forces actually contribute certain effect on the vehicle handling behavior especially when the vehicle is driven at high speed. However, this effect is not considered in the model due to the difficulty in the parameter measurement.

This research focuses on the multi-body formulation of the truck body, suspension system, steering system, and tire forces. The following figure illustrates role of each subsystem in the heavy truck model:

![Diagram showing the flow of forces and inputs in the truck model]

**Figure 4.1 Total heavy-duty truck model composition**

To define the vehicle motion, an appropriate coordinate system is needed. Here in the model we use the SAE standard axes shown in figure 4.2 [40]. In the global frame the
coordinate is called the earth-fixed system \((X,Y,Z)\) - This system is a right hand orthogonal system fixed on the earth. The \(X\)- and \(Y\)-axis are in a horizontal plane and the \(Z\)-axis is directed downward.

Another system which is a right-hand fixed orthogonal coordinate system originating at the center of gravity of the chassis of the heavy-duty truck moving with the truck steadily in a straight line on a level road is called the vehicle axis system \((x,y,z)\). The \(x\)-axis is substantially horizontal, points forward, and is in the longitudinal plane of symmetry. The \(y\)-axis points at driver's right and the \(z\)-axis points downward.

The orientation of the vehicle axis system \((x,y,z)\) with respect to the earth-fixed axis system \((X,Y,Z)\) is given by a sequence of three angular rotations: A yaw rotation, \(\Psi\), about the aligned \(z\)- and \(Z\)-axis, a pitch rotation, \(\Theta\), about the vehicle \(y\)-axis, and a roll rotation, \(\Phi\), about the vehicle \(x\)-axis.

![Figure 4.2 SAE standard axes](image-url)
4.2 Suspension Modeling

4.2.1 General Concept

There are three basic concepts in the suspension modeling: Leaf spring force, damping force, and roll center concept. The following describes the general idea of each one separately.

**Leaf spring force**

Leaf springs are the most popular suspension elements in commercial road vehicles. Many researchers work on the behavior of the leaf springs. Traditional calculations based on simple beam theory are extended to describe large deflections. These calculations neglect the inter-leaf friction but estimate the spring rate of large deflections [19]. Greening [20] created a spring model consisting of two or three cantilevered leaves contacting at discrete points at their tips. He suggested that the beams were locked together until the shear force at a given contact point exceeded the local friction, allowing the beams to slide. However, this model did not explain the different spring rates observed on loading and unloading. Whittemore et al. [21] developed a computer model which comprised four linear springs and two friction elements. This model showed similar characteristic to Greening’s model.

According to Fancher’s study [22], truck leaf springs are very complicated force-producing mechanisms which may show different levels of effective spring rate and damping, coulomb friction, or hysteresis; depending upon the amplitude of the oscillation and the loading of the spring. Leaf springs are nonlinear devices which
dissipate energy through inter-leaf friction during each cycle of oscillation. Fancher et al. performed a series of sinusoidal tests on heavy truck leaf springs and represented the force-deflection characteristics of leaf springs in equation from which was suitable for computer simulation. This study has been examined by D. Cebon [19] who examined the characteristics of three different types of road vehicle leaf springs and compared the results with Fancher’s empirical equation. He concluded that Fancher’s equation could accurately describe the leaf spring behavior under broad band random excitation.

Fancher first used five springs to measure the force-deflection properties. The springs tested were: multi-leaf front spring, tapered leaf front spring, multi-leaf rear spring with torque rod, one spring from a four-spring suspension with torque rod incorporating shackle connections at the equalizer link and slipper connections at the opposing ends of each spring, and one spring from a four spring suspension with torque leaves. During the test, it was found out that the “effective mass,” $M_e$, of the spring will noticeably distort the force results at relative high stroking accelerations. In order to correct for the mass effect, an accelerometer was used to measure the acceleration $\ddot{z}$. Then $M_e \ddot{z}$ was subtracted from the total stroking force. From the test data, it was clearly indicated that the force-deflection properties of truck leaf springs depended upon both the amplitude of motion and the nominal load. Then the average coulomb damping force $C_F$ and the effective spring rate $K_e$ were developed. The parameters $C_F$ and $K_e$ defined an
approximate hysteresis loop which had the same amplitudes of force and deflection as the measured hysteresis loop and the same energy dissipation per cycle as the leaf spring.

The following figure shows the estimation of the effective spring rate, where \( A \) is the area enclosed within the hysteresis loop generated in a plot of force versus displacement, and \( \delta \) is the amplitude of the spring deflection (peak-to-peak amplitude is \( 2\delta \)) [22]:

\[
A = (2C_F \times 2\delta) \quad \text{or} \quad C_F = \frac{A}{4\delta} \\
2f = 2C_F + 2\delta K_e \\
\text{or} \\
K_e = \frac{(2f - 2C_F)}{2\delta}
\]

Figure 4.3 Estimation of the effective spring rate

However, the representative spring rates and friction levels obtained this way are only valid for the operating conditions (load and stroke) from which they were derived. To represent the characteristics of the leaf springs over wide ranges of loading, deflection amplitude, and random reversals of velocity, an equation has been devised:

\[
F_i = F_{ENVI} + (F_{i-1} - F_{ENVI})e^{-|\delta_i - \delta_{i-1}|/\delta}
\] (4.1)
where

$F_i$ is the suspension force at the current simulation time step

$F_{i-1}$ is the suspension force at the last simulation time step

$\delta_i$ is the suspension deflection at the current simulation time step

$\delta_{i-1}$ is the force corresponding to the upper and lower boundaries of the envelope of the measured spring characteristics at the deflection, $\delta_i$

$\beta$ is an input parameter used for describing the rate at which the suspension force within a hysteresis loop approaches the outer boundary of the envelope, $F_{ENV}$. Different values of $\beta$ for increasing versus decreasing deflection may be specified.

**Damping force**

The damping force in the suspension system refers to the force generated by the shock absorber. The function of the shock absorber is to dissipate energy in the jounce and rebound directions. Although the Coulomb friction in the leaf spring provides the damping effect in the suspension, some heavy-duty truck suspension designs still need additional damping devices like shock absorbers to contribute to the damping force to make the suspension system stabilize efficiently.

To study the characteristics of the damping force generated by the shock absorber, a series of tests were performed [41]. The mathematical model for the shock absorbers commonly used in the vehicle dynamics modeling appears in the form of viscous
damping. That is, damping in which the force opposing the motion is proportional and opposite in direction to the velocity:

$$f = C \dot{x}$$  \hspace{1cm} (4.2)

where $f$ is the damping force generated by the shock absorber, $C$ is the damping constant for the rebound and jounce motion (sometimes we use $C_1$ for the jounce and $C_2$ for the rebound to get more accurate description). The following figure illustrates the model:

![Figure 4.4 Shock absorber model](image)

However, for the tests performed in the laboratory, the damping force is not always merely a function of the velocity. Some data shows the damping force may be the function of both velocity and position. And especially in the high frequency excitation, the force-velocity diagram shows a hysteresis loop instead of a straight line. This
phenomenon is explained as the existence of a gas phase or the compressibility of the fluid in the shock absorber. But for the typical vehicle dynamics modeling, the non-linearities are often neglected. A simple linear shock absorber model can reach certain degrees of satisfaction for most vehicle dynamicists.

Roll center concept

To describe the handling behavior of the heavy truck, the “roll axis” concept is very important. Roll axis is the straight line connecting the roll center of the front suspension and the roll center of the rear suspension. To define the roll axis, we have to define the “roll center” concept first. We assume that each suspension has a suspension roll center, which is defined in many ways. According to J. C. Dixon’s study [28] of roll center concept in vehicle handling dynamics, the roll center concept has been used since 1930’s. But there was no single paper that gave it a clear definition. J. C. Dixon shows that there are actually two roll centers or roll axes, and it is very important to distinguish between them. One is the kinematic roll center, and the other is force roll center.

Many authors define the roll center as a kinematic concept. For example, from Ellis [25]:

“The roll centers themselves have been taken as kinematic centers of rotation of the suspension, assuming that the wheels are rigid and do not move sideways on the road surface.”

From Segel et al. [26]:

“... a roll center, defined as the instant center about which the sprung mass must
roll with respect to the ground...” and

“...the roll axis... the instantaneous axis about which the sprung mass rolls.”

From Artamonov et al. [27]:

“The roll center is the point with respect to which the transverse section of the vehicle passing through the front or rear axle moves.” and

“The roll axis is the straight line with respect to which the body being inclined turns transversely.”

The kinematic roll center concept is based on assumption that all the members are as rigid links, with perfect pin joints between the members. We have to be very careful because symmetry is assumed in this concept to find the roll center. However, if the suspension is asymmetric or the roll angle of the vehicle is significant and the vehicle has a large lateral acceleration, then this method has to be modified.

SAE defines the roll center using another concept, i.e. the force roll center concept [29]:

“The point in the transverse vertical plane through any pair of wheel centers at which lateral forces may be applied to the sprung mass without producing suspension roll.”

The force roll center is developed using force analysis only, and is of good prediction for the behavior of the vehicle under high lateral acceleration. In other terms, the force roll center lies at a height coincident with the point where lateral forces are transmitted between the sprung and unsprung masses.
The roll axis is actually an instantaneous axis when we define the roll center above. As the body roll begins, the change in geometry will make the center of each suspension move. J. C. Dixon suggests a new incremental roll center concept which considers the load transfer and predicts the roll center height at the next time step.

Before we really get the measured roll center height from the laboratory, we can estimate the roll center height of each suspension using the empirical approximation as the following figures [24]:

![Diagram of roll center estimation](image)

Figure 4.5 Typical font axle suspensions

The front suspension of the model is the leaf spring type with shackles. Figure 4.5 shows the approximation of the roll center height of front suspension.
The rear suspension of the model four spring tandem axle type. Figure 4.6 shows the estimation of the roll center height of the rear suspension.

4.2.2 Front Suspension

The front springs of the heavy truck are the semi-elliptic type with the spring leaves being held together by spring clips. Springs are attached to the front axle beam pads by two U-bolts and secured at the frame side rails by a stationary bracket at the one end and a swing shackle at the other. The stationary bracket is bolted solidly to the frame rail and the spring is supported on a pin which is clamped in position by two binder bolts for the heavy duty suspension. The two main spring leaves are wrapped at one end to form an eye into which is pressed a replaceable bushing. The spring pin is drilled to accommodate a grease fitting for lubrication of the spring eye bushing.

The swinging shackle at the other end of the front spring permits lengthening or shortening of the spring while the vehicle is in motion.
The following figure illustrates the front suspension [23]:

![Front Suspension Diagram]

- 1. rear hanger bracket
- 2. shocks
- 3. front spring bracket
- 4. swing shackle
- 5. upper plate
- 6. U-bolt
- 7. pinch bolt
- 8. spring assembly
- 9. U-bolt
- 10. axle I beam
- 11. spring clip

Figure 4.7 Illustration of front suspension

To predict the kinematic motion between the front axles and the chassis of the truck, several models are shown:

The first one is trailing arm model. We know that every suspension system is equal to a trailing arm suspension. The following figure shows the trailing arm suspension:

![Trailing Arm Suspension Diagram]

Figure 4.8 Trailing arm suspension model
At point A, the trailing arm and the chassis are connected by a revolute joint. The revolute axis is about the vehicle y-axis. The trailing arm and front axle can be treated as the same body. The location of point A is determined from the experiment by fixing the chassis stationary and applying force from the tires and keeping the trajectory of the center of the axle in the x- and z-axis direction. Then we can estimate the location of point A. One thing that is very important is if we do not use the revolute joint at point A and just create a distance constraint between point A and point B, the axle can not be constrained in the x-z plane. The motion of the axle is then not desirable.

The second model provides more detail in the leaf spring kinematic behavior and it can model the leaf spring windup under the braking torque. SAE [42] has a method to approximate the leaf spring as three links. The model is based on the three links approximation method and is illustrated as the following figure:

![Three links approximation method](image)

Figure 4.9 Three links approximation method
The lengths and locations of link one, two, and three are determined according to the approximation method. Link four is added to account for the shackle effect. At point A, B, and C, revolute joints are used. At point D and E, only a spherical and a cylindrical joint are used to prevent overconstraining the system. At point B and C, torsional springs are used to simulate the leaf spring forces.

In this research, the x-axis direction of the axle relative to the chassis of the truck is not very important because it mainly affects the ride behavior. Besides, the braking capabilities is not of interest so we do not need the detailed three link approximation method to account for the leaf spring windup effect. We can assume the axle only moves in the z-axis direction of the chassis of the truck. Because we are only interested in the handling behavior of the heavy-duty truck without braking, a translational joint between the axle and the chassis is enough. This assumption eliminates many joints and bodies needed by the first two models and saves a lot of computation time in DADS without sacrificing much accuracy.

Another important concept in the suspension model is the roll center concept described previously. To model the vehicle handling behavior, for example the roll motion of the chassis under lateral force, there are two methods: The first one is to get all the precise geometries of the leaf spring, including connecting points between the leaf spring and the chassis and the points between the leaf spring and the axle. From the following figure we can see that the leaf springs “bend” when the chassis rolls. Then we have to perform the finite element calculation of the leaf spring to get the correct
deformation. This includes a lot of experimental work to get the stiffness matrix. Besides, the bushing element between the leaf spring and the chassis has to be simulated because it contributes a lot in the relative deformation between the chassis and the leaf spring. However, the bushing element is usually very stiff and often introduces high frequencies into the model. Integration step size thus will decrease and cost more computation time, which is not desirable.

![Diagram of truck suspension](image)

*Figure 4.10 Roll motion of the chassis of the truck*

For most passenger cars with independent suspension system, the simulation of the roll motion of the chassis can be achieved just by the lengthening and shortening of the coil spring along a straight line without introducing the "bending" effect of the coil spring. Thus, it is not necessary to perform the finite element calculation as heavy-duty
truck with solid axle suspension. The following figure illustrates the roll motion of a passenger car with independent suspension system.

![Diagram of roll motion](image)

Figure 4.11 Roll motion of a passenger car with independent suspension system

The second method is to adopt the roll center concept. This method is more efficient than the previous one because the only parameter we have to obtain from the laboratory is the location of the roll center, i.e. the roll center height, of the suspension. This relieves us from performing the time-consuming finite element calculation and modeling the high frequency response from the bushing element which is of no interest for the handling analysis. We define the revolute motion between the chassis and the axle is about the roll center of the suspension. Leaf spring force is applied between the chassis and the axle in a straight line without any constraint. Thus, we can conclude the relative motions between the chassis and the axle are: 1) The revolute motion at the roll center of the suspension. 2) The translational motion as discussed previously. We can use a dummy
body and set its mass properties to be very small. The two relative motions can be modeled by applying a revolute joint between the chassis and the dummy body and a translational joint between the dummy body and the axle. In DADS, there are composite joints between pairs of bodies available. In this case, the dummy body only serves as a kinematic constraint between bodies that are connected. We can use a composite joint "revolute-translational joint" instead which is convenient and computationally efficient. The intermediate dummy body thus is not needed.

In DADS, the revolute-translational joint between body \(i\) and body \(j\) is shown in Figure 4.12 [6]. A coupler between the bodies that is pivoted about \(\mathbf{h}_i\) in body \(i\). Body \(j\) can translate along vector \(\mathbf{h}_j\), but cannot rotate about this axis relative to the coupler. Analytical conditions that define the revolute-translational joint are that vectors \(\mathbf{h}_i\) and \(\mathbf{f}_j\) be parallel and that, if \(\mathbf{d}_{ij} \neq 0\), it must be parallel to vector \(\mathbf{h}_j\), that is:

\[
\Phi^1(\mathbf{h}_i, \mathbf{f}_j) = 0
\]

\[
\Phi^2(\mathbf{h}_j, \mathbf{d}_{ij}) = 0
\]

(4.3)

The four scalar constraint equations of Eq. 4.3 allow two relative degree of freedom between bodies \(i\) and \(j\).
From above, now we can construct the front suspension by using the revolute-translational concept as following figure:

![Figure 4.13 Model of front suspension](image)

The roll center height obtained for the front suspension from [5] is $29.0 \text{ in} = 0.737 \text{ m}$, and the leaf spring force and shock absorber force application location from [43] is $d = 16.3 \text{ in} = 0.414 \text{ m}$. The leaf spring force and shock absorber force are implemented in DADS by creating the translational spring-damper actuator element (TSDA). The TSDA
is a force-producing element. It can contain any or all of the following: a linear spring, a nonlinear spring, a free length, a linear damper, a nonlinear damper, a constant force-producing actuator, a time-varying force producing actuator and either a Bi-directional, Tension- or Compression-only toggle mode. The TSDA element is defined by two triads, one on each of the two bodies connected by this element and the force produced is a straight line connecting the triad origins referenced. The following figure [16] shows the TSDA element concept:

\[ F = K(d - d_0) + cd + F_A + F_K(d) + F_C(d) + F_A(t) \]

Total Force = Spring Constant * (relative distance)  
+ Damping Coefficient * (relative distance dot)  
+ Constant Actuator Force  
+ Nonlinear Spring Force  
+ Nonlinear Damper Force  
+ Time-varying Actuator Force

Figure 4.14 Translational Spring-Damper-Actuator
For the front suspension, we create four TSDA elements to model left and right leaf spring force and shock absorber force separately. Linear damping force is included in the model. From the experiment [24], the damping coefficient for the shock absorber is 4903 N·sec/m. However, the leaf spring behavior is much more complicated from the previous discussion. The empirical formula: 

\[ F_i = F_{ENVi} + (F_{i-1} - F_{ENVi})e^{-\frac{\delta_i - \delta_{i-1}}{\beta}} \]

has to be included in the model. In DADS, we can set the spring constant to be zero and open the user-defined FORTRAN file UFRC10 in the DADS subdirectories. UFRC10 is used to modify the force calculations in the TSDA element. All the information defined in the TSDA element including the spring free length, triads position and orientation, and current distance will be passed into UFRC10 file. We can use the information and program the leaf spring force in the file. Because only one UFRC10 routine is allowed for any simulation model even if there are more than one TSDA element to modify, we use the variable ENMBR to distinguish between them. This number starts at one and is incremented by one for each TSDA. Thus the number will correspond to the order in which DADS encounters the TSDA during input.

The experiment data [22] for the front multi-leaf spring are as follows:

<table>
<thead>
<tr>
<th>( \delta ), in</th>
<th>( F_{ENV} ), lbs</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>0.6</td>
<td>1080</td>
</tr>
<tr>
<td>1.2</td>
<td>1860</td>
</tr>
<tr>
<td>1.8</td>
<td>2640</td>
</tr>
<tr>
<td>2.4</td>
<td>3420</td>
</tr>
</tbody>
</table>
\begin{align*}
3.0 & \quad 4200 \\
3.6 & \quad 4980 \\
4.2 & \quad 5760 \\
4.8 & \quad 6540 \\
\beta = 0.08 \text{ in}
\end{align*}

For \( \delta_i < \delta_{i-1} \)

\begin{align*}
\delta, \text{ in} & \quad F_{ENV}, \text{ lbs} \\
0.0 & \quad 0.0 \\
0.6 & \quad 560 \\
1.2 & \quad 1220 \\
1.8 & \quad 1880 \\
2.4 & \quad 2540 \\
3.0 & \quad 3200 \\
3.6 & \quad 3860 \\
4.2 & \quad 4520 \\
4.8 & \quad 5180 \\
\beta = 0.076 \text{ in}
\end{align*}

To get the upper and lower envelopes from the experiment data. The CURVE element is included in the model. SPLINE CURVE and CUBIC INTERPOLATION are used in this case. Sets of \((x,z)\) points are defined explicitly. \(x\) is the independent value, while \(z\) is the dependent value. The result is a curve in the 2-dimensional \(x\)-\(z\) plane.

In UFRC10 routine, we call the curve element by the index of the element. The index is returned by the GETNUM statement. During the simulation, the values of the current upper and lower envelopes are obtained by SPL48 statement, which computes the quantities for a cubic or linear interpolated curve element. The spline curve algorithm fits a cubic polynomial between the data points that define the curve.
4.2.3 Rear Suspension

The rear suspension of the heavy truck is the four spring tandem axles type. In order to allow the transport of large payloads without incurring large axle loads, many trucks use this tandem axles design. The tandem axles suspension is defined as a group of two or more axles whose responses to vertical load are coupled by a mechanism basically intended to maintain equal vertical loading among those axles [24]. The four spring tandem axles suspension provides load equalization by articulating the short “load leveler” or “load equalizer” between the spring. For example, during braking, originally the load is increasing in the leading axle. But through the articulation of the load equalizer (rotating upward at the front), the load is transferred from the leading axle onto the trailing axle, i.e. the load equalizer rotates to maintain equal loads resulting in decreased loading of the leading axle and increased loading of the trailing axle. The following figure illustrates the situation [24]:

Figure 4.15 Interaxle load transfer of the four spring tandem axles suspension
Here is the detailed description of the rear suspension of the heavy truck: The torque acting on the suspension as a result of vehicle motion is absorbed through the radius spring leaf (torque leaf) on all four springs. Adjustment is made by shims at the equalizer and front spring brackets. The function of the shimming is to provide a means for tracking alignment of the tandem axles when installed on the chassis.

The torque leaves are cupped at the center spring bolt hole, providing a stronger and rigid mounting at the spring seat. Steel-backed rubber bushings are pressed into the radius leaf eyes and are installed to the equalizer and front spring brackets.

A steel rebound bolt is located at each frame spring bracket that stops the main spring leaf from slipping out of the frame bracket. The following figure shows the side view of the rear suspension [23]:

![Side view of rear suspension](image_url)
The tandem axles can be modeled by using two revolute-translational joints and adopting the roll center concept as the front suspension. Two additional bodies called the right leveler and the left leveler can be added to simulate the load equalizing effect. The load leveler is connected to the chassis of the truck with a revolute joint and the TSDA element is applied between the axle and the load leveler not the chassis. The following figure shows the model of the leaf suspension with the load leveler:

![Diagram of rear suspension with load levelers](image)

**Figure 4.17 Model of the rear suspension with load levelers**

The "interaxle load transfer" effect modeled by the load levelers is primarily important to the ride analysis but not in the handling analysis. In this research, we eliminate the two load levelers and apply the TSDA element directly between the axles and the chassis of the truck to make the model more efficient. The following figure shows the model of the rear suspension:
Figure 4.18 Model of the rear suspension

From the experiment [24], the roll center height for the rear suspension is 29.0 in = 0.737 m, and the leaf spring force and shock absorber force application location from [43] for the leading and trailing axles are \( d_1 = 20.2 \text{ in} = 0.513 \text{ m}, \ d_2 = 20.3 \text{ in} = 0.516 \text{ m} \) separately. The damping coefficient is 175 \( N \text{-sec/m} \) which is much smaller than the damping coefficient of shock absorbers in the front suspension.

The experiment data [22] for the rear four spring suspension with torque leaves are as follows:

<table>
<thead>
<tr>
<th>( \delta, \text{ in} )</th>
<th>( F_{ENV}, \text{ lbs} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>0.1</td>
<td>740</td>
</tr>
<tr>
<td>0.3</td>
<td>1500</td>
</tr>
<tr>
<td>0.6</td>
<td>2630</td>
</tr>
<tr>
<td>0.9</td>
<td>3800</td>
</tr>
<tr>
<td>1.2</td>
<td>5000</td>
</tr>
<tr>
<td>1.5</td>
<td>6520</td>
</tr>
</tbody>
</table>
\[ \beta = 0.02 \text{ in} \]

For \( \delta_i < \delta_{i-1} \)

<table>
<thead>
<tr>
<th>( \delta ), in</th>
<th>( F_{E_{nv}}, lbs )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>0.1</td>
<td>250</td>
</tr>
<tr>
<td>0.3</td>
<td>750</td>
</tr>
<tr>
<td>0.6</td>
<td>1550</td>
</tr>
<tr>
<td>0.9</td>
<td>2600</td>
</tr>
<tr>
<td>1.2</td>
<td>3750</td>
</tr>
<tr>
<td>1.5</td>
<td>5090</td>
</tr>
<tr>
<td>1.8</td>
<td>6700</td>
</tr>
<tr>
<td>2.0</td>
<td>7890</td>
</tr>
<tr>
<td>2.2</td>
<td>9050</td>
</tr>
</tbody>
</table>

\[ \beta = 0.02 \text{ in} \]

We can use the TSDA elements and UFRC10 routine again to model both the leaf spring force and shock absorber force of the rear suspension. The procedure is not restated here again.

4.3 Steering System Modeling

The design of the truck steering system has a great influence on directional response behavior. The majority of commercial trucks over 10,000 pounds gross vehicle weight built in the United States use a solid front I beam axle as the steering axle with a
leaf spring suspension [38]. In this research, the heavy truck has the same type of steering system. The following is the illustration of the truck steering systems [39]:

![Illustration of truck steering systems](image)

**Figure 4.19 Illustration of truck steering systems**

The steering wheel is connected by the steering column. Then through the U-joint and the steering shaft, the rotary motion from the steering is transformed by the steering gearbox to create the desired numerical reduction and the desired rotary direction. The Pitman arm then converts the output torque from the steering gearbox to the drag link. Steering arm converts the drag link force into a turning movement through the knuckle.
The following figure shows more detail of the mechanism from the steering arm to the road wheel [23]:

![Steering Linkage Diagram](image)

Figure 4.20 Steering linkage of the truck

The right tie rod is like a mirror image of the left and converts the force from the tie rod into a movement to turn the right knuckle, wheel and tire around the kingpin. The right knuckle and Kingpin assembly is similar to the left except that it does not have a steering arm attached to it. There is one thing we should notice is that ball joints are used on both ends of the drag link and tie rod to provide for the rotational movement and angular displacement of these parts due to the road wheel rotation and suspension deflection.
In addition to the linkage geometry, one important element of the steering system is the kingpin geometry. This geometry decides the moment and force reaction experienced in the steering system, and could affect deeply the vehicle performance. To define the kingpin geometry, SAE [29] gives the following terminology:

Wheel plane - The central plane of the tire, normal to the spin axis.

wheel center - The point at which the spin axis of the wheel intersects with the wheel plane.

Center of tire contact - The intersection of the wheel plane and the vertical projection of the spin axis of the wheel onto the road plane.

Kingpin inclination - The angle in front elevation between the steering axis and the vertical.

Kingpin offset - Kingpin offset at the ground is the horizontal distance in front elevation between the point where the steering axis intersects the ground and the center of tire contact.

Caster angle - The angle in side elevation between the steering axis and the vertical. It is considered positive when the steering axis is inclined forward.

The following figure shows the kingpin geometry [40]:

[Diagram of kingpin geometry]
Historically, the steering axis has the name “kingpin axis” [40], and it is usually not vertical, but tipped outward at the bottom, causing a kingpin inclination angle. For heavy trucks, the angle is usually in the range of 0-8 degrees. Another angle, caster angle, provides the capability to produce self-aligning moment in the steering system for returnability.

In this research, the steering system is modeled as the kinematic steering system. In other words, we predict the steering motion from the handwheel to the roadwheel by the geometry and the connecting joints, but ignore the compliance existing among the linkages. An arbitrary steering motion is kinematically driven as we want. The handwheel torque input and the torque feedback from the aligning moment acting upon the roadwheel is not of our interest.

Four bodies are included in the steering model: The front axle, left wheel assembly, right wheel assembly, and the pitman arm. The left wheel and the right wheel are connected to the front axle by two revolute joints. In DADS, the revolute joint (or
rotational joint) between bodies $i$ and $j$ is constructed with a bearing that allows relative rotation about a common axis, but precludes relative translation along this axis, as shown in the following figure [16]. To define the revolute joint, the center of the joint is located on bodies $i$ and $j$ by points $P_i$ and $P_j$. The axis of relative rotation is defined in bodies $i$ and $j$ by points $Q_i$ and $Q_j$, and unit vectors $\mathbf{h}_i$ and $\mathbf{h}_j$ along the respective $z$ axes of the joint definition frames. The analytical formulation of the revolute joint is that points $P_i$ and $P_j$ coincide and that body-fixed vectors $\mathbf{h}_i$ and $\mathbf{h}_j$ are parallel, leading to the constraint equations

$$\Phi^i(P_i, P_j) = 0$$

$$\Phi^{ji}(\mathbf{h}_i, \mathbf{h}_j) = 0$$

(4.4)

These five scalar constraint equations yield only one relative degree of freedom, rotation about the common axis of the bearing.

Figure 4.22 Revolute joint
In the steering system model, the drag link which connects the pitman arm and the left wheel assembly and the tierod which connects the left and right wheel assembly are not modeled as bodies but distance constraints instead. If we model the drag link and the tierod as bodies, the spherical joints are applied as the connecting joints between the bodies. One thing that is very important is that an additional rotational constraint has to be imposed on the drag link and the tierod respectively, otherwise the drag link and the tierod will spin about themselves, which is not a desirable motion.

To make the model more efficient, the distance constraint is a better choice. In DADS, the distance between points \( P_i \) and \( P_j \) on bodies \( i \) and \( i \) can be fixed and equal to \( C \neq 0 \), as shown in the following figure [16], by direct application of the distance constraint of Eq. 4.5

\[
\Phi^{SS}(P_i, P_j, C) = 0 \tag{4.5}
\]

The scalar constraint equation permits five relative degrees of freedom between bodies \( i \) and \( j \).

![Distance constraint](image-url)
Another revolute joint included in the steering system is applied to connect the pitman arm and the chassis. Actually the pitman arm is connected to the gear box as the output and the handwheel followed by the steering column is connected to the gear box as the input. Since the gear box is fixed to the chassis of the truck. We can lump its mass to the mass of the chassis. At this joint, we can create a DRIVER element which defines a set of variables (generalized coordinates) that are to be kinematically driven and the way in which they are driven. To simulate the steering input, the driver type is set to be REL ANGLE (relative angle). It drives the relative angle from the $X^*$- axis on the joint's first body (pitman arm) to the $X^*$- axis of the joint's second body (chassis). One of several standard functions may be specified for the coordinate that is to be driven. We can define the coordinate as a POLYNOMIAL function in time or a HARMONIC function in time. The driven coordinate may also be specified from a CURVE element in the GENERAL function option. Finally, the value of the variable can be prescribed by any continuously twice-differentiable function of time by compiling a user-defined subroutine USRDRV3D.

From the measurement of steering geometry, caster angle is 2 deg, kingpin inclination angle is 8 deg, and the camber angle is 0 deg.

The following figure shows the basic geometry:
In DADS, the revolute joint is defined by two triads, one on each of the two bodies connected by the joint. The $Z^-$ axis of the triads define a common axis about which the two bodies are allowed to rotate. From the above measured data, we first rotate the triads about the $Y^-$-axis 2 deg, then rotate the triads about the new $X^-$-axis 8 deg. Thus we can get the exact steering axis orientation.

From the measured geometry [43], we can also get the length of the drag link and the length of the tierod, in other words, the distances specified by the distance constraints we created: The length of the drag link is $0.80519 \, m$ and the length of the tierod is $1.6479 \, m$. One thing we should notice is that the distance constraint element in DADS requires the distance between the two triads origins be the value we specify for distance or as
close as possible to avoid exceeding the assembly tolerance. Otherwise, the analysis program will be aborted due to the failure of assembly.

The following figure shows the steering system model:

![Steering System Model](image)

**Figure 4.25** Steering system model

### 4.4 Tire Forces

To fully describe the handling behavior of the heavy truck, tire forces modeling plays an very important role. Many unsatisfactory vehicle dynamics simulation in the early days were claimed to be due to their lack of good tire force models. So only a clear understanding of the tire forces can lead to a successful vehicle dynamics modeling. To
precisely describe the operating conditions, forces, and moments experienced by a tire, SAE [29] has defined the axis system shown as the following figure:

Figure 4.26 SAE tire axis system

The X-axis is the intersection of the wheel plane and the road plane with a positive direction downward. The Y-axis is in the road plane and its direction is chosen to let the axis system orthogonal and right-hand.
The existing models that are widely used in vehicle dynamics modeling are: The Calspan model [30,31], developed by McHenry, R. R.; the STI model, developed by Systems Technology Inc. [32]; the Non-Dimensional model, developed by Radt, H.S. [33,34]; and the Magic Formula model, developed by Pacejka, H. B. [35,36]. The Calspan model and the STI model are very popular in the United States. Actually the STI model is a consolidation of the Calspan model. The Normalized model is recognized because of its concise parameter descriptions. And the Magic Formula model is mainly used by European researchers.

However, most existing tire models do not have the capability to fully describe the dynamic characteristics of the output responses during severe maneuvers like crash-avoidance type maneuvers. Though in the Magic Formula model, Pacejka pointed out the dynamic factors on the tire output responses, he did not include the experimental data to support his theory. STI model is the only model which has included the dynamic output responses. And according to Lee's study [37], compared with the experimental data, the STI model is the best in overall prediction capabilities among the existing empirical models described above. Due to the difficulty of getting all the parameters in the experiment data, a simple linear tire model is used in this research.

To perform all the tire force calculation, a user-defined FORTRAN file FRC48 is used. The FRC48 routine is called at each analysis time step to calculate the components of the user-defined force expressions. This routine is useful because information such as
the position, velocity, and acceleration is passed onto the FRC48 routine and allows us to perform the calculation based on the information.

The first step is to get the normal force of the tire. We can assume that the normal force is a function of both the deflection of the tire and the vertical velocity of the wheel hub in the local wheel coordinate system (or the deflection rate of the tire). A constant spring-damper concept is used to get the normal force of the tire:

\[ F_z = T_k \delta + T_d \dot{\delta} \]  

(4.6)
where \( F_z \) is the normal force, \( T_k \) is the vertical stiffness of the tire, \( T_d \) is the vertical tire damping coefficient \( \delta \) and \( \dot{\delta} \) are the deflection and the deflection rate of the tire respectively. \( \delta \) and \( \dot{\delta} \) can be obtained by using Q,QD arrays and GETPVA statement to get the current position and velocity of the wheel center. From the experiment [24], \( T_k \) is 900000 \( N/m \) and \( T_d \) is 3000 \( N\cdot sec/m \).

The next step is to get the lateral force of the tire. First we have to define the slip angle \( \alpha \). The slip angle of the tire is the angle between its direction of heading and its direction of travel. That is,

\[ \text{slip angle } \alpha = a \tan \left(\frac{V_y}{V_z}\right) \]  

(4.7)
where \( V_z \) is the longitudinal speed and \( V_y \) is the lateral speed of the wheel center in the local wheel coordinate system. The following figure illustrates the slip angle \( \alpha \):
At low slip angles (under 4 deg), the lateral force of the tire $F_y$ is proportional to the slip angle $\alpha$. That is:

$$F_y = C_\alpha \alpha \quad (4.8)$$

where $C_\alpha$ is called “cornering stiffness”.

$C_\alpha$ is determined directly from the experiment [44]. The following figure shows the relationship between the lateral force $F_y$ and the slip angle $\alpha$:

![Graph showing the relationship between lateral force and slip angle](image)

**Figure 4.28** Experiment data for $F_y$ Vs $\alpha$
From the figure above, we can estimate the slope of the curve at $\alpha = 0$. The slope is $-709 \text{ lbs/deg.} = -3150 \text{ N/deg} = -181000 \text{ N/rad}$. In FRC48 routine, we can obtain the local longitudinal and lateral velocities of the wheel center. Then calculate the slip angle from the definition. Finally multiply by the slope of the curve obtained from the experiment and get the lateral force of the tire.

Longitudinal force $F_x$ is another force of the tire we have to calculate. Similarly, we must define the slip ratio $S$ first. Slip ratio is defined non-dimensionally, as a percentage of the forward speed, as:

$$\text{Slip ratio } S(\%) = \left(1 - \frac{r*\omega}{V}\right) \times 100$$ (4.9)

where $r$ is the tire effective rolling radius, $\omega$ is the wheel angular velocity, and $V$ is the forward velocity or longitudinal speed of the wheel center in the local wheel coordinate system. The following figure shows the longitudinal slip ratio $S$:

![Diagram showing longitudinal slip ratio](image)

Figure 4.29 Longitudinal slip ratio
In this research, braking capability of the heavy-duty truck is not to be included in the model. Specifically, we applied the cruise control concept to keep the truck at the constant speed. So we assume that the truck experiences low slip ratio during the handling test. From many studies of the relationship of the slip ratio $S$ and the tire longitudinal force $F_x$, we notice that at low slip ratio, the relationship between $S$ and $F_x$ is linear. In other words, it is reasonable to define a “longitudinal stiffness” $C_S$, such that:

$$F_x = C_S \cdot S \quad (4.10)$$

$C_S$ is obtained from the experiment data [5]. The following figure shows the relationship between the longitudinal force $F_x$ and the slip ratio $S$:

![Figure 4.30 Experiment data for $F_x$ Vs $S$](image-url)
From the figure above, we can estimate the slope of the curve at \( S = 0 \). The slope is 61900 \( \text{lb} / \text{unit slip ratio} \) = 275000 \( \text{N} / \text{unit slip ratio} \). In FRC48 routine, we can obtain the local longitudinal velocity of the wheel center. Then calculate the slip ratio from the definition. Finally multiply by the slope of the curve obtained from the experiment and get the longitudinal force of the tire.

Other moments of the tire as overturning moment \( M_x \), and rolling resistance moment \( M_y \), are not significant and can be neglected. However, aligning moment \( M_z \) has certain contribution to the handling analysis and cannot be neglected. Aligning moment (or aligning torque) \( M_z \) is defined as the component of the tire moment vector tending to rotate the tire about the local tire z-axis. Aligning moment is proportional to the lateral force in small slip angle condition. Again, we can assume there is a constant \( KMZ \), such that

\[
M_z = KMZ \cdot F_y \tag{4.11}
\]

In this research, according to the data from the experiment, \( KMZ \) is determined as 0.1 \( \text{ft} \).

Now we have calculated the longitudinal force \( F_x \), lateral force \( F_y \), and aligning moment \( M_z \) of the tire. We can apply the calculated forces to the tires in the model. This is done by using the APFORCE statement, which applies the force and torque to the desired bodies. The tire and the road surface is contacted within a surface called contact
patch. To simplify the calculation, we can assume there is an effective "contact point" between the two bodies. We apply the forces and moments directly at this point.

In the front axle, there is only a single tire at each side. So we can assume this contact point to be on the center of the tire tread. However, the rear two axles have dual tires on each side. The dual tire wheel model is definitely not the same as the single tire wheel model. Theoretically we have to evaluate the geometric position of the contact points of each tire and apply the forces and torques to the two points separately. Yet this is computationally complicated. To make the model more efficient, an alternative dual tire model is illustrated in the following figure:

![Diagram of dual tire model]

**Figure 4.31 Dual tire model**
Wheel geometry and normal force calculations are based on the single point of contact model AB. While the tire tractive force calculations are evaluated at point $P_1$ and $P_2$. This model reduces the complexity of dual wheel calculation by performing one point of contact calculation. However, this model is only valid for the truck running on the perfectly flat road surface.

4.5 Complete Model Description

In this research, the heavy-duty truck model is comprised of seven rigid bodies, three revolute-translational joints, three revolute joints, two distance constraints, and one driver constraint. The system has a total of twelve degrees of freedom. The following table summarizes the detail:

Table 4.1 Multi-body summary of the truck

<table>
<thead>
<tr>
<th>3D ELEMENT TYPE</th>
<th>NUMBER</th>
<th>DOF ADDED</th>
<th>DOF REMOVED</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rigid bodies</td>
<td>7</td>
<td>$7 \times 7 = 49$</td>
<td></td>
</tr>
<tr>
<td>Euler normalizations</td>
<td>7</td>
<td></td>
<td>$7 \times 1 = 7$</td>
</tr>
<tr>
<td>Revolute-translational joint</td>
<td>3</td>
<td></td>
<td>$3 \times 4 = 12$</td>
</tr>
<tr>
<td>Revolute joint</td>
<td>3</td>
<td></td>
<td>$3 \times 5 = 15$</td>
</tr>
<tr>
<td>Distance constraint</td>
<td>2</td>
<td></td>
<td>$2 \times 1 = 2$</td>
</tr>
<tr>
<td>Driver constraint</td>
<td>1</td>
<td></td>
<td>$1 \times 1 = 1$</td>
</tr>
<tr>
<td>Total generalized coordinates</td>
<td></td>
<td>49</td>
<td></td>
</tr>
<tr>
<td>Total constraint number</td>
<td></td>
<td>37</td>
<td></td>
</tr>
<tr>
<td>Total degrees of freedom</td>
<td></td>
<td>$49 - 37 = 12$</td>
<td></td>
</tr>
</tbody>
</table>
The following figure shows the complete heavy-duty truck model:

Figure 4.32 Complete truck model
Initial conditions are very important in the simulation. In DADS analysis program, if no initial conditions are defined, all the velocities in the model will automatically be set to be zero. However, in this research, the truck is supposed to travel at a constant speed of 25 \text{ m/sec} (about 55 \text{ miles/hour}). So it is necessary to set the initial conditions in the model.

After we analyze the model in DADS, the output information file will give us the information about the independent coordinates. The number of the independent coordinates is the number of degrees of freedom. In this model, there are 12 degrees of freedom. And the 12 independent coordinates are:

<table>
<thead>
<tr>
<th>Body element</th>
<th>Coordinate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rear leading axle</td>
<td>E1</td>
</tr>
<tr>
<td>Rear trailing axle</td>
<td>Y</td>
</tr>
<tr>
<td>Rear leading axle</td>
<td>Z</td>
</tr>
<tr>
<td>Pitman arm</td>
<td>Y</td>
</tr>
<tr>
<td>Right front wheel assembly</td>
<td>X</td>
</tr>
<tr>
<td>Right front wheel assembly</td>
<td>Z</td>
</tr>
<tr>
<td>Pitman arm</td>
<td>E1</td>
</tr>
<tr>
<td>rear trailing axle</td>
<td>Z</td>
</tr>
<tr>
<td>rear trailing axle</td>
<td>E1</td>
</tr>
<tr>
<td>left front wheel assembly</td>
<td>Z</td>
</tr>
<tr>
<td>Front axle</td>
<td>E0</td>
</tr>
<tr>
<td>Chassis</td>
<td>X</td>
</tr>
</tbody>
</table>

The time derivative of the x-coordinate of the bodies should be set to the desired speed constant. Another important item is that the speed constant defined in the initial
conditions must be consistent to the speed constant defined in the FRC48 routine where we program the tire forces.

The driver constraint element defines the steering input for the simulation in this research:

![Graph](image)

**Figure 4.33** Driver constraint as steering input
CHAPTER V
SIMULATION RESULTS

The results shown here are for a J turn maneuver to check if the model is running properly. This allows us to look at the transient response of the model. In the simulation, the truck should run straight during the first second. Then there is a ramp steering input during the next second. Thus, the truck begins to change its direction of motion. Certain lateral accelerations are experienced by the truck and this causes the roll motion of the
sprung mass (chassis). The lateral forces are applied at each tire of the truck to offer the centripetal force when the truck is turning.

Thus, for the handling analysis, the roll angle and the lateral acceleration of the chassis are of interest. Steer angles of the left and right front tires can be used to check the steady-state gain. Lateral forces at each tire are also shown. The simulation results are presented in the following figures:

Figure 5.1 is the steer angle of the left front wheel, and Figure 5.2 is the steer angle of the right front wheel. Steer angle is the angle between the heading axis of the tire and the chassis. Because from time equals to 0 to 1 sec, there is no handwheel steering input, both wheel steer angles are zero. From time equals to 1 to 2 sec, handwheel steering angle increases with time, so the steer angles of both left and right front wheels increase corresponding to the steering input. After time equals to 3 sec, the handwheel remains at 82.50 deg, so the steer angles approach the steady-state, the steer angle of the left front wheel is 0.0410 rad = 2.349 deg, and the steer angle of the right front wheel is 0.0408 rad = 2.338 deg. The slight difference between the left and right steer angles is due to the Ackerman Angle geometry. The Ackerman geometry is used to denote the exact geometry of the front wheels. For low-speed turning, the steer angle of the outside wheel $\delta_o$ and the steer angle of the inside wheel $\delta_i$ are approximated by

$$
\delta_o \approx \frac{L}{(R + t/2)} \quad , \quad \delta_i \approx \frac{L}{(R - t/2)}
$$

(5.1)

where $R$ is the turning radius, $t$ is the tread width, and $L$ is the wheelbase.
Figure 5.1  Steer angle of the left front wheel

Figure 5.2  Steer angle of the right front wheel
Figure 5.3 is the roll angle of the chassis of the truck with respect to the ground. Roll angle is defined as the angle of rotation of the sprung mass (chassis) about the vehicle x-axis. From time equals to 1 to 2 sec, the truck undergoes lateral acceleration and the roll angle increases. After 2 sec, the roll angle oscillates and approaches the steady state value $0.11 \text{ rad} = 6.3 \text{ deg}$. The overshoot means the roll angle exceeds its equilibrium position when it oscillates. The truck has more apparent overshoot behavior than passenger cars because of its higher center of gravity and heavier weight. The roll angle per $g$ of lateral acceleration is $6.3/3.25 = 19 \text{ deg}/g$ which is similar to the result obtained from [45].

Figure 5.4 is the lateral acceleration of the chassis of the truck with respect to simulation time. From time equals to 0 to 1 sec, the vehicle travels straight ahead, so there is no lateral acceleration. During the next second, the steering angle increases so the chassis undergoes the lateral acceleration which increases corresponding to the steering input. Then the acceleration approaches the steady-state value, $3.25 \text{ m/sec}^2$. 
Figure 5.3 Roll angle of the chassis of the truck

Figure 5.4 Lateral acceleration of the chassis of the truck
Figure 5.5 Yaw velocity of the chassis of the truck

Figure 5.5 is the yaw velocity (yaw rate) of the chassis of the truck with respect to simulation time. Yaw velocity is the angular velocity of the chassis about the vehicle $z$-axis. At steady-state, it is also equal to the angular velocity with respect to the turning center. If the vehicle forward speed is $V$, turning radius is $R$, yaw velocity or the angular velocity with respect to the turning center is $r$, then

$$ V = R \cdot r $$

(5.2)

On the other hand, the lateral acceleration of the vehicle $a_y$ can be expressed as:
\[ a_y = \frac{V^2}{R} \]  

(5.3)

From Eq. 5.2 and Eq. 5.3, we get

\[ a_y = r \times V \]  

(5.4)

Eq. 5.4 is a very simple yet important method to check if the simulation results are reasonable. If we have yaw velocity \( r \) and the forward speed \( V \), the multiplication of the two must equal the lateral acceleration \( a_y \). From Figure 5.5, \( r \) is 0.13 rad/sec, \( V \) is 25.0 m/sec, and \( 0.13 \times 25.0 = 3.25 \) which is exact the same value of \( a_y \) read from Figure 5.4.

Figure 5.6 through Figure 5.11 are the lateral forces at each equivalent tire of the truck. There is a simple way to check if these forces are appropriate. From Newton’s Second Law, the sum of the lateral forces must equal the vehicle mass times the centripetal acceleration. The summation of the total lateral forces acting on the vehicle is divided by the total mass of the vehicle to get the lateral acceleration: 3.25 m/sec², which is the same value obtained from Figure 5.4.
Figure 5.6 Lateral force of the left front tire

Figure 5.7 Lateral force of the right front tire
Figure 5.8 Lateral force of the left rear leading tire

Figure 5.9 Lateral force of the right rear leading tire
Figure 5.10 Lateral force of the left trailing tire

Figure 5.11 Lateral force of the right rear trailing tire
How does one know if the simulation results are reasonable? Except for the two simple methods used to check the simulation results when discussing Figure 5.4 and Figures 5.6 through 5.11, are there any other methods to check the simulation result? Because the simulation is based on the equations derived from multi-body dynamics which involve very complicated calculations, one cannot tell immediately whether the results are correct or not. However, the steering input in this simulation is a ramp to steady-state input which makes the vehicle perform steady-state cornering. There are some simple cornering equations to check if the simulation results are in the reasonable range.

The steady-state cornering equations are derived from the application of Newton's Second Law. To simplify the analysis, the three-axle tractor can be represented by a two-axle tractor. The wheelbase is defined as the distance from the front axle to the equivalent rear axle. The following figure illustrates the geometry:

![Diagram showing the geometry of a three-axle tractor represented as a two-axle tractor]

Figure 5.12 Equivalent two axle tractor geometry
Because the wheelbase is much smaller than the radius of a turn at high speed, small angles can be assumed. The difference of steer angles between the inside front wheel and ousted front wheel can be neglected. A “bicycle” model is used to represent the vehicle. The following figure [5] shows the bicycle model:

![Diagram of a bicycle model](image)

**Figure 5.13** Cornering of a bicycle model

If the vehicle travels forward at speed $V$, the sum of the lateral forces must equal the mass times the centripetal acceleration.
\[ \sum F_y = F_{yf} + F_{yr} = \frac{MV^2}{R} \quad (5.5) \]

where \( F_{yf}, F_{yr} \) is the lateral force at the front and rear axle, \( M \) is the vehicle mass, \( R \) is the radius of turning, and \( V \) is the forward velocity.

From the moment equilibrium about the center of gravity,

\[ F_y b - F_{yr} c = 0 \quad (5.6) \]

substitute Eq. 5.6 to Eq. 5.5, we get

\[ F_{yr} = \frac{MbV^2}{LR} \quad (5.7) \]

\( Mb/L \) is the portion of the mass carried by the rear axle and is equal to \( W_r/g \), where \( W_r \) is the rear axle load.

Recall the relationship between the lateral force and the slip angle:

\[ F_y = C_a \alpha \quad (5.8) \]

also from Figure 5.13, we find

\[ \delta = 57.3L/R + \alpha_f - \alpha_r \quad (5.9) \]

combine Eq. 5.7, Eq. 5.8, and Eq. 5.9 yields:

\[ \delta = 57.3L/R + \left( \frac{W_f}{C_{af}} - \frac{W_r}{C_{ar}} \right) \frac{V^2}{gR} \]

\[ = 57.3L/R + Ka_y \quad (5.10) \]

where \( \delta \) (deg) is the steer angle at the front wheels, \( L \) (m) is the wheel base, \( W_f \) (N) is the load on the front axle, \( W_r \) (N) is the load on the rear axle, \( C_{af} \) (N/deg) is the
cornering stiffness of the front tires, \( C_{af} \) (N/deg) is the cornering stiffness of the rear tires, and \( K \) (deg/g) is the understeer gradient.

Rearranging Eq. 5.10, we can get the expression for lateral acceleration gain:

\[
\frac{\alpha_y}{\delta} = \frac{V^2}{\frac{57.3Lg}{1 + \frac{KV^2}{57.3Lg}}} \tag{5.11}
\]

Eq. 5.11 is a closed form for the approximation of lateral acceleration gain. Thus, we can approximate the lateral acceleration gain using Eq. 5.11 and compare with the lateral acceleration gain directly from the simulation result. If the difference is reasonably small, we can conclude that the simulation is successful. This method is very useful in examining the model especially when there is few experimental results for comparison.

In this model, understeer gradient is calculated as

\[
K = \left( \frac{W_f}{C_{af}} - \frac{W_r}{C_{ar}} \right) = \frac{(51900-43400)}{3150} = 2.70 > 0 \tag{5.12}
\]

which shows the tractor exhibits understeer characteristics. From Figure 5.11, \( L \) is 4.79 m, \( V \) is 25.0 m/sec, substitute Eq. 5.12 into Eq. 5.11, we get lateral acceleration gain \( \frac{\alpha_y}{\delta} \) is 0.143 (g/deg). If we read \( \alpha_y \) and \( \delta \) from the simulation result in Figure 5.1 and Figure 5.4, the lateral acceleration gain is 0.141 (g/deg), which is very close to the closed form solution. Thus, the simulation result is reasonable.
Another important property we may want to check is the yaw velocity gain. Similarly, the expression for yaw velocity gain can be derived by first considering the yaw velocity $r$

$$r = 57.3 \frac{V}{R}$$  \hspace{1cm} (5.13)

then substitute Eq. 5.13 into Eq. 5.11:

$$\frac{r}{\delta} = \frac{V}{L} \frac{1}{1 + \frac{KV^2}{57.3Lg}}$$  \hspace{1cm} (5.14)

Again the yaw velocity gain calculated from Eq. 5.14 is 3.20 (1/sec) which is close to the yaw velocity gain obtained from figure 5.1 and Figure 5.5, 3.17 (1/sec).

Other features of this model are discussed as follows:

Recall that we model the kinematic motion of the axles as along the z-axis direction of the chassis of the truck. We neglect the motion of axles in the x-axis direction of the chassis. This is reasonable because the motion in the x-axis direction mainly affects the ride analysis, not the handling. Besides, the leaf spring hysteresis behavior is measured in the laboratory by supplying the "vertical force" and measuring the "vertical motion" of the axle relative to the chassis of the truck with the chassis held fixed. The force-deflection diagram and the upper and lower envelopes refer to the vertical force in the z-axis of the chassis. Thus, we can simply examine the vertical motion of the axles.

Next, we study the "roll steer" behavior of the real truck. For the non-steering axles (rear leading and trailing axles), the roll steer behavior cannot be simulated in this
model. Roll steer is defined as the steering motion of the front or rear wheels with respect to the sprung mass that is due to the rolling motion of the sprung mass [24]. The majority of the suspensions are similar to trailing arm suspension systems. As the suspension rolls, one end of the axle will move forward a small distance and the other will move rearward. The result is a steered axle. The following figure shows the roll steer of a non-steering axle:

![Figure 5.14 Roll steer of a non-steering axle](image-url)
For the steering axle (front axle), however, the roll steer motion can be simulated in the model. Before we discuss the roll steer of the front axle, we should study the detailed relative motion of the steering linkages. For an ideal steering system, the drag link is designed such that the arc described by its ball joint connection to the steering arm (included as part of the left front wheel assembly in this model) exactly follow the ideal arc described by the front suspension during jounce-rebound deflections. Figure 5.15 [24] describes the ideal motion of the drag link during the suspension deflection:

![diagram]

Figure 5.15 Ideal motion of drag link during suspension deflection
However, the actual drag link does not usually follow the arc described by the suspension deflection. The "error arc" will act as the steering input during the suspension motions. This phenomenon is called the steering geometry error. There are two major forms, one is the bounce steer, and the other is the roll steer.

Bounce steer is usually expressed as the steer angle of the wheel per unit suspension deflection during the pure vertical motion of the suspension. We can simulate this quantity in the model by holding the chassis fixed and apply the vertical force at both wheels. From FRC48 routine, we can calculate the relationship between the steer angle of the wheel and the deflection of the suspension. The result is as follows:

![Graph showing steer angle vs. deflection](image)

Figure 5.16 Bounce steer characteristics of the steering axle
Similarly, roll steer is usually expressed as the steer angle of the wheel per unit sprung mass (chassis) roll. We can simulate this quantity in the model by holding the chassis fixed and apply the unequal vertical force at both wheels. From FRC48 routine, we can calculate the relationship between the steer angle of the wheel and of the roll angle of the chassis. The result is as follows:

![Graph showing the relationship between roll angle and steer angle](image)

Figure 5.17 Roll steer characteristics of the steering axle

The roll steer coefficient which is defined as the rate of change in roll steer with respect to the roll angle can be calculated. From Figure 5.17, roll steer coefficient $\epsilon = 0.002/0.03 = 0.066 (\text{deg/deg})$, which is in the reasonable range [24].
Actually there is another steering geometry error of heavy-duty trucks: axle windup (brake) steer. When the front brakes are applied, the braking torques are transmitted to the suspension system and rotate the front axle to a forward position. This will again cause an unwanted steering input to the front wheel assembly. In this research, braking maneuvers are not to be studied so we can neglect this effect.

Steer motion can also be induced by the lateral compliances in the suspension. It is called the compliance steer. According to past studies [24], for the non-steering axles with leaf spring suspension, lateral force and aligning moment steer effects are usually very small. For the steering axles, lateral force steer is usually small but aligning moment steer is generally significant and cannot be neglected. Dynamic steering systems describe the detailed compliances of the linkages in the steering system and can calculate the actual handwheel torque input, aligning torque feedback from the roadwheel and the internal reaction forces and moments.
CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

In this research, the model is to be transferred to RTRD (Real Time Recursive Dynamic) formulation, consequently, the model must be very computationally efficient to achieve the goal of "real time". As stated before, stiff compliances as bushing elements are not used because they mainly affect the ride performance in the high frequency
domain and they will decrease the integration step size costing a lot of extra computation time.

In Chapter 4, all the bodies are assumed to be rigid bodies when building the model, and friction forces existing at the bearings, ball joints, and other parts are assumed to be zero. Composite revolute-translational joints and distance constraints are used to avoid using the "massless" intermediate bodies and make the model more efficient.

In Chapter 5, different dynamic responses of the model are presented and analyzed. Several methods including checking the steady-state gain are used to check the correctness of the model. Other features which can be simulated by the model are also described.

From the above, the objectives of the research are fulfilled.

6.2 Recommendations for Future Study

Further study can be accomplished because of the availability of the multi-body dynamics software packages such as DADS used in this research. However, because of the complexity of the programs, the limitation of building the desired model will be the computer hardware. Even though the model in this research was performed on the Silicon Graphics Workstation, a powerful platform frequently used by many other researchers, simulation time is quite long when running multi-body programs as DADS. It may take twice as much computation time if adding just a few bodies to the original model. To achieve the "real time" simulation, complicated models which often generate more
accurate prediction of the real vehicle behavior are usually simplified because of the consideration of computational efficiency.

For future study, upgrading of the computer hardware to be used is very important. With more advanced computation power, the limitations in constructing the model will be minimized. Certain effects which were not modeled before because of the computation time can now be included to get a more accurate response of the vehicle.

As mentioned before, the disadvantage of the multi-body approach simulation is the extensive parameter input. For example, many properties such as the mass moment of inertia must be determined for each body in the model. And usually these properties are obtained by experiment. That means much experimental work and of course cost are involved in the research. Although some of the parameters might be found in the literature, the accuracy of the simulation will be sacrificed because of the inconsistency of the data. Consequently, to get a complete, consistent data for the particular system of interest is very important for future research.

As for the model itself, the future work should include the semitrailer into the model to study the tractor-semitrailer behavior. Dynamic steering system should also be constructed to study the actual driver-road response. The torque feedback from the roadwheel to the driver cannot be simulated by the kinematic steering system model as in this research. The compliance of each component of the steering system, and the friction force existing in the system must be considered to get the actual driver-input and truck-response relationship.
An accurate tire model such as the STI model should be included to study the non-linear effects shown in real tires. Tire contact patch concept should be modeled instead of contact point method used in this research.

Finally, the model can also be extended to study the ride performance by supplying proper suspension kinematic models as mentioned before. Bushing effects can also be modeled if high frequency response is emphasized.


