VEHICLE DYNAMICS OF A HYBRID ELECTRIC FORD EXPLORER: 
A CASE STUDY OF THE 
OHIO STATE UNIVERSITY FUTURETRUCK

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ABSTRACT

Hybrid Electric Vehicles are becoming an increasing presence in the automotive market. The added inertia of the hybrid system components changes, in some cases significantly, the vehicle inertial parameters with respect to a non-hybrid vehicle of the same type. Multi-body dynamics modeling offers a method for consideration of these changes in the design of a hybrid-electric vehicle. Modeling software, such as ADAMS (which is the method used in this research) is widely used in industry as a design tool. The specific case studied in this work compares a stock 2002 Ford Explorer to the Ohio State University’s experimental hybridized 2002 Explorer, the BuckHybrid. Inertial parameters as well as suspension data were acquired for both vehicles. The combination of this data allowed for a model to be built in ADAMS Software, which then provided a virtual test bed for the two vehicles. The ride, handling, and stability of the vehicles were compared throughout a number of computer simulated test maneuvers.
Dedicated to my parents
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CHAPTER 1:  
INTRODUCTION

A. Motivation

Hybrid Electric Vehicles (HEVs) are becoming an increasing presence in the United States (and worldwide) automotive market. The added inertia of the electric motor, batteries and other hybrid system components changes, in some cases significantly, the vehicle inertial parameters with respect to a non-hybrid vehicle of the same type. This change must be considered for ride, handling, stability and safety characteristics of the hybrid vehicle.

Multi-body dynamics modeling offers a method for consideration of these changes in the design of a hybrid-electric vehicle. Modeling software, such as ADAMS (which is the method used in this research) is widely used in industry as a design tool. The case studied in this thesis will utilize ADAMS models of both a stock vehicle, and a hybridized vehicle of the same type. This should give insight into the changes which hybridization causes in vehicle ride, handling and stability, as well as an example of ADAMS usage for application to hybrid electric vehicles.
B. Objective

The objective of this thesis is to present the results of a study based on experimental data acquired for both a stock 2002 Ford Explorer, and the Ohio State University's experimental hybridized 2002 Explorer, the BuckHybrid. Inertial parameters for both vehicles were measured using a VIMF (Vehicle Inertia Measurement Facility) and the changes in the inertial characteristics of the two vehicles will be compared. In addition, suspension data was acquired on a SPMD (Suspension Parameter Measurement Device). The combination of this data allowed for a model to be built in ADAMS Software, which then provided a virtual testbed for the two vehicles. The ride, handling, and stability of the vehicles will be compared throughout a number of computer simulated test maneuvers.

C. Organization of Thesis

Chapter 1 introduces the reader to the thesis topic. It also reviews the prior related research on topics of vehicle dynamics, modeling, and hybrid electric vehicles.

Chapter 2 gives more detailed background information on the topics addressed in this thesis. In particular, vehicle dynamics testing, modeling, and the ADAMS software used are discussed in more detail. In addition, the hybridization of
the BuckHybrid is discussed, its features outlined, and discussion of the FutureTruck competition is included.

Chapter 3 outlines the data acquired for input into the vehicle dynamics models of the stock Explorer and the BuckHybrid. The VIMF and SPMD are discussed in detail, as well as their results and conclusions about the vehicles which can be drawn from them.

Chapter 4 discusses the models and simulation which was done on the stock Explorer and the BuckHybrid. The modeling methods used in ADAMS/CAR are discussed in detail and results from these simulations are also presented.

Chapter 5 summarizes and concludes the thesis, as well as suggests future work.

D. Literature Review

1. Vehicle Dynamics Simulation

The benefit of computer simulation of vehicle dynamics has been documented extensively. Most commonly mentioned is the ability of vehicle dynamics simulation to reduce the development cycle of new vehicles [12, 13, 1, 8]. The ability to reduce the development cycle gives an edge to a company in both cost and in the ability to
quickly keep up with and match trends in the market. Reduction of the development cycle is achieved by allowing for equal performance with fewer iterations of full product prototypes.

Vehicle dynamics simulation has been used in many arenas, from the major automotive manufacturers' use of simulation in development [12], to the suppliers use of simulation to prepare their products [13], to performance based applications [15], as well as heavy-duty vehicle applications [8].

The program used in this study, ADAMS, has been used in the automotive industry as early as 1977 [11]. Companies such as the Ford Motor Company have done extensive development of multi-body dynamics codes to meet their specific development needs [17]. ADAMS implements a number of specific improvements over earlier (and other) multi-body dynamic simulators. In particular, ADAMS allows for the total vehicle model to include full steering and suspension geometry, enabling solution for true kinematic positions and geometries at every time interval. ADAMS also is capable of generating complete time response plots, and providing detailed animated graphics to represent the simulation output [17].

2. Hybrid Electric Vehicle Dynamics

In general, the handling dynamics of a hybrid electric vehicle are no different than any other vehicle. The work specifically focused on hybrid vehicle dynamics,
such as Toyota’s work on their hybrid Estima minivan [16], is concerned with using the electrical power available due to the hybrid nature of the vehicle to improve vehicle dynamic characteristics.

From a handling dynamics perspective the important characteristic of a hybrid electric vehicle is the different weight distribution caused by the addition of the electrical components. Work has been done on the affect of vehicle loading on the dynamics of the vehicle [10], however none of this has focused on hybrid-electric vehicles.
A. Classical Multi-Body Dynamics

1. Vectorial and Analytical Dynamics

Parts constrained by joints, gears, bearings, or other couplings make up a multi-body system. In order to model this system, it is necessary to model not only the parts, but also the constraints between them.

There are two fundamental ways to approach solving a dynamic problem: vectorial dynamics and analytical dynamics [21]. Vectorial dynamics is based on free-body diagrams, forces, momentum and accelerations, whereas analytical dynamics is based on kinetic energy, potential energy, and virtual work. In addition, although vectorial dynamics must consider individual bodies and the forces acting on them separately, analytical dynamics can consider the system as a whole [22]. The greatest advantage to using analytical dynamics instead of vectorial dynamics is that, because of the whole system approach of analytical dynamics, it is not necessary to consider constraint forces when developing the equations of motions.
2. LaGrange’s Equation

There are many formulations of LaGrange’s Equation, but shown below is one of the most general. This is the basic formulation for a system with n generalized coordinates.

\[
\frac{d}{dt} \left( \frac{\delta T}{\delta q_k} \right) - \frac{\delta T}{\delta q_k} \frac{\delta V}{\delta q_k} = Q_k^{(nc)} \quad k = 1, 2, \ldots, n
\]

Where:

- \( T \) = Total Kinetic Energy of the System
- \( V \) = Potential Energy of the System
- \( q_k \) = Generalized Coordinates (1 through \( k \))
- \( Q_k \) = Generalized Forces in the direction of each generalized coordinate
- \( Q_k^{(nc)} \) = Non-Conservative Generalized Forces for each coordinate

(Conservative Forces are included in the potential energy of the system)

In the above formulation of LaGrange’s equations all constraints have been used to reduce the number of generalized coordinates, and thus none are left in the formulation of the equations. Although the above formulation is useful because it is not necessary to consider constraint forces when using an analytical approach, oftentimes constraints are considered in order to calculate reaction forces. In this case constraints are not used to decrease the number of required generalized coordinates, instead they are left alone, and the system of equations is set up for what
are called constrained generalized coordinates. Below is the formulation for this case.

\[ \frac{d}{dt} \left( \frac{\delta T}{\delta \dot{q}_k} \right) - \frac{\delta T}{\delta q_k} = Q_k + \sum_{i=1}^{m} \lambda_i a_{ik} \quad k = 1, 2, \ldots, n \]

Where:

- \( T \) = Total Kinetic Energy of the System
- \( q_k \) = Generalized Coordinates (1 through \( k \))
- \( Q_k \) = Generalized Forces in the direction of each generalized coordinate
- \( \lambda_i \) = LaGrange Multipliers (1 through \( m \))
- \( a_{ik} \) = Coefficients of the constraint equations

Also, there are cases where the constraint equations cannot be used to reduce the number of generalized coordinates. Note that in this case the number of generalized coordinates of a system may not be equal to the number of degrees of freedom. This is due to the possibility of non-holonomic constraints. A non-holonomic constraint does reduce the number of degrees of freedom because it constrains the motion of the particle or body. However, it does not allow one of the generalized coordinates to be solved for and therefore it does not reduce the number of coordinates that are required to fully describe the system.
A holonomic constraint is any constraint which can be expressed in the form
\[ f(q_1, q_2, ..., q_k, t) = 0 \]

A non-holonomic constraint has the form:
\[ a_{q_1} dq_1 + a_{q_2} dq_2 + ... + a_{q_k} dq_k = 0 \]

Where \( a_{q_i} \) are functions of the generalized coordinates and or their velocities, and the equation cannot be integrated to be put in the form of a holonomic constraint. Non-holonomic constraints are generally constraints on velocities, such as a non-slip constraint.

B. Vehicle Dynamics

Vehicle Dynamics is the study of dynamics applied to the motions of a vehicle. These motions are split into three main groups: longitudinal, ride, and handling. Longitudinal motions are acceleration and braking. Ride is vertical motions and pitch. Handling is lateral motions, yaw and roll. The forces which cause these motions are results primarily of the tire-road interaction, but also may be caused by aerodynamics and gravity. The motions of a vehicle can be characterized according the classical multi-body dynamics method explained in IIA.

As in any dynamics problem, it is critical to clearly define the coordinate system. The most widely used coordinate system in vehicle dynamics is the SAE
coordinate system, which is shown below in Figure 1. Of course, it may be of value to define local part-oriented coordinate systems and their relative transformation, but results, and full-vehicle information is generally presented using the SAE coordinate system.

Figure 1: SAE Vehicle Axis System

C. Tire Modeling

According to Gillespie [1] the tire serves essentially three basic functions

1. It supports the vertical load, while cushioning against road shocks
2. It develops longitudinal forces for acceleration and braking

3. It develops lateral forces for cornering.

Many tire models have been developed to approximate the tire's performance of these functions. All of them are only an approximation to the very complex system that is a tire.

1. Introduction to the Magic Formula Tyre Model

   The tire model used in this work is the Magic Formula Tyre Model (MF-Tyre). This model was developed initially in the mid-eighties through a cooperation between Volvo Car Corporation and the Delft University of Technology [23]. The Magic Formula Tyre model uses a set of mathematical equations to take the inputs of longitudinal slip, lateral slip, camber angle and vertical force of the wheel to calculate the forces and moments acting on the tire. This input/output relationship can be seen below in Figure 2. Definitions of these variables may be seen in Table 1.
Table 1 specifies the major variables which are used in the MF-Tyre calculations. There are many other local-type variables which are not listed below.

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Name</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$</td>
<td>Slip Angle</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>Camber Angle</td>
</tr>
<tr>
<td>$F_z$</td>
<td>Longitudinal Force</td>
</tr>
<tr>
<td>$F_y$</td>
<td>Lateral Force</td>
</tr>
<tr>
<td>$F_z$</td>
<td>Vertical Load</td>
</tr>
<tr>
<td>$M_x$</td>
<td>Overturning Moment</td>
</tr>
<tr>
<td>$M_y$</td>
<td>Rolling Resistance Moment</td>
</tr>
<tr>
<td>$M_z$</td>
<td>Self Aligning Moment</td>
</tr>
<tr>
<td>-------</td>
<td>----------------------</td>
</tr>
<tr>
<td>$V_x$</td>
<td>Longitudinal Speed</td>
</tr>
<tr>
<td>$V_y$</td>
<td>Lateral Speed</td>
</tr>
<tr>
<td>$V_{sx}$</td>
<td>Longitudinal Slip Speed</td>
</tr>
<tr>
<td>$V_{sy}$</td>
<td>Lateral Slip Speed</td>
</tr>
<tr>
<td>$\Omega$</td>
<td>Tire Rolling Speed</td>
</tr>
<tr>
<td>$F_{zo}$</td>
<td>Nominal (Rated) Load</td>
</tr>
<tr>
<td>$B$</td>
<td>Low Load Stiffness Effective Rolling Radius</td>
</tr>
<tr>
<td>$D$</td>
<td>Peak Value of Effective Rolling Radius</td>
</tr>
<tr>
<td>$F$</td>
<td>High Load Stiffness Effective Rolling Radius</td>
</tr>
<tr>
<td>$R_0$</td>
<td>Unloaded Tire Radius</td>
</tr>
<tr>
<td>$m_{belt}$</td>
<td>Tire Belt Mass</td>
</tr>
<tr>
<td>$C_z$</td>
<td>Tire Vertical Stiffness</td>
</tr>
<tr>
<td>$\rho / \rho$</td>
<td>Tire Deflection / Tire Deflection Velocity</td>
</tr>
<tr>
<td>$K_z$</td>
<td>Tire Vertical Damping</td>
</tr>
</tbody>
</table>

**Table 1: Variables Used in MF-Tyre**

The “Magic Formula” itself is simply a complex group of mathematical formulas which empirically represent the tire behavior and convert the inputs to outputs. The magic formula uses trigonometric functions to match the measured curves. The basic structure of a “Magic Formula” is always the same. There is a sine version of the magic formula which is:

$$Y(x) = D \sin[C \arctan\{Bx - E(Bx - \arctan(Bx))\}]$$

As well as a cosine version of the magic formula which is:
\[ Y(x) = D \cos[C \arctan{Bx - E(Bx - \arctan(Bx))}] \]

The sine version of the magic formula is used to calculate the longitudinal and lateral forces on the tire, and the cosine version of the magic formula is used to calculate the self-aligning moment. These formulas use dimensionless parameters to match the curves, which allows for scaling.

![Diagram of Tydex C- and W-Axis Systems](image)

**Figure 3: Tydex C- and W-Axis Systems used by MF-Tyre [23]**

There are two axis systems that are used by the Magic Formula Tyre Model: The TYDEC C-Axis, and W-Axis systems. These coordinate systems can be seen in Figure 3 above. The primary difference between these two coordinate systems is the location of the point of origin. For the C-Axis system, the origin is located at the center of the wheel; whereas, for the W-Axis system, the origin is located at the road.
contact point. This road contact point is defined by the intersection of the wheel plane and the road tangent plane. Otherwise, the orientation of the coordinates is the same for both systems. It should also be noted that the forces and torques which are calculated by the Magic Formula Tyre model are then projected in the W-axis system for use in the rest of the dynamic calculations.

2. Normal Load Calculations

The first calculation done in the tire model is the calculation of the normal load on the tire. This is critical because it is an input to the rest of the model. There are two main assumptions made in this calculation:

1. The radius of curvature of the road is considered to be much greater than the radius of the tires. This allows the road to be approximated by its tangent plane, which is a valid assumption as long as the road radius of curvature is not smaller than approximately 2 meters.
2. The tire has only one contact point (C) with the road. [23]

After these assumptions have been implemented the normal load \( F_z \) of the tire can be calculated by:

\[
F_z = C_z \rho + K_z \dot{\rho}
\]

In addition the MF-Tyre model uses a normalized vertical load increment \( df_z \) which is defined as:
2. Effective Rolling Radius Calculations

The effective rolling radius of the tire is given by:

\[ R_e = \frac{V_x}{\Omega} \]

For radial tires there is a dependency in this equation on the tire normal load. This dependency is small when the load is close to the rated load, however, when the load is high compared to the normal load, this, as expected, decreases the effective rolling radius of the tire. If constant tire stiffness is assumed, this can be represented by:

\[ \rho = \frac{F_z}{C_z} \]

In the Magic Formula Model, an estimation of the effective rolling radius is made using a “Magic Formula approach” [94]. This is given by:

\[ R_e = R_0 - \rho_{F,0} \left( D \arctan(B \rho^d) + F \rho^d \right) \]

Where:

\[ \rho_{F,0} = \frac{F_{z0}}{C_z} = \text{Nominal tire deflection;} \]

\[ \rho^d = \frac{\rho}{\rho_{F,0}} = \text{Dimensionless radial tire deflection;} \]
\( D = \) Coefficient which defines the shift from the asymptote at high wheel loads.

\( B = \) Coefficient which stretches the ordinate of the arctangent function. A larger value of \( B \) is used if the slope around \( F_z = 0 \) is large.

\( F = \) Coefficient that defines the ratio between tire radial deformation and effective tire deformation. The stiffer the tire, the lower the value of \( F \).

A graphical representation of these quantities can be seen below in Figure 4.

**Figure 4: Effective Rolling Radius and Longitudinal Slip of an MF-Tyre [23]**

3. Calculation of Tire Slip Quantities

Tire slip quantities are calculated using complicated and very scalable equations which are based on the basic Magic Formula equations given above. The details of these equations can be seen in the ADAMS/TIRE Handbook [23].
D. Real-World Vehicle Testing for Handling and Stability

The purpose of any vehicle dynamics test is to characterize the ride and handling of the vehicle, as well as the performance of the vehicle. During actual real-world testing data is acquired through use of sensors, accelerometers, load cells, etc. These measurements are all subject to noise and error and are therefore generally filtered to achieve better results. Some common measurements are [8]:

1. Steering Wheel Angle
2. Steering Wheel Torque
3. 3-axis linear accelerations: x - longitudinal, y - lateral, and z - vertical
4. 3-axis rotational velocities: Roll - about the x-axis, Pitch - about the y-axis, and Yaw - about the z-axis.
5. Vehicle velocity or speed in the plane of motion
6. Sideslip angle: Angle between the tire’s x-axis and the vehicle velocity vector
7. Motion (or position) of each corner of the vehicle relative to the ground or wheel axis in the z-direction
8. Brake pedal force
9. Brake pedal travel
10. Additional acceleration measurements of seats, floor pan, steering column, etc, for NVH and ride studies.

These data, after going through filtering processes can be used to quantify some vehicle handling characteristics.

Vehicle dynamics testing falls into two main categories, open-loop and closed-loop testing. Closed-loop testing involves the driver in the system, and really is a
measure of the behavior of the vehicle-driver combination. Open-loop testing, on the other hand only involves the vehicle’s response to a given input, and does not close the loop by including driver feedback.

1. Constant-Radius Cornering

A constant radius cornering test is performed to learn the vehicle’s understeer and oversteer characteristics. In a constant radius cornering analysis the turn radius is held constant and the vehicle velocity is varied to produce increasing amounts of lateral acceleration. In order to determine the understeer/oversteer properties of the vehicle, the steer angle at each vehicle speed is measured. Then a plot of steer angle versus lateral acceleration can be made. The intercept of the curve is the Ackerman Angle. The slope of the curve is the understeer gradient. If the slope is positive, the vehicle understeers; if the slope is zero, the vehicle is at neutral steer; and if the slope is negative, the vehicle oversteers.

2. Constant-Velocity Cornering

A constant-velocity cornering test is also performed to learn the vehicle’s understeer and oversteer characteristics. In this test the vehicle velocity is held constant and the turn radius is varied to produce increasing amounts of lateral acceleration. As in a constant-radius turn, the steer angle is plotted versus the lateral
acceleration of the vehicle. In this case, the Ackerman Angle appears in the plot as a straight line, with a given slope. If the slope of the vehicle’s actual curve is greater than that of the Ackerman curve, the vehicle understeers. If the slope is equal to the Ackerman slope, then the vehicle is at neutral steer. And, if the slope is less than the slope of the Ackerman curve, the vehicle oversteers.

3. Single Lane Change

The single lane change maneuver is completed by giving the steering a complete sinusoidal-like cycle as an input over a specified amount of time.

4. Fishhook

The fishhook maneuver is used primarily to evaluate dynamic roll-over vehicle stability. The maneuver consists of a specified steering input – a certain angle over a period of time – in one direction; followed by a specified steering input in the other direction. Generally, the test is begun at constant speed. Then the vehicle is put into neutral gear and the turns are begun. [25]

5. Step Steer

A step steer analysis consists of increasing the steering input from one value to another over a given period of time. Step steer is primarily used to determine time-
domain transient response of the vehicle. [25]. Generally it is important to measure: steering angle, yaw rate, vehicle speed, and lateral acceleration.

E. Vehicle Dynamics Simulation

1. Advantages and Disadvantages of Vehicle Dynamics Simulation

Utilizing technology to model vehicles provides a number of benefits. Simulation results are relatively quick, low cost, and low risk; no actual vehicle components are required, no physical tests must be done. Currently, with mounting pressure to shorten the design cycle of new vehicles, the use of even simple dynamic models in initial stages of vehicle development is invaluable, as it can catch, with relative ease, any large design flaws. In addition, multi-body codes, such as ADAMS, make the task of vehicle dynamics modeling more efficient by providing the base code for the simulations. However, computer simulations are still error-prone, as well as dependent on input of accurate vehicle parameters. In addition, simulations cannot give any of the subjective feedback which test drives can give. The test engineers who evaluate model vehicles can provide feedback which cannot be duplicated in a quantitative computer code. Also, simulations may not be sensitive enough to detect small design changes, especially if developed for a quick result.
Ultimately, computer simulations have a definite place in the vehicle design process. Although they are incapable of providing the fine, subjective information which only a test driver can provide, they are capable of more readily showing cause and effect in suspension design. Computer simulations are also valuable for studying trends in vehicle response caused by other vehicle modifications, as is the case in this research.

2. About ADAMS

Automatic Dynamic Analysis of Mechanical Systems (ADAMS) is a powerful, and commonly used multi-body dynamics simulator. It is particularly useful for this application due to a module called ADAMS/CAR. This module is made specifically for modeling of suspension and full vehicles, and has a number of built in templates for building the vehicle model as well as many built in dynamic tests specific to vehicle modeling.
Figure 5: ADAMS/CAR Vehicle Axis System

The vehicle axis system defined by ADAMS/CAR is the axis system which will be used for the remainder of this thesis unless otherwise specified.

3. ADAMS/CAR File Structure

ADAMS/CAR uses a unique file structure which allows for great flexibility and functionality. The file structure is shown in Figure 6. The great strength of ADAMS/CAR’s file structure is its use of templates. Templates allow the user to define a system’s topology, without specifying all of the actual locations and
dimensions of a part. For example, one can produce a double-wishbone suspension template without any exact measurements of control arm dimensions. This allows for one template to be used for many different applications. Each specific application needs only define the actual dimensions and locations of parts in a data file, and create a subsystem using this data file and the template. Between the topologic data contained in the template, and the specific data contained in the data file, a subsystem can be created which fully represents a part or set of parts. Then, subsystems can be combined to make assemblies. The subsystems talk to each other using “communicators” which send data from one subsystem to another in an assembly.

Figure 6: ADAMS File Structure
4. Tire Modeling in ADAMS/CAR

Tire modeling in ADAMS/CAR is done using a set of shared object libraries called ADAMS/TIRE. When an analysis is performed, the solver in ADAMS invokes ADAMS/TIRE to calculate the forces and torques the tire produces. The basic structure of ADAMS/TIRE within the solver can be seen in Figure 7.

![Figure 7: ADAMS/TIRE Structure](image-url)
The tire property file defines the specific properties of the tire to be modeled, as well as specifics about which tire model should be used for calculations. The road property file specifies the inputs from the road. And, any information needed from the full-vehicle model can also be obtained by ADAMS/TIRE.

For the average user, ADAMS/TIRE is practically transparent. The tire property file must simply be modified to represent their tire, and then referenced by the full-vehicle model. All of the rest is done in the background by the solver routine.

5. ADAMS/CAR Full-Vehicle Models

ADAMS/CAR full-vehicle models consist of a number of subsystems linked together to form full-vehicle assembly. The required subsystems for a complete full-vehicle assembly are: body, front and rear suspension, front and rear tires, steering system, and a test rig. Other subsystems may also be included. This can be seen in Figure 8 below. Subsystems surrounded by dashed lines are optional subsystems. It should be noted that additional subsystems may also be needed to fulfill specific user needs. For example, it is possible to add an extra suspension and wheel subsystem to describe a multi-axle truck. Once the subsystems are linked together the full-vehicle model may then be tested as a normal vehicle would be by using the simulated tests in ADAMS.
Figure 8: Full Vehicle Assembly Components

6. ADAMS/CAR Full Vehicle Simulations

Once a full-vehicle assembly has been created in ADAMS/CAR, either by using standard inputs, or using conceptual suspension modeling, any number of simulations can then be done. In addition to the ability to create your own test maneuver, ADAMS/CAR has a number of built-in full-vehicle dynamic analyses. For the purpose of handling and stability evaluation, the open-loop steering analyses,
and cornering analyses are of the most interest. Included among the open-loop analyses are: drift, fishhook, impulse steer, ramp steer, single lane change analysis, step steer, and swept sine steer. The cornering analyses include: braking in a turn, constant radius cornering, cornering with steer release, lift-off turn-in, and power-off during cornering. In addition there are quasi-static constant velocity and constant radius cornering analyses to determine vehicle under/over steer characteristics.

7. Post-Processor

ADAMS also has a thorough post processor which allows the user to watch, record, and play back animations of the simulation, as well as plot specific data. Data is acquired through requests, and any requested data can be analyzed using ADAMS post processor. The post-processor is particularly useful because it’s animations allow the user to view the vehicle as it goes through a test. This can provide an overall “feel” for the vehicle’s response in the test, in addition to the numerical data. Also, numerical data can be easily plotted in post-processor.
F. The *BuckHybrid* - Vehicle Testbed

1. FutureTruck Competition

FutureTruck competition is a unique college engineering competition in which teams from 15 top engineering schools from across the United States and Canada are challenged to re-engineer a stock Ford Explorer into a more energy efficient and environmentally friendly sport utility vehicle (SUV). The teams are required to meet these goals while maintaining the performance, utility and safety of the stock vehicle. Each team was given a stock 2002 Ford Explorer, and seed money, and then set free to meet the challenge.

Ford Motor Company and the Department of Energy were the major competition sponsors for 2002 and 2003. In 2002 competition was held at Ford’s Desert Proving Grounds in Yucca, AZ, and in 2003 at Ford’s Michigan Proving Grounds in Romeo, MI. The 2004 Competition will also be at Ford’s Michigan Proving Grounds in Romeo, MI. Competition involves on-road testing for acceleration, fuel economy, handling, braking, off-road capability and towing capability, chassis dynamometer testing for emissions, as well as static design and consumer acceptability events.

In the 2002 FutureTruck Competition 13 of the 15 schools were either gasoline or diesel electric hybrids (the remaining two teams used fuel cell technology).
2. BuckHybrid Design Details

The Ohio State University FutureTruck team received their stock 2002 Ford Explorer in November 2001, and prepared it for competition in June 2002. For the 2002 competition, the OSU team chose a diesel-electric, post-transmission, charge-sustaining hybrid strategy. This strategy was chosen for a number of reasons. First of all, diesel engines are inherently more efficient than gasoline engines. Unfortunately, after-treatment of diesel engines is a challenge, but the Ohio State team has the ability to implement advanced after-treatment to meet this challenge. The post-transmission strategy was chosen for implementation and packaging reasons. A 55kW peak electric motor was available and had been used with success in previous years. Due to the size and power of this motor, it was appropriate for a post-transmission coupling. A charge-sustaining strategy was chosen for consumer acceptability reasons. The average US consumer wants higher fuel economy, but is not ready for the added complication of a charge-depleting, “plug-in” hybrid, which requires the consumer to charge the vehicle from the electrical grid.

In order to implement the strategy described above, most of the major powertrain components were removed and replaced with new parts. The stock engine was replaced with a 2.5 L VM Motori/Detroit Diesel Corporation advanced CID (Compression Ignition Direct Injection) engine. The stock transmission was
replaced with the Ford R4 5 speed manual transmission. In addition, an entire electric motor system was added to the vehicle. The electrical system is composed of a 55 kW peak Ecostar AC Induction motor was coupled, post-transmission, to the driveshaft. This motor was driven by an Ecostar inverter, and 27 Hawker Genesis batteries, each supplying 12 volts, and 16 Amp-hours.

Packaging was one of the most difficult tasks for the team, as room for additional components was scarce. Shown below in Figure 9 is the underbody packaging diagram for the BuckHybrid. All empty space under the vehicle was filled once hybridization was complete.

![BuckHybrid Packaging Diagram](image)

**Figure 9: BuckHybrid Packaging Diagram**
The electrical systems account most of the weight and weight distribution differences between the stock Explorer and the BuckHybrid. A summary of the major components which affected the weight and inertias of the BuckHybrid can be seen below in Table 2.

<table>
<thead>
<tr>
<th>Component</th>
<th>Stock Explorer</th>
<th>BuckHybrid</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine</td>
<td>VM Motor, 2.5 L CIDI</td>
<td>220 kg</td>
</tr>
<tr>
<td>Electric Motor</td>
<td>--</td>
<td>Ecostar, 55 kW peak, AC Induction</td>
</tr>
<tr>
<td>Electric Motor Coupling</td>
<td>--</td>
<td>Custom Built</td>
</tr>
<tr>
<td>Inverter</td>
<td>--</td>
<td>Ecostar, High Speed IGBT</td>
</tr>
<tr>
<td>Batteries &amp; Battery Boxes</td>
<td>--</td>
<td>Hawker Genesis, 12V, 16Ah, Lead Acid</td>
</tr>
<tr>
<td>Transmission</td>
<td>Ford 4 speed automatic</td>
<td>Ford M50D-R4, 5 speed manual</td>
</tr>
<tr>
<td>Spare Tire</td>
<td>Full-sized spare tire</td>
<td>Replaced with run-flat tire inserts</td>
</tr>
<tr>
<td>Interior Modifications</td>
<td>--</td>
<td>Replaced seat foam and carpeting with lightweight alternatives</td>
</tr>
</tbody>
</table>

Table 2: Major Component Weight Comparison
A. Vehicle Inertia Measurement Facility (VIMF)

Shortly after the 2002 FutureTruck competition, the BuckHybrid was taken for testing to identify its inertial parameters. This testing was done at the state of the art Vehicle Inertia Measurement Facility (VIMF) at SEA, Limited. The VIMF measures all inertial parameters by a single test device. This device is a stable pendulum for center of gravity height and pitch moment of inertia tests. The vehicle configuration is shown below in Figure 10.

![BuckHybrid during CG height and Pitch Measurements](image)

Figure 10: The BuckHybrid during CG height and Pitch Measurements
During the roll and yaw moment of inertia tests, the VIMF is an inverted pendulum. The roll and yaw bearings are contained in a “roll/yaw pivot assembly”. This assembly is placed under the vehicle support platform, providing this configuration, which can be seen in Figure 11. For roll inertia tests, the yaw bearing is locked. During the yaw inertia tests, the roll motion is stopped by a rigid load cell. This prevents roll motion, while measuring roll torque during yaw oscillations. This roll torque measurement is used to calculate the roll/yaw product of inertia.

The results from the VIMF can be seen below in Table 3.
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Stock 2002 Explorer</th>
<th>The BuckHybrid</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total Weight (kg)</td>
<td>2016.7</td>
<td>2536.1</td>
</tr>
<tr>
<td>CG Height (mm)</td>
<td>688.3</td>
<td>648.6</td>
</tr>
<tr>
<td>Lateral CG (mm)</td>
<td>-37</td>
<td>-12</td>
</tr>
<tr>
<td>Longitudinal CG (mm)</td>
<td>1305</td>
<td>1461</td>
</tr>
<tr>
<td>Pitch Inertia (kg•m²)</td>
<td>3538</td>
<td>4513</td>
</tr>
<tr>
<td>Yaw Inertia (kg•m²)</td>
<td>3638</td>
<td>4722</td>
</tr>
<tr>
<td>Roll Inertia (kg•m²)</td>
<td>673</td>
<td>924</td>
</tr>
<tr>
<td>Roll/Yaw Product (kg•m²)</td>
<td>33</td>
<td>7</td>
</tr>
</tbody>
</table>

**Table 3: Comparison of Inertial Parameters**

From the table above (Table 3), it is apparent that the hybridization of the Explorer caused significant changes to the weight and inertial parameters of the vehicle. The total weight was increased by approximately 500 kilograms. The location of the center of gravity was shifted in all directions, and roll, yaw, and pitch inertias were all increased. However, the roll-yaw product of inertia was decreased.

The measurement error can be seen below in Table 4. The research of Chrstos et al. and Durisek [19, 20], speaks of the full details of error-analysis and measurement repeatability of the VIMF.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>± 18.0 N</th>
</tr>
</thead>
<tbody>
<tr>
<td>CG Height</td>
<td>0.5% of Measured Value</td>
</tr>
<tr>
<td>Pitch Inertia</td>
<td>1.0% of Measured Value</td>
</tr>
<tr>
<td>Roll Inertia</td>
<td>3.0% of Measured Value</td>
</tr>
<tr>
<td>Yaw Inertia</td>
<td>1.0% of Measured Value</td>
</tr>
<tr>
<td>Roll/Yaw Inertia</td>
<td>± 6.8 kg-m²</td>
</tr>
</tbody>
</table>

**Table 4: Vehicle Inertia Measurement Facility Measurement Repeatability**
B. Suspension Parameter Measurement Device (SPMD)

In order to validate the full-vehicle model, suspension kinematics and compliances were measured using the Suspension Parameter Measurement Device (SPMD), whose use was graciously donated by SEA, Limited.

Figure 12: The *BuckHybrid* during Roll Testing
The SPMD allowed collection of all the characteristic curves of the BuckHybrid suspension. A total of eight tests were done on the BuckHybrid: Front and Rear Bounce, Front and Rear Roll, Front and Rear Lateral Compliance, and Front and Rear Steering Compliance.

The Bounce and Roll tests were both done by affixing steel cross-members to the frame of the BuckHybrid. In the bounce test these cross-members were moved up and down equally on both sides of the vehicle to drive the vehicle through its bounce motion. In the roll test the cross-members were moved up on one side of the vehicle.
and down on the other to move the vehicle through its roll motion. In order to eliminate forces and aligning moments at the tires, the tires were placed on free-floating pads. In addition, the steering wheel was locked in place, and the brake and parking brake were applied. Measurements were made using eleven string potentiometers, six linear displacement transducers, four vertical load scales, an inclinometer, and a load cell [18].

For the lateral compliance test a force was applied to the right side wheel pads to move them to the left. For the steering compliance test an aligning moment was applied to the right side wheel pads. All of the forces were measured using load cells, and data from the above mentioned instrumentation was also used as needed to characterize suspension response to the tests.
A. Model Development of the *BuckHybrid*

In ADAMS, there are seven required vehicle sub-systems for a full-vehicle model. These required sub-systems are the body, front and rear suspension, front and rear tires, steering system, and a test rig. Other subsystems such as powertrain and brakes are possible additions, but were left out in this model as they did not directly apply to the results. Almost all full-vehicle simulations can be done without a powertrain or brakes. The full-vehicle models of the *BuckHybrid* and of the Stock Explorer were compiled from these basic sub-systems, which are described in more detail below. A schematic of the full-vehicle assembly used for these models can be seen in Figure 14.
Figure 14: Full Vehicle Assembly Layout of the *BuckHybrid*/Stock Explorer
1. Front Suspension Model Development

a. Front Suspension Subsystem

The front suspension on the BuckHybrid is simply the stock 2002 Ford Explorer suspension. This is a short and long arm, coil-over-shock type suspension. A stock ADAMS template for a double-wishbone suspension was used as a basis for the suspension model. This template includes the entire basic layout of this type of suspension, including hardpoint defined geometry as well as joints. This template was modified to reflect the actual suspension by modifying the hardpoint locations, and inputting actual spring and damper data. The geometry points (hard points) were initially measured simply using tape measures. Locations of each of the hardpoints (shown on an ADAMS view of the suspension for clarity) can be seen below in Figure 15. The specific location of each of these hard points was modified within the ADAMS subsystem to match the initial measurements from the vehicle. The values for the front suspension geometry are shown below in Table 6.
The hardpoints located by number in Figure 15 are defined by description and name below in Table 5 and their actual values can be seen in Table 6.
<table>
<thead>
<tr>
<th>Number</th>
<th>Hardpoint Name</th>
<th>Hardpoint Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>lca_front</td>
<td>These three points describe the lower control arm.</td>
</tr>
<tr>
<td>2</td>
<td>lca_outer</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>lca_rear</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>lwr_strut_mount</td>
<td>This point is the bottom of the strut.</td>
</tr>
<tr>
<td>5</td>
<td>tierod_inner</td>
<td>These two points define the tierod location.</td>
</tr>
<tr>
<td>6</td>
<td>tierod_outer</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>top_mount</td>
<td>This defines the mount position of the top of the strut.</td>
</tr>
<tr>
<td>8</td>
<td>uca_front</td>
<td>These three points locate the upper control arm.</td>
</tr>
<tr>
<td>9</td>
<td>uca_outer</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>uca_rear</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>wheel_center</td>
<td>This point defines the center of the wheel.</td>
</tr>
</tbody>
</table>

Table 5: Hardpoint Descriptions - Front Suspension

<table>
<thead>
<tr>
<th>Name</th>
<th>X (in)</th>
<th>Y (in)</th>
<th>Z (in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>lca_front</td>
<td>-5</td>
<td>-15.75</td>
<td>2</td>
</tr>
<tr>
<td>lca_outer</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>lca_rear</td>
<td>15.5</td>
<td>-15</td>
<td>1.5</td>
</tr>
<tr>
<td>lwr_strut_mount</td>
<td>-2.5</td>
<td>-5</td>
<td>1.5</td>
</tr>
<tr>
<td>tierod_inner</td>
<td>-8.25</td>
<td>-12</td>
<td>6.25</td>
</tr>
<tr>
<td>tierod_outer</td>
<td>-6.5</td>
<td>1.25</td>
<td>5.25</td>
</tr>
<tr>
<td>top_mount</td>
<td>-1</td>
<td>-10</td>
<td>21.5</td>
</tr>
<tr>
<td>uca_front</td>
<td>-8.25</td>
<td>-11.5</td>
<td>20</td>
</tr>
<tr>
<td>uca_outer</td>
<td>-1.5</td>
<td>-5.25</td>
<td>18.5</td>
</tr>
<tr>
<td>uca_rear</td>
<td>7.5</td>
<td>-11.5</td>
<td>18</td>
</tr>
<tr>
<td>wheel_center</td>
<td>0</td>
<td>3.5</td>
<td>7</td>
</tr>
</tbody>
</table>

Table 6: Measured Front Suspension Geometry Points
The suspension hardpoints shown in Table 7 are the actual measurements taken on the vehicle. As previously mentioned, the ADAMS coordinate system is slightly different from the usual vehicle axis system. ADAMS coordinate system can be referenced in Figure 5. ADAMS uses x positive towards the front of the vehicle, y positive towards the passenger side, and z positive up. Therefore, adjustments needed to be made to the measurements in order to be input into ADAMS. In addition, the decision was made to use metric measurements in ADAMS, so a unit conversion was also performed. Shown below in Table 7 are these adjusted points.

<table>
<thead>
<tr>
<th>Name</th>
<th>X (mm)</th>
<th>Y (mm)</th>
<th>Z (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>lca_front</td>
<td>-1568</td>
<td>305</td>
<td>51</td>
</tr>
<tr>
<td>lca_outer</td>
<td>-1441</td>
<td>705</td>
<td>0</td>
</tr>
<tr>
<td>lca_rear</td>
<td>-1048</td>
<td>324</td>
<td>38</td>
</tr>
<tr>
<td>lwr_strut_mount</td>
<td>-1505</td>
<td>578</td>
<td>38</td>
</tr>
<tr>
<td>tierod_inner</td>
<td>-1651</td>
<td>400</td>
<td>159</td>
</tr>
<tr>
<td>tierod_outer</td>
<td>-1607</td>
<td>737</td>
<td>133</td>
</tr>
<tr>
<td>top_mount</td>
<td>-1467</td>
<td>451</td>
<td>546</td>
</tr>
<tr>
<td>uca_front</td>
<td>-1651</td>
<td>413</td>
<td>508</td>
</tr>
<tr>
<td>uca_outer</td>
<td>-1480</td>
<td>572</td>
<td>470</td>
</tr>
<tr>
<td>uca_rear</td>
<td>-1251</td>
<td>413</td>
<td>457</td>
</tr>
<tr>
<td>wheel_center</td>
<td>-1441</td>
<td>794</td>
<td>178</td>
</tr>
</tbody>
</table>

Table 7: Adjusted Front Suspension Geometry Points
By changing the locations of all of the hard points in the FutureTruck Front Suspension Subsystem, this subsystem was representative of the BuckHybrid front suspension.

\textit{b. Front Anti-Roll Bar Subsystem}

The Explorer suspension also includes a front anti-roll bar. This was modeled using an anti-roll bar template existing in ADAMS/CAR. Hardpoint locations were specified as shown in Table 8 below. Also, the anti-roll bar subsystem graphic can be seen below in Figure 16.

<table>
<thead>
<tr>
<th>Name</th>
<th>Measured Location (in)</th>
<th>Adjusted Location (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>X</td>
<td>Y</td>
</tr>
<tr>
<td>droplink_external</td>
<td>2.75</td>
<td>-6.5</td>
</tr>
<tr>
<td>droplink_bar</td>
<td>0</td>
<td>-6.5</td>
</tr>
<tr>
<td>arb_bend1</td>
<td>4</td>
<td>-7.25</td>
</tr>
<tr>
<td>arb_bushing</td>
<td>7.25</td>
<td>-13</td>
</tr>
<tr>
<td>arb_middle</td>
<td>7</td>
<td>-30.83</td>
</tr>
</tbody>
</table>

\textbf{Table 8: Front Anti-Roll Bar Hardpoint Locations}
c. Front Suspension Assembly and Testing

A suspension assembly in ADAMS/CAR consists of a suspension subsystem plus anti-roll bar, and any additional user defined subsystems. Shown below in Figure 17 is a schematic of the suspension assembly used to define the BuckHybrid and Stock Explorer front suspensions.
After the suspension subsystem was complete, a number of suspension tests were run in ADAMS using the included suspension test rig. The results from these virtual tests were compared to the actual results from the SPMD, and appropriate changes were made to the geometry points to minimize the error. Both bounce and roll tests were run on the suspension. The bounce test consisted of moving both wheels of the test suspension, in parallel, from -75 to +75 mm of suspension travel. Results from the bounce test can be seen in Figure 18, Figure 19, Figure 20, and Figure 21. The roll test consisted of moving both wheels of the test suspension, in opposition from -75 to +75 mm of wheel travel. The results from the roll test can be
seen in Figure 22, Figure 23, and Figure 24. In these graphs, ADAMS coordinates are used (see Figure 5). This means that positive suspension deflection is suspension compression, positive longitudinal deflection is towards the rear of the vehicle. Steer angle is defined as positive if the wheel is rotated in towards the center of the vehicle, and camber angle is positive if the top of the wheel leans outwards from the body.

![Front Bounce Track Change](image)

**Figure 18: Front Lateral Suspension Deflection vs. Vertical Suspension Deflection**
Figure 19: Front Bounce Camber
Front Bounce Steer

![Graph showing Front Bounce Steer](image)

**Figure 20: Front Bounce Steer**
It was also necessary to tune the suspension spring rate. Shown in Figure 21 is the plot which compares the spring rates. The main ride spring as well as the location at which the jounce bumper comes into effect were tuned in order to match this parameter.

**Front Suspension Spring Rate**

![Graph showing Front Suspension Spring Rate](image)

*Figure 21: Front Suspension Spring Rate*
Front Roll Camber

Figure 22: Front Roll Camber
Front Roll Steer

Figure 23: Front Roll Steer
The above figures show that there is a good correlation between the model suspension and the actual vehicle suspension. Note that the model suspensions were considered to be symmetric and this accounts for some of the error.
2. Rear Suspension Model Development

a. Rear Suspension Subsystem

The rear suspension on the BuckHybrid is also the stock 2002 Ford Explorer suspension. The 2002 Explorer features a new rear independent suspension, which is a short and long arm, coil-over-shock design. The correlation was done in the same way as for the front suspension. Table 9 summarizes the measured, and adjusted rear suspension geometry points. Figure 25 shows the ADAMS model of the BuckHybrid rear suspension.

<table>
<thead>
<tr>
<th>Name</th>
<th>Measured (in)</th>
<th>Adjusted (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>X</td>
<td>Y</td>
</tr>
<tr>
<td>lca_front</td>
<td>-17.25</td>
<td>-12.75</td>
</tr>
<tr>
<td>lca_outer</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>lca_rear</td>
<td>5.5</td>
<td>-21.5</td>
</tr>
<tr>
<td>lwr_strut_mount</td>
<td>4.25</td>
<td>-6.5</td>
</tr>
<tr>
<td>tierod_inner</td>
<td>10.5</td>
<td>-22.5</td>
</tr>
<tr>
<td>tierod_outer</td>
<td>3.05</td>
<td>0</td>
</tr>
<tr>
<td>top_mount</td>
<td>6.75</td>
<td>-8</td>
</tr>
<tr>
<td>uca_front</td>
<td>-3.5</td>
<td>-11.5</td>
</tr>
<tr>
<td>uca_outer</td>
<td>-3.5</td>
<td>0</td>
</tr>
<tr>
<td>uca_rear</td>
<td>8.75</td>
<td>-17</td>
</tr>
<tr>
<td>wheel_center</td>
<td>0</td>
<td>2.5</td>
</tr>
</tbody>
</table>

Table 9: Measured and Adjusted Rear Suspension Geometry Points
There is also an anti-roll bar included in the rear suspension of the Explorer. This was modeled, like the front, using the stock anti-roll bar template. Hardpoint locations are summarized in Table 10, and a figure depicting the rear anti-roll bar can be seen in Figure 26.
Table 10: Rear Anti-Roll Bar Hardpoint Locations

<table>
<thead>
<tr>
<th>Number</th>
<th>Name</th>
<th>Measured Location (in)</th>
<th>Adjusted Location (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>X</td>
<td>Y</td>
</tr>
<tr>
<td>1</td>
<td>droplink_external</td>
<td>-1.25</td>
<td>-6.5</td>
</tr>
<tr>
<td>2</td>
<td>droplink_bar</td>
<td>-1.25</td>
<td>-6.5</td>
</tr>
<tr>
<td>3</td>
<td>arb_bend1</td>
<td>-1.25</td>
<td>-11.5</td>
</tr>
<tr>
<td>4</td>
<td>arb_bushing</td>
<td>-6.75</td>
<td>-16</td>
</tr>
<tr>
<td>5</td>
<td>arb_middle</td>
<td>-6.75</td>
<td>-27.38</td>
</tr>
</tbody>
</table>

Figure 26: Rear Anti-Roll Bar

c. Rear Suspension Assembly and Testing

Again, a suspension assembly was created of the rear BuckHybrid and Stock Explorer suspension. This suspension assembly was then tested in bounce and roll using the suspension test rig. Both bounce and roll tests were done from -75 to +75
mm of wheel travel. Some of the characteristic curves of the real and modeled suspension were compared. The comparisons for the bounce test can be seen in Figure 27, Figure 28, Figure 29, Figure 30, and Figure 31. The results from the roll test can be seen in Figure 32, Figure 33, and Figure 34.

Rear Bounce Base Change

Figure 27: Rear Longitudinal Suspension Deflection vs. Vertical Suspension Deflection
Figure 28: Rear Lateral Suspension Deflection vs. Vertical Suspension Deflection
Rear Bounce Camber

Figure 29: Rear Bounce Camber
Rear Bounce Steer

Figure 30: Rear Bounce Steer
Figure 31: Rear Suspension Spring Rate
Rear Roll Steer

Figure 32: Rear Roll Steer
Figure 33: Rear Roll Camber
Again, a good correlation is found between the measured and modeled suspensions. Although there are slight errors, they are within reasonable limits given that the final goal is only a comparison of trends caused by the hybridization.
3. Steering

The next component in the full-vehicle model of the BuckHybrid is the steering. The steering system which was used is a generic rack and pinion steering subsystem. This was taken from a stock template in ADAMS.

![Rack & Pinion Template ➔ Steering Subsystem](image)

**Figure 35: Steering Subsystem Architecture**

The only change made to the stock template was to the hardpoint locations. The subsystem was shifted so that the ends of the rack matched up with the location of the tie-rods in the front suspension. The steering subsystem can be seen below in Figure 36.

![Steering Subsystem](image)

**Figure 36: Steering Subsystem**
4. Body

The most important subsystem for this particular purpose is the body subsystem. This subsystem holds all of the information which defines the differences between the stock 2002 Explorer, and the *BuckHybrid*, in the form of the vehicle’s inertial data. The architecture of this subsystem can be seen below in Figure 37.

![Figure 37: Body Subsystem Architecture](image)

Table 3 in Chapter III summarized the different inertial parameters of the two vehicles. These parameters were matched in the simulation by using a built-in ADAMS tool called “Automatic Mass Adjustment”. Based on geometries and materials of subsystem parts, ADAMS has a built in calculator of vehicle weight and inertia. Once the calculator has been run, it is possible to modify the values and have ADAMS automatically add weight and inertia to the body subsystem to make the complete vehicle mass and inertia properties exactly what you specify. In a sense, this
addition of “weight” to a particular component is irrelevant to actual computations, however, this is the method employed by the software. A comparison of the measured inertia properties to those obtained in the software using the automatic mass adjustment can be seen below in Table 11. Since this is done with respect to the vehicle center of gravity, the measured center of gravity is shown as the point (0,0,0).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Stock 2002 Explorer Measured</th>
<th>Stock 2002 Model</th>
<th>The BuckHybrid Measured</th>
<th>BuckHybrid Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total Weight (kg)</td>
<td>2016.7</td>
<td>2016.7</td>
<td>2536.1</td>
<td>2536.1</td>
</tr>
<tr>
<td>CG Height (mm)</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Lateral CG (mm)</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Longitudinal CG (mm)</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0.1</td>
</tr>
<tr>
<td>Pitch Inertia (kg•m²)</td>
<td>3538</td>
<td>3480</td>
<td>4513</td>
<td>4510</td>
</tr>
<tr>
<td>Yaw Inertia (kg•m²)</td>
<td>3638</td>
<td>3580</td>
<td>4722</td>
<td>4720</td>
</tr>
<tr>
<td>Roll Inertia (kg•m²)</td>
<td>673</td>
<td>673</td>
<td>924</td>
<td>924</td>
</tr>
<tr>
<td>Roll/Yaw Product (kg•m²)</td>
<td>33</td>
<td>33</td>
<td>7</td>
<td>7.3</td>
</tr>
</tbody>
</table>

Table 11: Inertia Parameters - Comparison of Model to Measured

As can be seen, the inertia parameters match very closely to those measured. A figure of the body subsystem can also be seen below in Figure 38.
5. Wheels

The wheel subsystem in ADAMS/CAR consists of a wheel template and reference to a tire property file. This tire property file is the link to ADAMS/TIRE, whose capabilities were discussed in Chapter II.

![Figure 38: Body Subsystem](image)

![Figure 39: Wheel Subsystem Architecture](image)
ADAMS has available a number of tire models to use when modeling in ADAMS-CAR. In addition, they have a database containing a number of tire model parameters for a variety of different tires. Since tire data for the exact, low-rolling resistance tires used on the FutureTruck was unavailable, one of the ADAMS provided datasets was chosen. The data is provided in a tire property file. In this particular case, this file is actually encoded, so that full access to the tire parameters is actually not available. However, the basic tire size, and basic properties are not encoded. The tire used is a “light truck” tire 245/70 R16. This is the stock size on all but the base Explorer (XLT) model.

Figure 40: Wheel Subsystem
6. Full Vehicle

Finally, all of the components were compiled into one full vehicle assembly. This assembly is shown below in Figure 41.

![Full Vehicle Assembly of the BuckHybrid](image)

**Figure 41:** Full Vehicle Assembly of the *BuckHybrid*
B. Results of Full-Vehicle Simulations

Once the full-vehicle model of the BuckHybrid was complete, a number of simulated tests were run on both the BuckHybrid and the stock 2002 Explorer. These tests were as follows.

1. 90 degrees/9 seconds Step Steer
2. 90 degrees/1 second Step Steer
3. Constant Radius Turn

1. 90 Degree/9 second Step Steer

The first analysis performed on the models was a slowly increasing step steer. The rate of change of the steer angle was set to 10 degrees/second. Parameters for the simulation can be seen below in Table 12.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial Speed</td>
<td>104 kph</td>
</tr>
<tr>
<td>Initial Steer Angle</td>
<td>0 degrees</td>
</tr>
<tr>
<td>Final Steer Angle</td>
<td>90 degrees</td>
</tr>
<tr>
<td>Step Time</td>
<td>9 second</td>
</tr>
</tbody>
</table>

*Table 12: 90 Degree/9 Second Step Steer Parameters*

The steering request can be seen graphically in Figure 42.
This simulation was used mostly as a check of the two (BuckHybrid and Stock Explorer) models. Graphs of yaw rate, lateral acceleration, etc, were examined for spikes, unexpected peaks, and other obvious errors. In addition, as shown in Figure 43, the lateral acceleration of the models was plotted versus the longitudinal velocity multiplied by the yaw rate. During the linear region of the vehicle’s response these quantities should be equal. Figure 43 shows that indeed for both models the simulation shows a result close to this.
Figure 43: Lateral Acceleration vs. Longitudinal Velocity * Yaw Rate

2. 90 Degree/1 second Step Steer

The second test performed on the pair of vehicles was a more severe step steer analysis. The parameters for this analysis can be seen in Table 13.
The initial speed was chosen so that at the beginning of the turning maneuver the vehicle velocity would be 100 kph. Aerodynamic forces were turned off in the simulation, since they only serve to decelerate the vehicle’s longitudinal motion, and therefore do not have any affect on the simulation. This allowed for the vehicle speed to remain more constant over the simulation time. Selected results are shown below in Figure 44, Figure 45, Figure 46, Figure 47, Figure 48, Figure 49, Figure 50, and Figure 51.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial Speed</td>
<td>104 kph</td>
</tr>
<tr>
<td>Initial Steer Angle</td>
<td>0 degrees</td>
</tr>
<tr>
<td>Final Steer Angle</td>
<td>90 degrees</td>
</tr>
<tr>
<td>Step Time</td>
<td>1 second</td>
</tr>
</tbody>
</table>

Table 13: 90 Degree/1 Second Step Steer Parameters
Figure 44: Step Steer 90°/1 s - Longitudinal Speed vs. Time
Figure 45: Step Steer 90°/1 s - Steering Request
Figure 46: Step Steer 90°/1 s - Vehicle Path
Figure 47: Step Steer 90°/1 s - Chassis Lateral Acceleration Vs. Time
Figure 48: Step Steer 90°/1 s - Roll Angle
Figure 49: Step Steer 90°/1 s - Yaw Rate
Figure 50: Step Steer $90^\circ/1$ s - Tire Lateral Slip Vs. Time
As expected, the dynamic response of the BuckHybrid lags behind that of the Stock Explorer. This is due to the increased inertia parameters of the vehicle. In particular, as shown in Figure 51, for a given lateral acceleration, the BuckHybrid shows significantly less roll. This could be considered one of the main positive changes made to the handling of the vehicle when hybridized - mainly from a stability standpoint.
3. Constant Radius Cornering

The final simulation which was done was a constant radius turn. This allowed for evaluation of the understeer/oversteer characteristics of the vehicle. A simple powertrain was added to the model in order to complete this maneuver, since it cannot be done in ADAMS without one. The powertrain model was added with a small amount of mass, and no transient affects. The powertrain served only to add torque at the wheels to increase the speed of the vehicle. Parameters for this test can be seen below in Table 14.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turn Radius</td>
<td>250 m</td>
</tr>
<tr>
<td>Initial Speed</td>
<td>104 kph</td>
</tr>
<tr>
<td>Final Speed</td>
<td>150 kph</td>
</tr>
</tbody>
</table>

*Table 14: Constant Radius Cornering Analysis Parameters*

Figure 52 shows the speed profile that the vehicles had throughout the simulation.
Figure 52: Constant Radius Turn - Vehicle Speed

The path taken by the vehicle in the analysis can be seen below in Figure 53. This plot shows that the Stock Explorer has the ability to stay on path longer than the BuckHybrid. Again, this is due to the smaller handling limits of the hybridized Explorer.
Finally, shown in Figure 54 is the steering angle vs. lateral acceleration of the BuckHybrid and the Stock Explorer. This plot can be used to see the understeer gradient of the vehicles. As described in Chapter II, the understeer gradient is the slope of the curve in this plot. Both the BuckHybrid and the Stock Explorer exhibit
understeer all the way to their handling limit. This is the desired result from a safety standpoint, as the average driver is able to deal with an understeer vehicle more easily.

It should be noted that since the steering system (particularly its compliance) was not modeled in this case, the understeer characteristics introduced by the steering system are not fully represented. In addition, the final handling limit shown on these plots is lower than that of an actual Explorer. This is most likely due to the tire model used.

![Steering Angle Displacement vs. Lateral Acceleration](image)

**Figure 54: Understeer Gradient**
A. Conclusions

According to the results from simulations, the hybridization of the Explorer had a definite impact on the vehicle's handling characteristics. Specifically, it increased the vehicle's understeer gradient, lowered the limit of the lateral acceleration that the vehicle can sustain, and decreased the roll gradient of the vehicle. Although this means that the handling of the BuckHybrid may not be on par with that of stock Explorer, the stability of the BuckHybrid was actually improved, due to the lower center of gravity, and increased roll inertia. This stability increase is evidenced in the decreased roll gradient of the BuckHybrid. All of the differences in the dynamics of the vehicle are a result of the different inertial parameters of the BuckHybrid as compared to a stock Ford Explorer. Although the results may not give an indication of the actual value of these characteristics, the comparison is certainly useful.
B. Future Work

1. Actual Tire Data

Although the tire data used in this work was sufficient to demonstrate the pattern of change in the dynamics of a hybrid vehicle, it was not close enough to predict actual behavior. Tire characteristics have a strong effect on the turning behavior of a vehicle, and more exact tire modeling would be necessary in order to match results such as actual maximum lateral acceleration the vehicle can sustain. It is possible to tune the tire parameters within the ADAMS/TIRE model, this would be a good start towards this end. In addition, tire data specific to the real tires on the vehicle should be sought out from the manufacturer, or the characteristics of the actual tire should be measured on a tire test machine.

2. Optimization of Hardpoint Measurements

Although the hardpoints measurements were slightly adjusted to match the suspension behavior to that measured on the SPMD, a full optimization could be performed. This would be a major task, but similar endeavors have been successful [9]. It is recommended that ADAMS/INSIGHT be used to generate the data which can then be input into an optimization code. INSIGHT allows the user to set up a
number of runs of the same experiment, and output the data in a form which would allow for easy searching of the optimal result.

3. Further Model Verification

Additional model verification would also add to the strength of the model for use as a predictor of actual behavior. Although comparisons from stock to hybrid are valid, more verification would make the actual results more meaningful. In particular, some full-vehicle testing results would allow for verification of the overall behavior of the vehicle. One or two tests such as a step steer, or constant radius test performed at a proving ground could provide enough information to validate the full-vehicle model with a much greater degree of accuracy.

4. Powertrain and Brake System Modeling

A welcome addition to the full-vehicle ADAMS model would be a more detailed powertrain system. Although a simple powertrain was implemented in this work, that powertrain was not really representative of the powertrain in the actual vehicle. Inclusion of the powertrain could be done at a number of levels of increasing detail. The first step would be to include the torque-speed curve of the overall hybrid powertrain. Then, if more detail were desired, the actual complete powertrain could be modeled in ADAMS and even transient effects from the
powertrain could be demonstrated. This would allow for exceptional results in an
acceleration test for example, since the gear shifts could be modeled, as well as the
dynamics of the coupling of the different power sources (in this case engine and
electric motor) in the vehicle.

In addition, it would be helpful to add a braking system to the models of the
vehicles. This would allow for additional tests to be run such as braking while
cornering.

5. Improving the FutureTruck Suspension

Another important use of this type of ADAMS modeling would be to
improve the actual FutureTruck Suspension. Particularly, the spring and damper
properties could be optimized in ADAMS, and then implemented on the actual
vehicle.
REFERENCES


3. FutureTruck 2002 Rules and Regulations


7. ADAMS/CAR Conceptual Suspension Modeling


18. SEA Inc, Vehicle Dynamics Division: Graphical SPMD Results for the BuckHybrid, Sept. 2002.


