A computational design tool for the evaluation of the interaction between pneumatic trucks seats and truck suspensions

Cynthia L. McCoy

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Aug 1994
A COMPUTATIONAL DESIGN TOOL FOR THE EVALUATION OF
THE INTERACTION BETWEEN PNEUMATIC TRUCKS SEATS
AND TRUCK SUSPENSIONS

A Thesis
Presented for the
Master of Science
Degree
The University of Tennessee, Knoxville

Cynthia L. McCoy
August 1994
DEDICATION

This thesis is dedicated to my parents, Robert and Eleanor McCoy. Without their never ending love and support (and nagging) throughout the years, I would never have arrived at this point.
ACKNOWLEDGEMENTS

The research presented in this thesis was funded by National Seating. The research has been an ongoing investigation since 1991.

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I would like to thank my friends, especially Kevin Brown for not letting me take it to seriously, and my family, especially Mom and Dad for their support and belief in me.
ABSTRACT

To improve driver comfort and safety, tractor and cab suspensions and driver seat suspensions have been investigated. The suspension designs have used nonlinear air springs and nonlinear shocks. The interaction between these systems has not been systematically studied. The purpose of this research is to develop and verify computer models of pneumatic truck seats and to study the interaction between the truck seat models and truck models.

A general purpose industrial program, ADAMS, was used to develop the computer models of the seats and trucks. The natural frequencies and the magnitudes of the responses of the seat models and seat-truck system models are investigated.

Experimental testing results were used to verify the seat models. The experimental tests investigated the transmissibilities and natural frequencies of the seats. The truck models were not verified by experimental results.

Results show that the computer seat models agree with the experimental results. Due to the lack of verification, only general trends could be observed in the seat-truck systems. The results show, as a general trend, the magnitudes of the responses of the seat models and the truck models decreases in the combination systems. The natural frequencies of the models may increase or decrease in the combination systems; they do not follow a general trend. With improvements in the computer models and verification of the truck models, the seat-truck models can prove to be a valuable design tool in making appropriate seat selections.
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CHAPTER I

INTRODUCTION

A. Project Overview

During the past three years, National Seating has funded research to study and evaluate air suspension truck seats and their interaction with truck suspensions at the University of Tennessee. National Seating, located in Vonore, Tennessee, manufactures air suspension truck seats. There are four existing seat models: Standard 95, Easyaire 95, Standard 96 and Easyaire 96. The seats differ by the amount of travel the seat can experience, the location of the shock and the location of the spring. The results in this thesis are for the two model 95 seats. The research involves experimentally testing the seats and the individual seat components and developing computer models to study the interaction of the seats with tractor suspensions and for use as seat design tools.

Initially, the project began by experimentally testing National Seating's existing seats. The seats were tested to determine fundamental natural frequencies and transmissibilities. Also, the energy absorption in both the resonance zone of the seats and the 4 - 8 Hz zone, the most sensitive zone for vertical spinal vibration, were determined. The results in this thesis are based on current seat components, while testing has included a new air spring and alternative shock. Tests were performed on the current individual seat components. The shock was tested to determine its temperature dependency, and the bumpers were tested to determine their behavior. Furthermore, tests also analyzed the use of a dead weight to simulate a driver sitting on the seat.

The current research involves the development of computer models of the tractor
seats on a general purpose industrial program, ADAMS. The experimental results are used to verify these models. The computer models allow evaluation of future seat modifications and selection of appropriate seats for different tractor suspensions by studying the interaction of the seat model with the tractor model. The construction and verification of the computer models as well as the initial evaluation of the seat-tractor interaction comprise the subject matter of this thesis.

B. Research Objectives

One long-term goal of the National Seating project is to develop computer models of the tractor seats for use as design tools. Another goal is the development of a computer model for the evaluation of the seat-tractor system. This thesis covers the initial development and verification of the computer seat models, created in ADAMS, Automatic Dynamic Analysis of Mechanical Systems, and the interaction between those seat models and tractor suspension models. The research follows three steps:

I. Experimental Testing

II. Initial Computer Model Creation and Verification

III. Initial Evaluation of the Interaction Between the Seat Models and Truck Suspension Models.
CHAPTER II
BACKGROUN

A. Ride Comfort

Truck drivers suffer from back problems more than any other occupational group, and vibration was determined to be the leading cause of these problems (Pope, 279). Human reaction to vibration depends on the amplitude and frequency of the vibration and exposure time (Elmadany, et al., 307). The International Standard Organization, ISO, provides limits for human exposure to vibration, covering a frequency range of 1 - 80 Hz and including both periodic and random excitation (Elmadany, et al., 308). Wasserman notes that according to the ISO the human body's resonance zone for vertical vibration is 4 Hz to 8 Hz; the resonance zone for fore-aft vibration is 1 Hz to 2 Hz (133). Figure 1 displays the ISO vertical vibration limit curves.

To reduce the occurrence of back problems, there has been an increased interest in improving the ride quality of tractor-trailer systems. Elmadany and Abdulajabbar describe ride comfort as follows.

Ride comfort in a vehicle is the general feeling of well being.

It is affected by a large number of physical and psychological factors. Among these factors are the noise, vibrations, accelerations, pressure changes, temperature, environment, and visual disturbances (325).

Ride comfort is therefore very subjective. Only the factors directly related to the vehicle and seat can be modified to improve driver ride comfort. A number of advances in vehicle and
Figure 1. Vertical acceleration limits as a function of frequency and exposure time (Wasserman, 134).
seat design have been made. These improvements involve reducing the resonant frequency and amplitude of response at resonance that the driver experiences.

B. Pneumatic Truck Seats

Pneumatic truck seats are an attempt to improve driver ride comfort. The seats isolate vertical vibration from the vehicle. The air suspension seat offers two major advantages over the typical static seat. First, the natural frequencies in the suspended seats are lower than those in the static seats, which results in lower transmissibility for a suspended seat. Transmissibility, the ratio of force output to force input, is often indicated as the ratio of output acceleration to input acceleration. Secondly, the air suspended seat also offers the advantage of easy height adjustment when driver's weight changes (Foster, 7). These seats also have drawbacks. The seat motion causes the driver to move relative to the clutch, accelerator and other controls (Foster, 7). If the motion of the seat relative to the cab exceeds the travel of the seat, impacts occur part of topping and bottoming out.

The stiffness and damping control the seat characteristics. Over time, the stiffness and damping decrease due to the heating of the shock and spring. The stiffness increases with increasing air pressure in the spring. Air pressure is increased by both increasing the load on the spring and decreasing the height of the spring. This effect results in a system natural frequency independent of driver weight. Friction in the suspension increases the stiffness and damping at low input magnitudes (Fairley, 134).

Fairley investigated the effect of using a weight to model a person sitting in the seat. Figure 2 displays the transmissibility of an air suspension seat with a person in the seat and
Figure 2. Transmissibility of an air suspension seat: (a) person on the seat, (b) mass on the seat (Fairley, 130-131).
the transmissibility of the same seat with a weight in the seat. While the natural frequencies were the same, the seat response with a person was less than that of the seat with a weight. Fairley attributed this to the effect of the legs (131).

Fairley showed that the seat cushions have little effect on the transmissibility and resonance frequency zone of an air suspension seat (135). Figure 3 shows the transmissibility and resonance frequencies of the cushions, the seat without cushions and the complete seat for three input amplitudes. The cushions have a natural frequency near 4 Hz. The complete seat has a less sharp resonance peak and lower natural frequency than the seat without cushions. The transmissibility of the cushions decreases with increasing input magnitude; the transmissibility in the seat increases with increasing input magnitude.

C. Tractor-Trailer System

The tractor-trailer system is very complex, containing multiple natural frequencies and motions. Dokainish and Elmadany define a seven-degrees-of-freedom basic model. The degrees of freedom include: the bounce and pitch of the tractor, the pitch of the trailer, the vertical motion of the wheels and axles, and the vertical motion of the driver and seat (91). The complexity of the system is further increased when a suspended cab is added to the system.

After the seat, the tractor and cab have the most direct effect on ride comfort. The response of the tractor varies with tractor geometry and suspension. The two most widely used tractor geometries, illustrated in Figure 4, are the conventional tractor and the COE, cab over engine, tractor. The basic differences between the two tractors are the location of the
Figure 3. Transmissibility of seat cushions, typical seat suspension and complete seat (Fairley, 135).
Figure 4. Tractor geometries: (a) conventional, (b) COE
Engine and the location of the seat. In the conventional tractor, the engine is located at the front of the tractor. In the COE tractor, the seat is situated toward the front of the tractor. The suspensions vary among tractors and cabs, producing a wide range of tractor natural frequencies. The natural frequencies fall into two frequency ranges. High frequency tractors have natural frequencies ranging from 3 Hz to 5 Hz. Low frequency tractors have natural frequencies ranging from 1.5 Hz to 2.5 Hz.

Elmadany and Abduljabbar studied the effect of suspending the cab. They found that, without a suspended cab, the tractor natural frequency was 5 Hz. As shown in Figure 5, suspending the cab reduced the natural frequency to 2.5 Hz and reduced the amplitude of response (329). Figure 5 plots vertical acceleration against frequency. The ISO vibration limit curves for one, four, and eight hour exposures are superimposed on the plot. With a suspended cab, the vibration still exceeds the ISO eight hour exposure limits. The tractor in their study was a COE tractor.

Foster analyzed a typical tandem COE tractor-trailer with a suspended cab. He found that the first resonant frequency at the seat location was 3.2 Hz. The driver perceives motion associated with this frequency as bounce or vertical motion. There was another resonant peak at 4.3 Hz; the driver perceives this motion as a combination of bounce and pitch (5). Pitch is a fore-aft rocking motion.

Crosby and Allen studied the effect of cab geometry. They found that the fundamental frequency of a COE tractor was 3 Hz (3). The fundamental frequency of the conventional tractor was 4 Hz (7). They also investigated the effect of driving speed. In both tractor models, increased driving speed increased the response of the system and decreased the first
Figure 5. Comparison of tractor models (Elmadany and Abduljabbar, 329).
natural frequency (3). The driving tests were performed on the same road.

The difference between road test results and lab test results was also investigated. Road test natural frequency was higher than the lab results (Ribarits, et al., 18). While the road test natural frequency was 1.9 Hz, the lab test natural frequency was 3.5 Hz. Ribarits noted that the soft suspended cab did improve ride comfort and also "eliminates the frame vibration dependent ride problem" (Ribarits, et al. 18).

The trailer affects both the ride and system's first natural frequency. The unloaded ride is very rough and less well-controlled than the loaded ride (Dokainish and Elmadany, 89). The natural frequency increases when the trailer is loaded. Dokainish and Elmadany note an increase in first natural frequency from 1.6 Hz in an unloaded system to 2.5 Hz in the loaded system.

D. Vibration Theory

1. One Degree of Freedom

The basic vibration model, Figure 6, consists of a mass and a massless spring and damper. This is a one degree of freedom system because a single coordinate describes its motion. The number of degrees of freedom in a system is equal to the number of independent coordinates necessary to describe its motion (Burton, 1968).

a. Free Vibration

The equation of motion for free vibration of this system is:

\[ m\ddot{x} + c\dot{x} + kx = 0 \]
Figure 6. Basic vibration model
The dots in the above equation represent time derivatives. When disturbed, the undamped system oscillates at the system's natural frequency. In a linear system, the natural frequency, \(\omega_n\), is a function of the system mass, \(m\), and spring constant, \(k\). It is defined as:

\[
\omega_n = \sqrt{\frac{k}{m}}
\]

The response of the damped system, an exponentially decaying function of time, depends on the damping ratio, \(\zeta\), of the system. The damping ratio varies with the critical damping, \(c_c\), and the system damping, \(c\).

\[
c_c = 2m\omega_n
\]

\[
\zeta = \frac{c}{c_c}
\]

If the system is underdamped, \(\zeta < 1\), the response is damped oscillatory motion. The frequency of oscillation, the damped natural frequency, \(\omega_d\), is defined as:

\[
\omega_d = \omega_n\sqrt{1-\zeta^2}
\]

If the system is overdamped, \(\zeta > 1\), the response is an aperiodic exponentially decaying function. The response of the critically damped, \(\zeta = 1\), system, also an aperiodic exponentially decaying function, depends on the initial conditions of the system (Thomson, 1988).
b. Forced Vibration

If the system is subjected to a forcing function or excitation, the equation of motion becomes:

\[ mx + cx + kx = F(t) \]

The solution becomes a combination of the solution to the homogeneous equation and the particular solution. The particular solution depends on the type of excitation. If the excitation is harmonic, the particular solution is also harmonic with the same frequency as the excitation. If the frequency of excitation corresponds to the natural frequency, resonance occurs. The magnitude of the response at resonance depends on the damping in the system.

Figure 7 demonstrates the effect of forcing function frequency and the effect of damping on resonance. As damping increases, the resonant frequency and the resonant amplitude peak decrease. In the first part of the Figure, the ratio of system response to system static deflection is plotted against the ratio of forcing function frequency to natural frequency. As the forcing frequency increases toward the natural frequency, the response increases. The maximum response occurs when the forcing function nears the natural frequency. Due to the damping, the resonant frequency does not coincide with the natural frequency. As the forcing function frequency continues to increase, the response decreases.

The second part of Figure 7 shows the phase angle between the input and the response. As damping increases, the curves become flatter because damping tends to smooth out the sharpness of the phase angle diagram. For all levels of damping, the phase angle passes through 90 degrees at resonance.
Figure 7. Response of typical single degree of freedom system to harmonic forcing function: (a) amplitude response for various damping levels, (b) phase angles for various damping levels (Den Hartog, 51).
2. Two-Degrees-of-Freedom Systems

The single-degree-of-freedom vibration theory is readily extended to two-degrees-of-freedom systems. A simple two degrees of freedom system is shown in Figure 8. This system has two natural frequencies. The equation of motion becomes a set of two equations, one equation for each mass. As in the single degree of freedom system, the undamped free response is harmonic. Associated with each system natural frequency is a natural mode shape. The natural mode shape, a characteristic of the system, depending on inertia and stiffness, is the displacement configuration that the system assumes (Thomson, 1988). Each mode can be excited independently. Like the single degree of freedom system, the forced response is also a combination of a particular response and the free response.

When a two-degree-of-freedom system has one mass considerably larger than the other, the smaller mass can act as a dynamic vibration absorber. When the forcing function frequency is the same as the natural frequency of the absorber mass, the main mass will have very little motion; the absorber mass absorbs the motion. Both the systems' masses and the damping affect the vibration absorption. The masses of the system determine the natural frequency ratio. If the absorber mass is very small, relative to the main mass, the two natural frequencies should be nearly equal to provide the best absorption. As absorber mass increases, its natural frequency, relative to the main mass natural frequency, decreases. Figure 9 displays the effect of damping. In Figure 9, amplitude ratio of the main mass is plotted against the ratio of the forcing function frequency to main mass natural frequency. In the system analyzed for this plot, the main mass is twenty times larger than the absorber mass, \( \mu = 1/20 \), and the natural frequencies of both masses are the same, \( f = 1 \). The plot shows that,
Figure 8. Basic two degrees of freedom model.
Figure 9. Effect of absorber damping on response of main mass in a typical dynamic vibration absorber (Den Hartog, 97).
if there were no damping, there would be two resonance zones. If the damping were infinite, there would be only one resonance zone because the masses move together. For other levels of damping, the curve lies between these two extremes.

3. Nonlinear Systems

The response of a nonlinear system is much more complex than its linear counterpart because of the lack of linear predictability. A nonlinear system is one whose mass, spring constant or damping coefficient cannot be defined by constants, creating nonlinear differential equation(s), which results in specific instead of general solutions. The response of a nonlinear system is difficult to predict. Only a few simple systems have closed-form solutions. For most systems, approximate solutions must be found. There are two basic methods used to solve these systems, linearization, which increases the number of equations defining the system, and numerical solutions. The linearization method is not useful for systems with strongly nonlinear characteristics; these systems must be solved numerically.

In systems containing nonlinear springs, the system's natural frequency is affected by forcing function amplitude (Den Hartog, 361). In a system with softening springs, the natural frequency decreases with increasing forcing function amplitude (Burton, 132). Hardening springs display the opposite trend. Unlike the system with nonlinear springs, the natural frequency of a system with nonlinear damping is generally not affected by forcing function amplitude (Den Hartog, 361).

The response of the system with nonlinear springs is also affected by continuous forcing frequency changes as illustrated in Figure 10. The arrows in Figure 10 follow the
Figure 10. The effect of altering forcing function frequency on systems with nonlinear springs: (a) soft spring system with frequency increased from zero, (b) soft spring system with frequency decreased from a high value, (c) hard spring system with frequency increased from zero, (d) hard spring system with frequency decreased from a high value (Burton, 138).
response path, indicating direction of frequency sweeping. The dotted lines designate changes in the response due to the frequency changes.

Riganti investigated the differences in solutions due to the approximate technique chosen. Responses of a two degree of freedom dynamic absorber with a nonlinear spring obtained by three different approximation techniques were compared. The first techniques assumed a sinusoidal solution; the second technique assumed the sinusoidal response was perturbed slightly (Riganti, 176). The final method was the Runge-Kutta numerical method. Figure 11 displays the amplitudes determined by each approximation method. In Figure 11, amplitude is plotted versus a non-dimensional frequency parameter. Using the Runge-Kutta solution as a reference, the first method under-predicted the amplitudes and the second method over-predicted the amplitudes. Riganti noted that as the frequency parameter increased, the solution from the second method converged to the Runge-Kutta solution (182). Clearly, the choice of approximate technique affected the solution.

E. ADAMS

Automatic Dynamic Analysis of Mechanical Systems, ADAMS, produced by Mechanical Dynamics Incorporated, Ann Arbor, Michigan, is a commercial software package for analysis of mechanical systems; it simulates forces and motions of these systems. As a numerical solver, it accommodates complex nonlinear systems undergoing large displacements. It analyzes both kinematically determinat, zero degree of freedom, and multi degrees of freedom systems (Mechanical Dynamics, ADAMS Reference Manual 6.1, 1-3).
Figure 11. Amplitude responses of masses, $m_1$ and $m_2$, and relative motion, $\eta$ (Riganti, 178).
The use of ADAMS is divided into these parts:

I. Preprocessing

II. Analysis

III. Post processing.

Preprocessing involves creating a computational model of a mechanical system. The model consists of geometric definitions, motion definition between parts, forces between model components and external excitations. A part must have mass, a defined center of mass and can include inertial properties and moments of inertia, either user-defined or computed by ADAMS. All part positions and force and constraint locations are defined by points called markers. Markers are also used to designate centers of mass.

Constraints apply motion restrictions to parts in a model. Constraints on a model are imposed through joints, time dependent motion and user-written subroutines. ADAMS contains various joint types, including revolute, translation and fixed joints. Revolute joints allow only rotation between the two mating parts. Translation joints allow linear displacement between the two connected parts. Fixed joints do not permit any motion between the two mating parts.

The forces include active elements of the system as well as gravity. The ADAMS gravity force permits user-defined gravity constants to be set in three directions. Forces in a model may be vectors or single component. For a single component force, the direction of action is along the line connecting the two force defining markers. If the force is negative, it causes an attraction between the two markers (Mechanical Dynamics, 2-20). A vector force is a multi-component force. This force may be applied to a single marker or between
two or more markers with the axes of the marker coordinate system defining the direction of action of the force components. In a vector force, all three components must be specified (Mechanical Dynamics, 2-20). ADAMS contains specialized force statements to model specialized elements, like a linear spring-damper, in a model. To model other forces in a system, ADAMS contains general functions, including a standard polynomial, a simple harmonic function and a step function. The simple harmonic function is a continuous sine wave with a user specified amplitude and frequency. The step function is used when a force has a discontinuity, jumping from one value to another. These functions may be used as motions provided the dependent variable is time. To further aid in creating forces and motions, ADAMS supports FORTRAN intrinsic functions.

Also, during preprocessing, requests for results files are made. Displacement or acceleration requests return the magnitude and individual components of translational displacement or acceleration between two specified markers. In addition to translational results, acceleration requests also return rotational accelerations about fixed axes. Velocity requests return the magnitude and individual components of translational or rotational velocity. Also, velocity requests return the relative velocity of one marker with respect to another. If the two markers are separating, the relative velocity is positive; otherwise, it is negative (Mechanical Dynamics, ADAMS Reference Manual 6.1, 4-16).

Once the model is constructed, analysis can be performed. Analysis or simulation has four parts:

I. Verification

II. Equation Generation
III. Equation Solution

IV. Results Creation.

First, the model is verified. Verification guarantees that a part has a force acting on it or is connected to another part. Verification ensures that each part has mass and checks joints to insure that the mating parts are at the same location. In addition, the number of degrees of freedom is computed and the system is checked for redundant constraints (Mechanical Dynamics, 6-6).

Once the model has been verified, the equations governing the motion of the system are generated. Twelve first order equations are generated for each part, relating forces to accelerations and positions to velocities. One algebraic constraint and one differential equation are created for each system constraint. Finally, one algebraic equation is created for each single component force (Mechanical Dynamics, 3-10). After the equations are compiled, they are solved using a predictor-corrector method. The system solution can include time-dependent kinematic behavior, static equilibrium, or the time-dependent nonlinear dynamic behavior.

Finally, ADAMS creates the output file sets. These files contain time histories of system displacements, velocities, accelerations and forces. These files may also contain a map of the equations and variables used in the system, initial displacement and velocity values and part inertias and orientations. Additionally, files containing time histories of requested output are created (Mechanical Dynamics, 7-4).

Finally, post processing occurs. ADAMS presents time history results in tabular form or as plots. Analysis can be performed on the time history results. Single results sets can be
integrated, differentiated, or filtered. Using an FFT, fast Fourier transform, time domain solutions can be converted to frequency domain solutions. Two sets can be averaged or combined through multiplication, division or addition. Plots can be created for any results set or modified set. ADAMS also animates the model. The motion of the model can be viewed as either still-frame displays at a specified time or as continuous motion which allows the user to observe the model motion and part interaction.

ADAMS offers several advanced techniques to increase the quality of model construction and model analysis. ADAMS accepts direct transfer of data from CAD programs for use in building models. The program also contains a special preprocessor and post processor aimed toward vehicle simulation. Furthermore, ADAMS provides a method to easily create a multi-body model of a person. This model may be incorporated into any existing model (Ryan, 146).
CHAPTER III

EXPERIMENTAL PROCEDURES

Experimental tests were performed at both the National Seating and University of Tennessee facilities to determine the natural frequencies and the magnitude of response at resonance for each seat. Swept sine tests were used to perform the testing.

A. Apparatus

The experimental apparatus set up is shown in Figure 12. The truck seat, which is bolted to the front edge of the shaker platform, is mounted in the same plane as the hydraulic cylinders. The canister weight on the seat models the driver. Two accelerometers are used to record the response of the platform-seat system. The PCB Seat Pad accelerometer, positioned between the weight and the seat, measures the seat response. The PCB 393C seismic accelerometer, which is mounted on the platform near the seat, measures the platform response. Both accelerometers are connected to a PCB power unit before connecting to the analyzer.

The Hewlett Packard 3562A Digital Signal Analyzer, HP, performs two functions. First, the analyzer is the input function generator. The source signal from the HP is sent to the Servo Controller and then is sent to the hydraulic cylinders on the shaker platform. The resulting motion is angular, but because the table is long (approximately eight feet long) compared to the vertical cylinder motion, angles of less than five degrees are produced. For small angles of rotation, the motion is assumed vertical. The HP's second function is to perform the frequency response analysis and coherence analysis of the system. The test
Figure 12. Experimental setup (Mullinix, 13).
method used is a swept sine test. The analyzer set up for swept sine testing is shown in Figure 13.

B. Swept Sine Testing

The swept sine test inputs a harmonic excitation to the system. The frequency of the excitation is swept up until a phase angle of $90^\circ$ is detected. Then, that frequency is locked in until the system stabilizes. Once the system has stabilized, data is taken. It is a steady state test.

Swept sine is the only test which documents any nonlinearities in a system, producing a particular solution to a nonlinear system rather than a linear approximation. The drawbacks to this test are the length of time and amount of equipment required to perform the test. Also, because it documents system nonlinearities, it is difficult to obtain a linear model of a nonlinear system for comparison to a linear computer model (Wasserman, J., 63).
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Setting</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>AVERAGE:</strong></td>
<td><strong>INTGRT TIME</strong></td>
</tr>
<tr>
<td></td>
<td>50.0 ms</td>
</tr>
<tr>
<td></td>
<td><strong># AVGS</strong></td>
</tr>
<tr>
<td></td>
<td>5</td>
</tr>
<tr>
<td><strong>FREQ:</strong></td>
<td><strong>START</strong></td>
</tr>
<tr>
<td></td>
<td>500 MHz</td>
</tr>
<tr>
<td><strong>STOP</strong></td>
<td>8.5 Hz</td>
</tr>
<tr>
<td><strong>SPAN</strong></td>
<td>8.0 Hz</td>
</tr>
<tr>
<td><strong>RESLTN</strong></td>
<td>40.0 MHz</td>
</tr>
<tr>
<td><strong>SWEEP:</strong></td>
<td><strong>TYPE</strong></td>
</tr>
<tr>
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<td>Linear</td>
</tr>
<tr>
<td><strong>DIR</strong></td>
<td>Up</td>
</tr>
<tr>
<td><strong>EST TIME</strong></td>
<td>2.18 Min</td>
</tr>
<tr>
<td><strong>EST RATE</strong></td>
<td>61.1 mHz/Sec</td>
</tr>
<tr>
<td><strong>AU GAIN:</strong></td>
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</tr>
<tr>
<td><strong>INPUT:</strong></td>
<td><strong>RANGE</strong></td>
</tr>
<tr>
<td>CH 1</td>
<td>AutoRng</td>
</tr>
<tr>
<td></td>
<td>949 mV/EU</td>
</tr>
<tr>
<td>CH 2</td>
<td>AutoRng</td>
</tr>
<tr>
<td></td>
<td>105 mV/EU</td>
</tr>
<tr>
<td><strong>COUPLING</strong></td>
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</tr>
<tr>
<td><strong>SOURCE:</strong></td>
<td><strong>TYPE</strong></td>
</tr>
<tr>
<td></td>
<td>Swept Sine</td>
</tr>
<tr>
<td><strong>LEVEL</strong></td>
<td>3.09 Vpk</td>
</tr>
<tr>
<td><strong>OFFSET</strong></td>
<td>0.0 Vpk</td>
</tr>
</tbody>
</table>

Figure 13. Analyzer setup for swept sine tests (Mullinix, 14).
CHAPTER IV
COMPUTER MODELS

A. Seat Models

Due to the seat symmetry and the fore-aft isolator, the National Seating Standard 95 and Easyaire 95 seats are modeled as single-degree-of-freedom systems. The fore-aft isolator allows the seat to move in the fore-aft direction so that the seat seems to float. The isolator helps isolate small random impulses to the seat that are often seen during highway driving. Because the frequency of impulses is much greater during city driving, the isolator is not as effective and, as a result, is often locked during city driving. A photograph of the Standard 95 seat is shown in Figure 14. Both seats consist of an upper base, including seat cushions, connected to a lower base by two connecting arms with revolute joints. Each seat contains a nonlinear Firestone air spring and a nonlinear cofap shock absorber. The mass of each seat is 6.74 slugs, including a representative driver mass. The representative driver mass is a lumped mass fixed to the upper base and seat cushions. The only difference between the two seats is the location of the spring. The ADAMS models of the Standard 95 and Easyaire 95 seats are shown in Figures 15 and 16, respectively.

The stroke of each seat is controlled by the seat spring, seat shock and the seat geometry. Each seat has a set of rubber bumpers to soften the 'bottoming out' of the seat by reducing the impact; the deflecting bumpers dissipate the impact. When a seat reaches the bottom of its stroke, it imparts a sharp impact to the driver because the driver stops with the seat. The impact at the top of the seat stroke does not have as strong an effect on the driver
Figure 14. Standard 95
Figure 15. ADAMS Standard 95 model
Figure 16. ADAMS Easyaire 95 model
because the seat belt, not the seat itself, stops the driver’s motion. There are no rubber bumpers to soften the impact at the top of the stroke; it is a metal to metal impact. The bumper forces were determined by curve-fitting experimental data. The experimental testing showed that the rubber bumpers act differently under compression and rebound. Therefore, the lower bumper force, \( B_L \), is a two-part force. The upper bumper force, \( B_W \), mimics the metal to metal impact. The bumper forces, which vary with vertical seat displacement, \( X_y \), are:

\[
B_L = 763.263X_y - 2117.02X_y^2 + 4074.6X_y^3 \\
B_{L2} = 523.8X_y \\
B_W = -(35000X_y)
\]

Step functions are used to engage and disengage the bumper forces. Therefore, the force will actually start before the bumpers come into contact. This effect is small and assumed negligible. The actual forces used in ADAMS are located in appendix B.

The cofap shock absorber is a nonlinear device with separate load velocity curves dependent on direction, and the rebound curve of the shock is a sigmoidal curve separated into three separate forces, \( SH \). The forces were determined by curve-fitting experimental data. The experimental tests measured the force necessary to displace the shock a specified amount at a specified oscillating velocity. The shock forces used in ADAMS are reduced from the experimentally determined forces. In the seat, the shock force bends the shock attachment bar and rocks the seat on its supports. Therefore, the force distance becomes less accurate. As a result, the force imparted to the seat is not as strong. The forces vary with the velocity magnitude, \( V_m \), or the relative velocity, \( V_r \), of the shock. Like the bumper forces, the shock forces are engaged or disengaged by step functions. The shock forces with
their corresponding velocities ranges are:

\[ SH_p = -(251 + 14.4V)_r, \quad V_r > 1.63 \]
\[ SH_f = -(87.82 + 114.6V)_r, \quad -1 < V_r < 1.63 \]
\[ SH_n = -13.57 + 20.25V_m, \quad V_r < -1 \]

There is a spacer included in the shock forces because of the tolerance in the mounting pins of the shock. When the shock changes curves, there is some free play or rattle in the pin connections which reduces the shock forces.

In the Standard 95 seat, the air spring, which is characterized by a fifth-order power series equation, is connected between the lower connection arm and the upper base; the air spring is connected between the lower base and the upper base in the Easyaire 95 seat. The lower connection point of the spring is located at the same height in both models, but it is attached to different parts. The spring location in Figure 16 is exaggerated to illustrate the difference in connection point locations between the two seats. Tests conducted at Firestone determined the dynamic spring rates and the spring damping rate. To produce the equations, measurements were taken while cycling the air spring with a frequency of 1 Hz. The dynamic spring rates are not the same, forming a hysteresis loop; by definition, the difference between them is the damping in the spring (Burton, 1968). Therefore, only two of the three equations determined by Firestone are needed to characterize the spring. The spring damping force, \( SP_s \), is proportional to the relative velocity, \( V \), of the spring. The dynamic spring force, \( SD \), varies with displacement, \( X \), of the spring. The dynamic force determined at Firestone is modified to insure that the static equilibrium position of the seat is at the midpoint of the seat stroke; the seat can travel three inches above or below the equilibrium position. To insure that the static equilibrium position of the seat is maintained, the dynamic spring force in the
Easyaire 95 model is different from the dynamic spring force in the Standard 95 model. The spring forces are:

\[ SP_d = -2.206V \]
\[ SD_{\text{std}} = 440.6 + 204.6X + 66.5X^2 + 2.7X^3 - 6.94X^4 + 4.83X^5 \]
\[ SD_{\text{Easy}} = 211.4 + 98.2X + 31.9X^2 + 1.3X^3 - 3.33X^4 + 2.3X^5 \]

All external excitations or motions that are input to the system are simple harmonic time-dependent functions. The motion enters the system through the translational joint connecting the lower base to the stationary ground. The motion is transferred to the upper base through the connecting arms and seat suspension.

**B. Cab Models**

A tractor is a multi-degree-of-freedom system with multiple suspensions and system inputs. However, the cab subassembly can be modeled as a single-degree-of-freedom system. Therefore, a single-degree-of-freedom cab model is added to the existing seat models. The terms tractor and cab will be used interchangeably to indicate the single-degree-of-freedom system added to the seat models. The two different geometries, discussed in chapter 2, are used. Both cab models are hinged to the stationary ground at the front of the cab which allows rotation of the front end of the cab relative to the ground but does not allow translation relative to the ground. The suspension is located at the rear of the cab. Most commercial cabs are hinged to the tractor at the front end with the suspension at the rear. The mass of each cab is 65 slugs. The center of mass is at the midpoint of the cab length. The difference in the two models is the location of the seat. In the COE cab, the seat is
positioned forward of the cab's center of mass, halfway between the front end and the center of mass. In the conventional cab, the seat is positioned behind the cab center of mass, halfway between the rear end and the center of mass. This geometrical difference insures that, for any input to both cab systems, there is more motion transferred to the seat base in the conventional cab system than in the COE cab system. Figure 17 shows the COE cab ADAMS model; Figure 18 shows the conventional cab ADAMS model. The seat is included in both Figures 17 and 18 to illustrate the difference between the conventional and COE models.

The motion is input at the rear of the cab. There is an input block at the rear of the cab to aid in entering the motion at the cab suspension. The block is connected to the stationary ground by a translational joint, and the cab suspension is attached to the block. The motion enters the block and is directly transferred to the suspension.

Two natural frequencies are used with each cab geometry, representing the two cab frequency ranges. The low frequency cabs have natural frequencies of 1.7 Hz. The high frequency cabs have natural frequencies of 3.7 Hz. These frequencies were achieved by using specific suspension characteristics.

Both linear and nonlinear springs are used in the cab model suspensions. The nonlinear spring is a modified version of the Firestone spring used in the seat. The force is modified to insure that the systems' natural frequencies remain the same as the linear systems' natural frequencies. The damping forces in both systems are linear.

In the linear system, the low frequency cab models share the same damping force, but each cab model has a different spring force. The damping force, D, is proportional to the
Figure 17. ADAMS COE tractor model.
Figure 18. ADAMS conventional tractor model.
relative velocity, $V_r$, and the spring forces, $S$, are proportional to displacement, $X_m$. The spring and damping forces for the low frequency linear cab models are:

\[
S_{COB} = 1101.34 + 1300X_m \\
S_{conv} = 120638 + 1175X_m \\
D_{COB} = 14.5V_r \\
D_{conv} = 13.5V_r
\]

In the high frequency models, both the spring force and damping force are different.

\[
S_{COB} = 1101.34 + 280X_m \\
S_{conv} = 120638 + 260X_m \\
D = 6.5V_r
\]

In the nonlinear systems, the low frequency cab models have different damping and spring forces. The spring force is a function of displacement, $X_m$; the damping force is proportional to relative velocity, $V_r$. The low frequency forces are:

\[
S_{COB} = 1101.34 + 126.4X_m + 41X_m^2 + 1.67X_m^3 - 4.28X_m^4 + 2.98X_m^5 \\
S_{conv} = 120638 + 126.4X_m + 41X_m^2 + 1.67X_m^3 - 4.28X_m^4 + 2.98X_m^5 \\
D_{COB} = 6V_r \\
D_{conv} = 6.5V_r
\]

Likewise, the high frequency models have different damping and spring forces. These forces are:

\[
S_{COB} = 1101.34 + 617X_m + 200.5X_m^2 + 8.1X_m^3 - 20.9X_m^4 + 14.5X_m^5 \\
S_{conv} = 120638 + 560.9X_m + 182.3X_m^2 + 7.4X_m^3 - 19.0X_m^4 + 13.2X_m^5 \\
D_{COB} = 14.5V_r \\
D_{conv} = 14V_r
\]
CHAPTER V

RESULTS AND DISCUSSION

A. Experimental Results

The experimental testing results of the truck seats are presented for completeness. The author did not perform the experimental tests. The results are used to verify the ADAMS seat models.

Because the isolator is not included in the ADAMS seat models, the results presented are for the seats with the fore-aft isolator locked. As a general trend, unlocking the isolator reduces both the natural frequency of the seat and the magnitude of the response at the natural frequency.

1. Standard 95

The results of the swept sine testing of the Standard 95 seat with the fore-aft isolator locked are presented in Figure 19. The Figure plots the transmissibility, the ratio of seat acceleration to platform acceleration, and the coherence of the test. The transmissibility plot indicates a natural frequency of 1.95 Hz. The transmissibility increases until the natural frequency is reached and then decreases, displaying a typical damped, single degree of freedom system response to a harmonic forcing function. The magnitude of the transmissibility at the natural frequency is 3.156 dB. The dB value is converted to magnification factor of 1.438 by:

\[ dB = 20 \log_{10}(MF) \]

The coherence is an indication of the quality of the test by illustrating the agreement between
Figure 19. Transmissibility of Standard 95 (Mullinix, 27).
averages taken during testing. For a swept sine test, because it is a steady state test and averages are not taken until the system stabilizes, the coherence should remain near one.

2. Easyaire 95

In Figure 20, the transmissibility of the Easyaire 95 seat and the coherence of the swept sine testing are presented. The transmissibility curve follows the same general trend as the Standard 95 transmissibility curve shown in Figure 19. The transmissibility curve indicates a natural frequency of 2.74 Hz and a corresponding magnitude of 7.104 dB. The magnification factor for the Easyaire 95 is 2.266. The response of the Easyaire 95 is greater than the Standard 95 because, due to the differences in spring connections, the spring experiences more displacement in the Easyaire 95, and, as a result, the spring in the Easyaire 95 is operates at different locations on the spring force curve. As expected, the coherence remains near one. The experimental results for both seats are summarized in Table 1.

<table>
<thead>
<tr>
<th>Seat</th>
<th>Natural Frequency (Hz)</th>
<th>Magnification Factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Standard 95</td>
<td>1.95</td>
<td>1.438</td>
</tr>
<tr>
<td>Easyaire 95</td>
<td>2.74</td>
<td>2.266</td>
</tr>
</tbody>
</table>

B. Computer Simulation

All models are excited by continuous harmonic functions. An analysis consists of a number of simulations. During an analysis, the amplitude is constant, but the excitation
Figure 20. Transmissibility of Easyaire 95 (Mullinix, 32)
frequency varies with each simulation. For each simulation, an integration time of five seconds and five hundred time steps are used. Increasing the number of time steps beyond one hundred steps per second did not improve the solution.

1. Seat Model Analysis

The results of the analyses are presented as amplitude ratios. During each simulation, the absolute displacement, the displacement of the upper seat base relative to the ground, and the relative displacement, the displacement of the upper seat base relative to the lower seat base, are determined. The amplitude ratios are computed by dividing the corresponding displacement by the peak-to-peak input amplitude.

a. Standard 95 Model

The Standard 95 model was analyzed for model verification, and the results are presented in Figure 21. The natural frequency of the model is 1.9 Hz, which is 2.5% lower than the experimentally determined natural frequency. The magnification factor is 1.635, which is 13.5% greater than the experimental value. In Figure 21, results for both absolute and relative amplitude ratios are presented. As the frequency increase beyond the natural frequency, the relative amplitude ratio approaches unity and the absolute amplitude ratio approaches zero. This indicates that the displacement of the upper seat base diminishes and the relative displacement is due to the input motion as the system moves away from the natural frequency.

The effect of input amplitude was investigated. Figure 22 shows that the input
Amplitude Ratio

Figure 21. Response of Standard 95 model.
Figure 22. Input amplitude effects on Standard 95 model.
amplitude has a significant effect on both the natural frequency of the model and the magnitude of response. As input amplitude increases, the model natural frequency decreases and the magnitude of response increases. Table 2 summarizes the effect of the input amplitudes on the Standard 95 model.

Table 2. Input amplitude effect on the Standard 95 seat model

<table>
<thead>
<tr>
<th>Amplitude</th>
<th>Natural Frequency (Hz)</th>
<th>Magnification Factor</th>
</tr>
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<tbody>
<tr>
<td>1.0&quot; P-P</td>
<td>4.0</td>
<td>1.385</td>
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<td>1.5&quot; P-P</td>
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<td>1.441</td>
</tr>
<tr>
<td>2.0&quot; P-P</td>
<td>1.9</td>
<td>1.635</td>
</tr>
<tr>
<td>2.5&quot; P-P</td>
<td>1.7</td>
<td>1.784</td>
</tr>
</tbody>
</table>

b. Easyaire 95 Model

The model was analyzed for verification, and the results are presented in Figure 23. The natural frequency of the Easyaire 95 model is 2.8 Hz, and the magnification factor of the response is 2.37. The computed natural frequency is 2.2% greater than the experimental natural frequency. The experimental magnification factor is 4.5% lower than the computed magnification factor. The response trends follow those of the Standard 95 model trends.

The input amplitude effect on the model was also investigated. Figure 24 shows that, like the Standard 95 model, the input amplitude has a great effect on the model response. Like the Standard 95 model, the magnitude of response increases with increasing input amplitude. However, the natural frequency does not follow the same trend as the Standard
Figure 23. Response of Easyaire 95 model.
Figure 24. Input amplitude effects on Easyaire 95 model.
The natural frequency decreases with increasing amplitude, but then, as the amplitude continues to increase, the natural frequency increases. The results of this investigation are summarized in Table 3. The results for the final table entry, 2.5" P-P, are not located on Figure 24. The magnification factor is not determined by the seat suspension but by the seat geometry. The relative displacement is limited to six inches because the large input causes the seat to impact the upper and lower bumpers.

Table 3. Input amplitude effect on response of Easyaire 95 model

<table>
<thead>
<tr>
<th>Amplitude</th>
<th>Natural Frequency (Hz)</th>
<th>Magnification Factor</th>
</tr>
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<tbody>
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<td>1.0&quot; P-P</td>
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<td>2.0&quot; P-P</td>
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<td>2.2&quot; P-P</td>
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</tr>
<tr>
<td>2.5&quot; P-P</td>
<td>2.90</td>
<td>2.400</td>
</tr>
</tbody>
</table>

2. Cab Model Analysis

Each geometry investigated, COE and conventional, has a linear and nonlinear suspension, and both geometries also have low frequency and high frequency suspensions. This creates eight total models. Figure 25 displays the results of the linear cab analyses. The figure plots relative amplitude ratio as a function of forcing function frequency. The relative amplitude ratio is the relative displacement of the cab divided by the peak-to-peak input amplitude. The relative cab displacement is the difference between the displacement of the rear end of the cab and the input block displacement. For each analysis, the input amplitude
Figure 25. Response of linear tractor models.
is one inch peak-to-peak. Figure 25 shows that the two low frequency models have a natural frequency of 1.7 Hz, and the response of the COE model is greater than the response of the conventional model. Both high frequency models have a natural frequency of 3.7 Hz. As in the low frequency models, the COE model has the greater response, but the difference is not as pronounced.

In the nonlinear models, the responses of the cabs follow the same trends as the linear models as seen in Figure 26. For each analysis, the input amplitude is two inches peak-to-peak. The natural frequencies are 1.7 Hz and 3.7 Hz. The COE cab models have more response at the natural frequencies. The magnitude of the response of the nonlinear cabs are less than those of the linear cabs. The responses of the nonlinear cabs are input amplitude dependant. Both the natural frequency and the magnitude of response decreases with decreasing input amplitude. Table 4 summarizes the cab model analyses.

Table 4. Summary of cab responses.

<table>
<thead>
<tr>
<th>Cab</th>
<th>Linear Model</th>
<th>Nonlinear Model</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Natural Frequency (Hz)</td>
<td>Magnitude</td>
</tr>
<tr>
<td>COE</td>
<td>1.7</td>
<td>4.11</td>
</tr>
<tr>
<td>Conventional</td>
<td>1.7</td>
<td>3.76</td>
</tr>
<tr>
<td>COE</td>
<td>3.7</td>
<td>4.03</td>
</tr>
<tr>
<td>Conventional</td>
<td>3.7</td>
<td>3.92</td>
</tr>
</tbody>
</table>
Figure 26. Response of nonlinear tractor models.
3. Seat-Cab System Analysis

The seat models were added to the eight existing cab models. The input to these seat-cab systems are continuous harmonic functions. As in the cab models, the motion is input to the seat-cab systems through the input block at the rear of the cab. Like the seat models and cab models, the seat-cab system responses are presented as relative amplitude ratios. The relative amplitude ratio of the seat is the displacement of the upper seat base relative to the lower seat base divided by the absolute displacement of the cab at the seat location. The relative amplitude ratio of the cab is the displacement of the rear of the cab relative to the displacement of the input block divided by the peak-to-peak input amplitude.

The responses of the seats and cabs alone are used as references to which the responses of the seats and cabs in the seat-cab systems are compared. It is difficult to accurately compare the responses of the seats in the seat-cab systems with the seat only. Unlike the seat only, the input amplitude to the seat in the seat-cab systems is not constant; every measurement point has a different input amplitude. Also, the responses of the seats in the seat-cab systems is the transient, not steady state, response.

a. Standard 95-Cab System Models

The responses, relative amplitude ratios, of the seat in the linear and nonlinear low frequency (1.7 Hz) conventional cab systems are displayed in Figure 27. The input amplitude for the linear cab system is one inch peak-to-peak, and the amplitude for the nonlinear cab systems is two inches peak-to-peak. The response of the seat alone is included in Figure 27 for reference. In both seat-cab systems, the magnitude of response is greater and the natural
Figure 27. Comparison of Standard 95 responses in linear and nonlinear low frequency (1.7 Hz) conventional tractor systems.
frequency is lower than the response of the seat only. Unlike the response of the seat only, the responses of the seats in the systems sharply decline past the natural frequency. The seat in the nonlinear system rebounds to follow the decaying trend of the seat only.

The effects of the seat on the cabs' responses is shown in Figure 28 and Figure 29. In the linear system, including the seat reduces the magnitude of response of the cab but increases the natural frequency. In the nonlinear system, the natural frequency and magnitude of response are greatly reduced.

The effects of the linear and nonlinear low frequency (1.7 Hz) COE cabs on the Standard 95 are shown in Figure 30. The input amplitude is two inches peak-to-peak for both seat-cab systems. In both systems, the natural frequency of the seat is reduced slightly. However, there is a significant decrease in the magnitude of seat response. The effects of the seat on the two cab models is illustrated in Figure 31 and Figure 32. Figure 31 shows that the seat causes an increase in cab natural frequency and a decrease in magnitude of response in the linear system. The seat's effect on the nonlinear model is small as seen in Figure 32. Both the natural frequency and magnitude of response increase slightly.

The high frequency seat-cab systems were also analyzed. The responses of the Standard 95 seat with the linear and nonlinear high frequency (3.7 Hz) conventional cabs are shown in Figure 33. The input amplitude for the linear cab system is one inch peak-to-peak, and the input amplitude for the nonlinear cab system is 1.5 inches peak-to-peak. The magnitude of response of the seat in the linear system is decreased, but the natural frequency is increased. The magnitude of response of the seat in the nonlinear system is greatly increased. The response shows two resonance zones. The first zone corresponds to the
Figure 28. Comparison of linear low frequency (1.7 Hz) conventional tractor model with and without the Standard 95 model.
Figure 29. Comparison of nonlinear low frequency (1.7 Hz) conventional tractor model with and without the Standard 95 model.
Figure 30. Comparison of Standard 95 responses in linear and nonlinear low frequency (1.7 Hz) COE tractor systems.
Figure 31. Comparison of linear low frequency (1.7 Hz) COE tractor model with and without the Standard 95 model.
Figure 32. Comparison of nonlinear low frequency (1.7 Hz) COE tractor model with and without the Standard 95 model.
Figure 33. Comparison of Standard 95 responses in linear and nonlinear high frequency (3.7 Hz) conventional tractor systems.
natural frequency of the seat, while the second zone is due to the influence of the cab. Figure 34 shows the response of the nonlinear cab. The two reference curves in Figure 34 are for cab input amplitudes of one inch peak-to-peak and two inches peak-to-peak. The input amplitude for the cab system is between these two values. The magnitude of the response of the cab in the seat-cab system is greatly reduced when compared to the two inch peak-to-peak input response of the cab alone; the reduction is not as significant when compared to the one inch peak-to-peak input response of the cab alone. The natural frequency is between the two reference natural frequencies. The decrease in cab response coupled with the increased magnitude and two resonant zones of the seat response indicate that the seat is acting as a vibration absorber. It is important to note that the input amplitudes for the three systems shown in Figure 34 are not the same. The amplitude dependency of the cab makes comparisons difficult. The vibration absorbing effect is not seen in the linear system. Figure 35 shows that the linear cab causes a reduction in seat response not an increase as seen in the nonlinear system. Figure 35 shows that the magnitude of response of the linear is reduced, but the natural frequency is only slightly affected by the seat.

Finally, the effects of the combining the Standard 95 with the linear and nonlinear high frequency (3.7 Hz) COE cab models were studied. The input amplitude for both cab systems is two inches peak-to-peak. Figure 36 shows that the magnitude of response of the seat in the linear system is reduced, and the magnitude of response in the nonlinear system is slightly elevated near 2.5 Hz. Though the responses of both seat do indicate two resonance zones, the seats do not act as vibration absorbers. The amplitude input to the seats below the cab natural frequency is significantly reduced, and as seen previously, input amplitude has a strong
Figure 34. Comparison of nonlinear high frequency (3.7 Hz) conventional tractor model with and without the Standard 95 model.
Figure 35. Comparison of linear high frequency (3.7 Hz) conventional tractor model with and without the Standard 95 model.
Figure 36. Comparison of Standard 95 responses in linear and nonlinear high frequency (3.7 Hz) COE tractor systems.
effect on seat response. This reduced amplitude causes the reduced response at the seat natural frequency. Because the seat input increases near the cab natural frequency, the response of the seat also increases forming the second resonance zone. The cab responses are shown in Figures 37 and 38. The responses of these cabs are very similar to their low frequency counterparts. The effect of adding the Standard 95 model to the cab models is summarized in Table 5.

Table 5. Effects of Standard 95 model on cab model responses.

<table>
<thead>
<tr>
<th>Cab</th>
<th>Linear Model</th>
<th>Nonlinear Model</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>% Increase in</td>
<td>% Increase in</td>
</tr>
<tr>
<td></td>
<td>Natural Frequency</td>
<td>Magnitude</td>
</tr>
<tr>
<td>Low Frequency</td>
<td>5.8</td>
<td>-5.3</td>
</tr>
<tr>
<td>Conventional</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Low Frequency</td>
<td>5.8</td>
<td>-11.8</td>
</tr>
<tr>
<td>COE</td>
<td></td>
<td></td>
</tr>
<tr>
<td>High Frequency</td>
<td>8.1</td>
<td>-18.4</td>
</tr>
<tr>
<td>Conventional</td>
<td></td>
<td></td>
</tr>
<tr>
<td>High Frequency</td>
<td>8.1</td>
<td>-9.9</td>
</tr>
<tr>
<td>COE</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

b. Easyaire 95-Cab System Models

The Easyaire 95 model was also added to the eight cab models, and the resulting systems were analyzed. In Figure 39, the seat responses for the seat alone and the seat added to the linear and nonlinear low frequency (1.7 Hz) conventional cab models are displayed. The input amplitude to both cabs is one inch peak-to-peak. The responses of the seats when combined with the two cab models is similar, indicating that the seat reacts the same to the
Figure 37. Comparison of linear high frequency (3.7 Hz) COE tractor model with and without the Standard 95 model.
Figure 38. Comparison of nonlinear high frequency (3.7 Hz) COE tractor model with and without the Standard 95 model.
Figure 39. Comparison of Easyaire 95 responses in linear and nonlinear low frequency (1.7 Hz) conventional tractor systems.
nonlinear conventional cab as it does to the linear cab. The natural frequency of the seat is increased in the seat-cab system, and the magnitude of response is reduced.

The effects of the seat on the responses of the two low frequency conventional cab models are shown in Figures 40 and 41. In both the linear and the nonlinear seat-cab systems, the natural frequency of the cab is slightly reduced. The magnitude of response of the cabs is also reduced. The reduction is more significant in the linear cab system. The seat reaction to the linear low frequency (1.7 Hz) COE cab model is different from the seat reaction to the nonlinear cab model as illustrated in Figure 42. The input amplitude to the nonlinear cab system is one inch peak-to-peak, and the input amplitude to the linear cab system is two inches peak-to-peak. The nonlinear cab system has a smaller input amplitude to prevent the seat from impacting the top and bottom bumpers which occurred during analysis with the larger input amplitude. In the linear cab system, the seat natural frequency increases and the magnitude of response increases. In the nonlinear cab system, the seat natural frequency decreases and the magnitude of response of the seat increases.

The linear and nonlinear COE cab responses are shown in Figures 43 and 44. The seat has a stronger effect on the response of the nonlinear cab than on the response of the linear cab. The natural frequency of the linear cab increases slightly and the magnitude of response decreases slightly when the seat is added. The magnitude of response of the nonlinear cab decreases when the seat is added, but the natural frequency is not affected.

In Figure 45, the effect of the linear and nonlinear high frequency (3.7 Hz) conventional cabs on the Easyaire 95 is displayed. The input amplitude is one inch peak-to-peak for both seat-cab systems. In both systems, the magnitude of seat response is
Figure 40. Comparison of linear low frequency (1.7 Hz) conventional tractor model with and without the Easyaire 95 model.
Figure 41. Comparison of nonlinear low frequency (1.7 Hz) conventional tractor model with and without the Easyaire 95 model.
Figure 42. Comparison of Easyaire 95 responses in linear and nonlinear low frequency (1.7 Hz) COE tractor systems.
Figure 43. Comparison of linear low frequency (1.7 Hz) COE tractor model with and without the Easyaire 95 model.
Figure 44. Comparison of nonlinear low frequency (1.7 Hz) COE tractor model with and without the Easyaire 95 model.
Figure 45. Comparison of Easyaire 95 responses in linear and nonlinear high frequency (3.7 Hz) conventional tractor systems.
increased. The natural frequency of the seat in both systems is slightly reduced, but the reduction is larger in the linear system than in the nonlinear system. The seat effects on the cab responses are shown in Figures 46 and 47. The seat causes a decrease in the magnitude of response of the linear cab. There is also a slight decrease in cab natural frequency. The effect of the seat on the nonlinear cab is more pronounced. The magnitude of response of the nonlinear cab decreases, but the natural frequency increases.

Finally, the effects of adding the Easyaire 95 to the linear and nonlinear high frequency (3.7 Hz) cab models were studied. The input amplitude in both systems is two inches peak-to-peak. The responses of the seat are shown in Figure 48. The magnitude of the response of the seat in the linear system is reduced, while the magnitude of the seat response in the nonlinear system is increased. The addition of the nonlinear cab causes a reduction in seat natural frequency, but addition of the linear system causes an increase in seat natural frequency. The cab responses are shown in Figures 49 and 50. The natural frequency of both cabs is reduced. However, adding the seat to the cab models causes a slight increase in the magnitude of response of the nonlinear cab, but a decrease in the magnitude of response of the linear cab. The effects of the Easyaire 95 on the cab models is summarized in Table 6.
Figure 46. Comparison of linear high frequency (3.7 Hz) conventional tractor model with and without the Easyaire 95 model.
Figure 47. Comparison of nonlinear high frequency (3.7 Hz) conventional tractor model with and without the Easyaire 95 model.
Figure 48. Comparison of Easyaire 95 responses in linear and nonlinear high frequency (3.7 Hz) COE tractor systems.
Figure 49. Comparison of linear high frequency (3.7 Hz) COE tractor model with and without the Easyaire 95 model.
Figure 50. Comparison of nonlinear high frequency (3.7 Hz) COE tractor model with and without the Easaire 95 model.
Table 6. Effects of Easyaire 95 model on cab model responses.

<table>
<thead>
<tr>
<th>Cab</th>
<th>Linear Model</th>
<th>Nonlinear Model</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>% Increase in Natural Frequency</td>
<td>% Increase in Magnitude</td>
</tr>
<tr>
<td>Low Frequency Conventional</td>
<td>-11.8</td>
<td>-17.9</td>
</tr>
<tr>
<td>Low Frequency COE</td>
<td>5.8</td>
<td>-10.2</td>
</tr>
<tr>
<td>High Frequency Conventional</td>
<td>8.1</td>
<td>-26.6</td>
</tr>
<tr>
<td>High Frequency COE</td>
<td>8.1</td>
<td>-11.2</td>
</tr>
</tbody>
</table>
CHAPTER VI

CONCLUSIONS

The National Seating Standard 95 and Easyaire 95 seat models, developed in ADAMS, were verified by experimental data. The initial interaction between these seat models and various cab models was investigated. However, because the cab models were not verified, only general trends of the seat-cab systems could be observed.

A. Seat Models

Both the Standard 95 and Easyaire 95 ADAMS models were verified. Experimental testing showed that the Standard 95 has a natural frequency of 1.95 Hz and a corresponding magnification factor of 1.438. The ADAMS models of the Standard 95 has a natural frequency of 1.9 Hz with a corresponding magnification factor of 1.635. The computationally determined natural frequency is 2.5% lower than the experimentally determined natural frequency. The computationally determined magnification factor is 13.5% greater than the experimentally determined one. The input amplitude effect on the model response was investigated. In the Standard 95 model, increase in input amplitude cause increases in the magnification factor and decreases in the model natural frequency.

The experimentally determined natural frequency of the Easyaire 95 is 2.74 Hz. The corresponding magnification factor is 2.266. The computationally determined natural frequency, 2.8 Hz, is 2.2% greater than the experimentally determined natural frequency. The computationally determined magnification factor, 2.37, is 4.5% greater than the experimentally determined factor. The effect of the input amplitude on the response of the
Easyaire 95 model was also determined. The magnification factor of the model response increase with increasing input amplitude. The model natural frequency decreases with increasing amplitude, but, as amplitude continues to increase, the natural frequency also increases.

**B. Cab Models**

A total of eight cab models were developed. Four of these models contain linear suspensions, while the remaining four contain nonlinear suspensions. Two geometries, conventional and COE, which represent the current commercial cab geometries, are used. In addition to linear and nonlinear suspensions, each geometry has a low frequency and a high frequency suspension.

The responses of the eight cab models were investigated. The natural frequency of the low frequency models is 1.7 Hz. The magnification factors vary from 3.29 to 4.11 depending on both model geometry and suspension type. The natural frequency of the high frequency models is 3.7 Hz. The corresponding magnification factors which range from 3.35 to 4.03 also depend on model geometry and suspension type. The nonlinear conventional cab models the lowest magnification factors, and the linear COE cab models have the greatest magnification factors.

**C. Seat-Cab Models**

The Standard 95 model was added to the cab models, and the combined systems were analyzed. When combined with the linear and nonlinear low frequency (1.7 Hz) conventional
cab models, the natural frequency of the seat model decreases with a corresponding increase in response magnitude. The magnitude of response is greater in the linear cab system than in the nonlinear system, but the natural frequency is lower in the nonlinear cab system. When the seat model is added to the linear and nonlinear low frequency (1.7 Hz) COE cab models, natural frequency of the seat model decreases slightly and the magnitude of response also decreases. There is little difference between the responses of the seat model in the linear cab system and the nonlinear cab system. There is a significant difference in the responses of the seat model when it is added to the linear and nonlinear high frequency (3.7 Hz) conventional models. In the linear cab system, the natural frequency of the seat model increases and the magnitude of response decreases. In the nonlinear cab system, the response of the seat model indicates two resonance zones and the magnitude of response is greatly increased. This change in seat response combined with the decrease in the nonlinear cab model response indicates that the seat is acting as a vibration absorber. When combined with the linear and nonlinear high frequency (3.7 Hz) COE cab models, the response of the seat again shows two resonance zones. However, the magnitude of the seat response is only slightly increased in the nonlinear system and decreased in the linear system, indicating that the seat is not acting as a vibration absorber.

The natural frequencies of the linear cab models increase when the seat is added. The corresponding magnitude of response decreases. The strongest decrease occurs in the high frequency (3.7 Hz) conventional cab model. The natural frequencies of the nonlinear conventional cab models decrease, with a corresponding decrease in magnitude, when the Standard 95 model is added. However, the natural frequencies of the nonlinear COE cab

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models increase with virtually no change in magnitude of response.

The Easyaire 95 model was combined with the existing cab models, and the seat-cab models were analyzed. The magnitude of response of the seat model, when combined with the linear and nonlinear low frequency (1.7 Hz) conventional cab models, is significantly decreased. The seat model natural frequency is significantly increased. When combined with the linear and nonlinear low frequency (1.7 Hz) COE cab models, the magnitude of response is increased. The seat model natural frequency increases in the linear cab system, but the natural frequency decreases in the nonlinear system. When the Easyaire 95 model is added to the linear and nonlinear high frequency (3.7 Hz) conventional cab models, the seat model natural frequency decreases slightly with an increase in magnitude of response. Addition of the seat model to the linear and nonlinear high frequency (3.7 Hz) COE cab models causes a decrease in magnitude of seat response in the linear system. The magnitude of seat response increases in the nonlinear system. In the linear cab system, the seat natural frequency increases. However, in the nonlinear system, the seat natural frequency decreases.

When combined with the Easyaire 95 model, the magnitudes of the responses of the linear cab models decrease. The natural frequencies of all the models except the low frequency conventional cab model increase. The magnitudes of the responses of all the nonlinear models except the high frequency COE cab model decrease when the seat model is added to the system. There is no significant change in the magnitude of the high frequency COE cab model response. The natural frequency of the models also decrease except in the high frequency COE cab model. The decrease in natural frequency and magnitude of response is partially due to the reduction in input amplitude.
D. Summary

The responses of the Standard 95 and Easyaire 95 models developed in ADAMS compared favorably with the experimentally determined seat responses. The seat-cab system investigations indicate that generally the seat response is slightly increased when the seat is combined with a cab. The results of these investigations indicate that the Easyaire 95 should be used in combination with the conventional cabs. The magnitude of response of the Standard 95 model in the low frequency conventional cab systems is greater than the magnitude of response of the Easyaire 95 model. In the nonlinear high frequency conventional cab system, the Standard 95 acts as a vibration absorber. While combining the seats with the COE cabs reduces the magnitude of response of Standard 95, it increases the magnitude in the Easyaire 95. Therefore, the Standard 95 should be used in combination with COE cabs. The computational results will only improve with improvements in the ADAMS models.
CHAPTER VII
RECOMMENDATIONS

To enhance the research conducted in this thesis, there are several areas that can be improved. These areas fall within two main categories, experimental testing and computer modeling. These sections address concerns and improvements for future work in computer simulation of seat-cab systems.

A. Experimental Testing

The most significant problem with the computer models developed in this thesis is the lack of verification of the cab models. It is recommended that a basic experimental cab model is developed and tested. A steel plate with the suspension located at one end and the other end hinged to the ground would be a sufficient experimental model. To match the computer model, the input motion should be applied directly to the suspension and a canister weight representing the seat should be applied at the correct seat location. The response should be measured at both the seat location and the suspension end of the model. Specific experimental model characteristics including mass, center of mass, and inertial properties, should be incorporated into the computer model.

In the current research, the driver is modeled as a lump mass. Past research has shown that the contribution of the driver to the seat response can be significant (Fairley, 130). To determine the driver effect on National Seating seats, testing of the seats with a driver should be performed. This would require equipment that meets the ISO criteria for vibration testing using human subjects.
In addition to testing the seats with a driver, more extensive tests on the seats alone need to be performed to determine the input amplitude dependency of the seats. The effect of input amplitude on the computer models was investigated but was not substantiated by experimental results. Also, road testing of the seat and cab should be conducted. This would provide a better understanding of the interaction between the seat and cab, as well as provide data to compare to the seat-cab models.

**B. Computer Modeling**

The ADAMS seat models can be improved in a number of ways. First, using CAD drawings of the seat insures that the geometry and inertial properties of the models are correct. Also, correct location of the center of mass of each part, especially the upper seat base, is extremely important. This became apparent during model analysis; the response of the seat changed when the center of mass location of the upper base was modified. Using CAD information would help insure correct center of mass locations. Once the center of mass is defined, the mass of each part should be accurately determined, not estimated. Finally, if experimental testing indicated that the effect of the driver is significant, a driver model should be added to the existing seat models using ADAMS/Android.

The current seats are modeled using rigid parts, but experimental testing has shown that the seat frame is very flexible and undergoes bending during seat motion. To accurately model this phenomenon in ADAMS, the parts in the models need to be constructed from flexible elements or the forces in the active elements, like the shock and spring, need to be modified to represent the actual force values during seat motion.
The National Seating Standard 96 and Easyaire 96 seats should be modeled and verified. Once verification has occurred, the model 96 seats should be added to the cab models, and the system responses investigated. The improvements previously mentioned for the model 95 seat models should be incorporated into the Standard and Easyaire 96 seat models.

The cab models should also be improved to provide a more accurate measure of the seat-cab interaction. The current ADAMS cab model is very basic, allowing only one system input and one suspension. A more complex model should be developed which includes the multiple inputs that occur in tractors. The improved models may also include the multiple suspensions found in tractors with suspended cabs.
BIBLIOGRAPHY
BIBLIOGRAPHY


APPENDIXES
APPENDIX A

STANDARD 95 ADAMS MODEL

!------------------------------------------------ Default Units for Model ----------------------------------------------------!

defaults units &
   length = inches &
   angle = deg &
   force = pound_force &
   mass = slug &
   time = sec

defaults units &
   coordinate_system_type = cartesian &
   orientation_type = body313

!------------------------------------------------ Default Attributes for Model -----------------------------------------------!

defaults attributes &
   inheritance = bottom_up &
   icon_visibility = on &
   grid_visibility = off &
   size_of_icons = 1.0 &
   spacing_for_grid = 1.0

!------------------------------------------------ ADAMS/View Model --------------------------------------------------------!

model create &
   model_name = e95

!------------------------------------------------ Rigid Parts -------------------------------------------------------------!

Create parts and their dependent markers and graphics

!------------------------------------------------ ground ---------------------------------------------------------------!

***** Ground Part *****

defaults model &
   part_name = ground

100
defaults coordinate_system &
default_coordinate_system = /e95/ground

****** Markers for current part ******

marker create &
  marker_name = /e95/ground/mar3 &
  adams_id = 3 &
  location = 0.0, 4.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

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  marker_name = /e95/ground/mar4 &
  adams_id = 4 &
  location = 0.0, 8.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/ground/mar48 &
  adams_id = 48 &
  location = 0.0, 0.0, 0.0 &
  orientation = 0.0d, 90.0d, 0.0d

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  marker_name = /e95/ground/mar50 &
  adams_id = 50 &
  location = 0.0, 0.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/ground/mar52 &
  adams_id = 52 &
  location = 20.0, -2.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/ground/mar53 &
  adams_id = 53 &
  location = 0.0, -2.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/ground/ref &
  adams_id = 62 &
  location = 6.5, 10.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

****** Graphics for current part ******

geometry create shape block &
  block_name = /e95/ground/BOX13 &
  adams_id = 2 &
corner_marker = /e95/ground/mar53 &
diag_corner_coords = 20.0, 2.0, 1.0

model display &
model_name = e95

!------------------------------------------------------------- LBASE -------------------------------------------------------------!

defaults coordinate_system &
  default_coordinate_system = /e95/ground

part create rigid_body name_and_position &
  part_name = /e95/LBASE &
  adams_id = 2 &
  location = 0.0, 0.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

defaults coordinate_system &
  default_coordinate_system = /e95/LBASE

****** Markers for current part ******

marker create &
  marker_name = /e95/LBASE/mar1 &
  adams_id = 1 &
  location = 0.0, 0.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

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  marker_name = /e95/LBASE/mar1 &
  visibility = off

marker create &
  marker_name = /e95/LBASE/spring &
  adams_id = 9 &
  location = 6.5, 4.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/LBASE/mar23 &
  adams_id = 23 &
  location = 0.0, 8.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/LBASE/mar25 &
  adams_id = 25 &
  location = 0.0, 4.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

102
marker create &
marker_name = /e95/LBASE/shock &
adams_id = 31 &
location = 0.0, 0.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
marker_name = /e95/LBASE/cm &
adams_id = 34 &
location = 6.25, 5.976, 0.5 &
orientation = 43.42966702d, 0.0d, 0.0d

marker create &
marker_name = /e95/LBASE/mar45 &
adams_id = 45 &
location = 0.0, 0.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
marker_name = /e95/LBASE/mar47 &
adams_id = 47 &
location = 0.0, 0.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
marker_name = /e95/LBASE/mar49 &
adams_id = 49 &
location = 0.0, 0.0, 0.0 &
orientation = 0.0d, 90.0d, 0.0d

marker create &
marker_name = /e95/LBASE/mar51 &
adams_id = 51 &
location = 0.0, 0.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
marker_name = /e95/LBASE/UBUMP &
adams_id = 55 &
location = 12.5, 7.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
marker_name = /e95/LBASE/LBUMP &
adams_id = 57 &
location = 12.5, 1.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
marker_name = /e95/LBASE/LDM &
adams_id = 58 &
location = 12.5, 2.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
marker_name = /e95/LBASE/shock2 &
adams_id = 60 &
location = 3.0, 6.75, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
marker_name = /e95/LBASE/ref &
adams_id = 61 &
location = 6.5, 10.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

part create rigid_body mass_properties &
part_name = /e95/LBASE &
mass = 0.1 &
center_of_mass_marker = cm &
ixx = 16.28683646 &
iyy = 12.61107955 &
izz = 28.71542614 &
ixy = 0.0 &
izx = 0.0 &
iyz = 0.0

****** Graphics for current part *******

geometry create shape block &
block_name = /e95/LBASE/BOX14 &
adams_id = 14 &
corner_marker = /e95/LBASE/LBUMP &
diag_corner_coords = 1.0, -1.0, 1.0

geometry create shape block &
block_name = /e95/LBASE/BOX16 &
adams_id = 16 &
corner_marker = /e95/LBASE/UBUMP &
diag_corner_coords = 1.0, 1.0, 1.0

defaults coordinate_system &
default_coordinate_system = /e95/LBASE/marl

geometry create shape extrusion &
extrusion_name = /e95/LBASE/EXT1 &
reference_marker = /e95/LBASE/marl &
points_for_profile = 0.0, 0.0, 0.0 &
, 0.0, 4.0, 0.0 &
, 0.0, 8.0, 0.0 &
, 3.0, 8.0, 0.0 &
, 10.0, 2.0, 0.0 &
, 12.5, 2.0, 0.0 &
, 12.5, 0.0, 0.0 &
, 0.0, 0.0, 0.0 &
length_along_z_axis = 1.0

defaults coordinate system &
default_coordinate_system = /e95/LBASE

model display &
model_name = e95

!---------------------------------------------------------- UBASE ----------------------------------------------------------!

defaults coordinate_system &
default_coordinate_system = /e95/ground

part create rigid_body name_and_position &
part_name = /e95/UBASE &
adams_id = 3 &
location = 0.0, 0.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

defaults coordinate_system &
default_coordinate_system = /e95/UBASE

***** Markers for current part *****

marker create &
marker_name = /e95/UBASE/mar2 &
adams_id = 2 &
location = 10.0, 4.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker attributes &
marker_name = /e95/UBASE/mar2 &
visibility = off

marker create &
marker_name = /e95/UBASE/bump &
adams_id = 5 &
location = 12.5, 4.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
marker_name = /e95/UBASE/mar6 &
adams_id = 6 &
location = 12.5, 8.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
marker_name = /e95/UBASE/spring &
adams_id = 10 &
location = 6.5, 10.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/UBASE/mar16 &
adams_id = 16 &
  location = 0.0, 0.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/UBASE/mar18 &
adams_id = 18 &
  location = 0.0, 0.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/UBASE/mar27 &
adams_id = 27 &
  location = 12.5, 8.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/UBASE/mar29 &
adams_id = 29 &
  location = 12.5, 4.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/UBASE/mar38 &
adams_id = 38 &
  location = 0.0, 12.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/UBASE/seat2 &
adams_id = 39 &
  location = 10.5, 14.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/UBASE/seat3 &
adams_id = 40 &
  location = 12.5, 26.5, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/UBASE/seat4 &
adams_id = 41 &
  location = 14.5, 26.5, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
marker_name = /e95/UBASE/seat5 &
adams_id = 42 &
location = 12.5, 14.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/UBASE/seat1 &
adams_id = 43 &
location = 0.0, 14.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/UBASE/mar44 &
adams_id = 44 &
location = 12.5, 12.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/UBASE/mar46 &
adams_id = 46 &
location = 0.0, 0.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/UBASE/mar54 &
adams_id = 54 &
location = 10.0, 4.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/UBASE/UDM &
adams_id = 59 &
location = 12.5, 2.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

part create rigid_body mass_properties &
  part_name = /e95/ubase &
mass = 6.5 &
center_of_mass_marker = cm &
iwx = 0.0 &
iwy = 0.0 &
iwz = 0.0 &
ixz = 0.0 &
iyz = 0.0 &

****** Graphics for current part ******

geometry create curve outline &
  outline_name = /e95/UBASE/OUTLINE &
adams_id = 12 &
marker_name = /e95/UBASE/mar38, &
```
/e95/UBASE/seat1, &
/e95/UBASE/seat2, &
/e95/UBASE/seat3, &
/e95/UBASE/seat4, &
/e95/UBASE/seat5, &
/e95/UBASE/mar44, &
/e95/UBASE/mar38 &
close = no

geometry create shape block &
    block_name = /e95/UBASE/BOX15
    adams_id = 15 &
    corner_marker = /e95/UBASE/bump &
    diag_corner_coords = 1.0, 0.5, 1.0

geometry create shape extrusion &
    extrusion_name = /e95/UBASE/EXT2 &
    reference_marker = /e95/LBASE/mar54 &
    points_for_profile = 0.0, 0.0, 0.0 &
                        , 2.5, 0.0, 0.0 &
                        , 2.5, 8.0, 0.0 &
                        , -10.0, 8.0, 0.0 &
                        , -10.0, 6.0, 0.0 &
                        , -7.0, 6.0, 0.0 &
                        , 0.0, 0.0, 0.0 &
    length_along_z_axis = 1.0

defaults coordinate_system &
    default_coordinate_system = /e95/UBASE

model display &
    model_name = e95

!-------------------------------------------------------------------------- LCL --------------------------------------------------------------------------!

defaults coordinate_system &
    default_coordinate_system = /e95/ground

part create rigid_body name_and_position &
    part_name = /e95/LCL &
    adams_id = 4 &
    location = 0.0, 0.0, 0.0 &
    orientation = 0.0d, 0.0d, 0.0d

defaults coordinate_system &
    default_coordinate_system = /e95/LCL

***** Markers for current part *****

marker create &
    marker_name = /e95/LCL/mar7 &
```
adams_id = 7 &
location = 0.0, 4.0, 0.0 &
orientation = 0.0d, 0.0d, 0.0d

marker attributes &
  marker_name = /e95/LCL/mar7 &
  visibility = off

marker create &
  marker_name = /e95/LCL/mar8 &
  adams_id = 8 &
  location = 12.5, 4.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

marker attributes &
  marker_name = /e95/LCL/mar8 &
  visibility = off

marker create &
  marker_name = /e95/LCL/mar17 &
  adams_id = 17 &
  location = 0.0, 0.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/LCL/mar19 &
  adams_id = 19 &
  location = 0.0, 0.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/LCL/mar24 &
  adams_id = 24 &
  location = 0.0, 4.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/LCL/mar28 &
  adams_id = 28 &
  location = 12.5, 4.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/LCL/cm &
  adams_id = 35 &
  location = 6.25, 4.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

part create rigid_body mass_properties &
  part_name = /e95/LCL &
  mass = 9.2150614e-02 &
  center_of_mass = cm &
ixx = 1.47728234e-02 &
yyy = 1.397339914 &
izz = 1.406113349 &
ixy = 0.0 &
izx = 0.0 &
ysz = 0.0

****** Graphics for current part *****

geometry create shape link &
  link_name = /e95/UCL/LNK3 &
  i_marker = /e95/UCL/mar7 &
  j_marker = /e95/UCL/mar8 &
  width = 0.9842519685 &
  depth = 31.000062

model display &
  model_name = e95

!------------------------------------------------------------------ UCL ------------------------------------------!

defaults coordinate_system &
  default_coordinate_system = /e95/ground

part create rigid_body name_and_position &
  part_name = /e95/UCL &
  adams_id = 5 &
  location = 0.0, 0.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

defaults coordinate_system &
  default_coordinate_system = /e95/UCL

****** Markers for current part *****

marker create &
  marker_name = /e95/UCL/shock &
  adams_id = 11 &
  location = 0.0, 8.0, 0.0 &
  orientation = 0.0d, 0.0d, 0.0d

marker create &
  marker_name = /e95/UCL/marl2 &
  adams_id = 12 &
  location = 0.0, 8.0, 0.0 &
  orientation = 326.3099325d, 0.0d, 0.0d

marker attributes &
  marker_name = /e95/UCL/marl2 &
  visibility = off
marker create &
    marker_name = /e95/UCL/marl3 &
    adams_id = 13 &
    location = 3.0, 6.0, 0.0 &
    orientation = 326.3099325d, 0.0d, 0.0d

marker attributes &
    marker_name = /e95/UCL/marl3 &
    visibility = off

marker create &
    marker_name = /e95/UCL/marl4 &
    adams_id = 14 &
    location = 3.0, 6.0, 0.0 &
    orientation = 11.88865804d, 0.0d, 0.0d

marker attributes &
    marker_name = /e95/UCL/marl4 &
    visibility = off

marker create &
    marker_name = /e95/UCL/marl5 &
    adams_id = 15 &
    location = 12.5, 8.0, 0.0 &
    orientation = 11.88865804d, 0.0d, 0.0d

marker attributes &
    marker_name = /e95/UCL/marl5 &
    visibility = off

marker create &
    marker_name = /e95/UCL/marl22 &
    adams_id = 22 &
    location = 0.0, 8.0, 0.0 &
    orientation = 0.0d, 0.0d, 0.0d

marker create &
    marker_name = /e95/UCL/marl26 &
    adams_id = 26 &
    location = 12.5, 8.0, 0.0 &
    orientation = 0.0d, 0.0d, 0.0d

marker create &
    marker_name = /e95/UCL/cm &
    adams_id = 36 &
    location = 7.445437247, 7.0, 0.0 &
    orientation = 9.685753035d, 0.0d, 0.0d

part create rigid_body mass_properties &
    part_name = /e95/UCL &
    mass = 4.53822946e-02 &
    center_of_mass = cm &
ixx = 8.4095315e-03 & iy = 0.475632153 & izz = 0.4823343518 & ix = 0.0 & iy = 0.0 & iz = 0.0

****** Graphics for current part ******

geometry create shape link &
   link_name = /e95/UCL/LNK3 &
   i_marker = /e95/UCL/marl2 &
   j_marker = /e95/UCL/marl3 &
   width = 0.9842519685 &
   depth = 38.7500775

geometry create shape link &
   link_name = /e95/UCL/LNK3_2 &
   i_marker = /e95/UCL/marl4 &
   j_marker = /e95/UCL/marl5 &
   width = 0.9842519685 &
   depth = 9.9200198386

model display &
   model_name = e95

!------------------------------------------------- Joints ---------------------------------------------------!

constraint create joint revolute &
   joint_name = /e95/JOI1 &
   adams_id = 1 &
   i_marker_name = /e95/UCL/mar22 &
   j_marker_name = /e95/LBASE/mar23

constraint create joint revolute &
   joint_name = /e95/JOI2 &
   adams_id = 2 &
   i_marker_name = /e95/LCL/mar24 &
   j_marker_name = /e95/LBASE/mar25

constraint create joint revolute &
   joint_name = /e95/JOI3 &
   adams_id = 3 &
   i_marker_name = /e95/UCL/mar26 &
   j_marker_name = /e95/UBASE/mar27

constraint create joint revolute &
   joint_name = /e95/JOI4 &
   adams_id = 4 &
   i_marker_name = /e95/UCL/mar28 &
constraint create joint translational &
joint_name = /e95/JOI6 &
adams_id = 6 &
i_marker_name = /e95/LBASE/mar49 &
j_marker_name = /e95/ground/mar48

!---------------------------------------------------------------------------------------- Forces!----------------------------------------------------------------------------------------!

force create direct single_component_force &
single_component_force_name = /e95/spring_rate &
adams_id = 1 &
type_of_freedom = translational &
i_marker_name = /e95/LBASE/spring &
j_marker_name = /e95/UBASE/spring &
action_only = off &
function = ""

force create direct single_component_force &
single_component_force_name = /e95/spring_damp &
adams_id = 2 &
type_of_freedom = translational &
i_marker_name = /e95/LBASE/spring &
j_marker_name = /e95/UBASE/spring &
action_only = off &
function = ""

force create direct single_component_force &
single_component_force_name = /e95/shock_pos &
adams_id = 4 &
type_of_freedom = translational &
i_marker_name = /e95/LBASE/shock &
j_marker_name = /e95/UCL/shock &
action_only = off &
function = ""

force create direct single_component_force &
single_component_force_name = /e95/LBUMP &
adams_id = 6 &
type_of_freedom = translational &
i_marker_name = /e95/UBASE/bump &
j_marker_name = /e95/LBASE/LBUMP &
action_only = off &
function = ""

force create direct single_component_force &
single_component_force_name = /e95/shock_trans &
adams_id = 8 &
type_of_freedom = translational &
i_marker_name = /e95/LBASE/shock &
j_marker_name = /e95/UCL/shock &
action_only = off &
function = ""

force create direct single_component_force &
single_component_force_name = /e95/shock_pos &
    adams_id = 9 &
type_of_freedom = translational &
i_marker_name = /e95/LBASE/shock &
j_marker_name = /e95/UCL/shock &
    action_only = off &
function = ""

force create direct single_component_force &
single_component_force_name = /e95/ubump &
    adams_id = 10 &
type_of_freedom = translational &
i_marker_name = /e95/UBASE/bump &
j_marker_name = /e95/LBASE/LBUMP &
    action_only = off &
function = ""

!----------------------------------------------------------------------
Motions
----------------------------------------------------------------------!

constraint create motion_generator &
motion_name = /e95/mot1 &
    adams_id = 1 &
type_of_freedom = translational &
joint_name = /e95/J016 &
    function = ""

constraint modify motion_generator &
motion_name = /e95/mot1 &
    function = "SHF(time, 0.0, 1.0, 720D, 0.0, 0.0)"

force modify direct single_component_force &
single_component_force_name = spring_rate &
    function = "POLY(-DM(/e95/LBASE/spring, /e95/UBASE/spring)+6, 0.0, 440.6, 204.6, 66.5, 2.7, 
-6.94, 4.83) * STEP(VR(/e95/LBASE/spring, /e95/UBASE/spring), 110.0, 1.0, 110.1, 0.0)"

force modify direct single_component_force &
single_component_force_name = spring_damp &
    function = "-VR(/e95/LBASE/spring, /e95/UBASE/spring)*2.206"

force modify direct single_component_force &
single_component_force_name = shock_neg &
    function = "POLY(VM(/e95/LBASE/shock, /e95/UCL/shock), 0.0, -13.57, 20.25) * 
STEP(VR(/e95/LBASE/shock, /e95/UCL/shock), -1.1, 1.0, -1.0, 0.0) * STEP(DM(/e95/LBASE/shock2,
force modify direct single_component_force &
  single_component_force_name = lbump &
  function = "POLY(-abs(DY(/e95/UBASE/bump, /e95/LBASE/bump)), 0.0, 0.0, 35000) * STEP(DY(/e95/UBASE/bump, /e95/LBASE/bump), 0.0, 1.0, 0.05, 0.0) * STEP(VR(/e95/UBASE/bump, /e95/UCL/shock), -0.05, 0.0, 0.0, 1.0) + POLY(-abs(DY(/e95/UBASE/bump, /e95/LBASE/bump)), 0.0, 0.0, 523.8) * STEP(DY(/e95/UBASE/bump, /e95/LBASE/bump), 0.0, 1.0, 0.05, 0.0) * STEP(VR(/e95/UBASE/bump, /e95/UCL/shock), -0.05, 1.0, 0.05, 0.0)"

force modify direct single_component_force &
  single_component_force_name = shock_trans &
  function = "-POLY(VR(/e95/LBASE/shock, /e95/UCL/shock), 0.0, 87.82, 114.6) * STEP(VR(/e95/LBASE/shock, /e95/UCL/shock), 1.60, 1.0, 1.63, 0.0) * STEP(VR(/e95/LBASE/shock2, /e95/UCL/shock), 0.04, 0.0, 0.05, 1.0)"

force modify direct single_component_force &
  single_component_force_name = shock_pos &
  function = "-POLY(VM(/e95/LBASE/shock, /e95/UCL/shock), 0.0, 251, 14.4) * STEP(VR(/e95/LBASE/shock, /e95/UCL/shock), 1.60, 0.0, 1.63, 1.0) * STEP(DM(/e95/LBASE/shock2, /e95/UCL/shock), 0.04, 0.0, 0.05, 1.0)"

force modify direct single_component_force &
  single_component_force_name = ubump &
  function = "POLY(-abs(DY(/e95/UBASE/bump, /e95/LBASE/bump)), 0.0, 0.0, 35000) * STEP(DY(/e95/UBASE/bump, /e95/LBASE/bump), 0.0, 1.0, 0.05, 0.0) * STEP(VR(/e95/UBASE/bump, /e95/UCL/shock), -0.05, 0.0, 0.0, 1.0) + POLY(-abs(DY(/e95/UBASE/bump, /e95/LBASE/bump)), 0.0, 0.0, 35000) * STEP(DY(/e95/UBASE/bump, /e95/LBASE/bump), 0.0, 1.0, 0.05, 0.0) * STEP(VR(/e95/UBASE/bump, /e95/UCL/shock), -0.05, 1.0, 0.05, 0.0)"

!--------------------------------------------- Requests ---------------------------------------------!
i_marker_name = /e95/UBASE/spring &
j_marker_name = /e95/LBASE/ref &
output_type = displacement

!---------------------------------------------------------- Dynamic graphics ----------------------------------------------------------!

geometry create shape spring_damper &
    spring_damper_name = /e95/GSPR3 &
    adams_id = 3 &
i_marker_name = /e95/UBASE/spring &
j_marker_name = /e95/LBASE/spring &
    coil_count = 3 &
    diameter_of_spring = 2.0 &
    damper_diameter_at_ij = 0.5, 1.0 &
    tip_length_at_ij = 1.0, 2.0 &
    cup_length_at_ij = 2.0, 2.0

geometry create shape spring_damper &
    spring_damper_name = /e95/GSPR6 &
    adams_id = 6 &
i_marker_name = /e95/LBASE/MAR47 &
j_marker_name = /e95/UCL/shock &
    coil_count = 3 &
    diameter_of_spring = 2.0 &
    damper_diameter_at_ij = 0.5, 1.0 &
    tip_length_at_ij = 3.0, 3.0 &
    cup_length_at_ij = 3.0, 3.0

!---------------------------------------------------------- Accgrav ----------------------------------------------------------!

force create body gravitational &
    gravity_field_name = ACC1 &
    x_component_gravity = 0.0 &
    y_component_gravity = -386.0 &
    z_component_gravity = 0.0

!---------------------------------------------------------- Analysis settings ----------------------------------------------------------!

output_control set output &
    model_name = e95 &
    reqsave = on &
grsave = on

output_control set results &
    model_name = e95 &
    formatted = on

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APPENDIX B

ADAMS FORCES

Standard 95
Object Name  /s95/Spring
Object Type  : Single Component Force
Parent Type  : Model
Adams ID : 1
I Marker  /s95/LCL/MAR9
J Marker  /s95/UBASE/MAR10
Length : None
Mode : Translational
Actiononly : False
Type : Bidirectional
Function : poly(-DM(/s95/LCL/MAR9, /s95/UBASE/MAR10)+6, 0.0, 440.6, 204.6, 66.5, 2.7, -6.94, 4.83)

Object Name  /s95/Spring-Damping
Object Type  : Single Component Force
Parent Type  : Model
Adams ID : 2
I Marker  /s95/LCL/MAR9
J Marker  /s95/UBASE/MAR10
Length : None
Mode : Translational
Actiononly : False
Type : Bidirectional
Function : -VR(/s95/LCL/MAR9, /s95/UBASE/MAR10)*2.206

Object Name  /s95/Shock-Neg
Object Type  : Single Component Force
Parent Type  : Model
Adams ID : 4
I Marker  /s95/LBASE/MAR31
J Marker  /s95/UCL/MAR11
Length : None
Mode : Translational
Actiononly : False
Type : Bidirectional
Function : poly(VM(/s95/LBASE/MAR31, /s95/UCL/MAR11), 0.0, -13.57, 20.25)*
          step(VR(/s95/LBASE/MAR31, /s95/UCL/MAR11), -1.1, 1, -1.0, 0)*
          step(DM(/s95/LBASE/MAR31, /s95/UCL/MAR11), 0.04, 0, 0.05, 1)

Object Name  /s95/Shock-Trans
Object Type  : Single Component Force
Parent Type  : Model
Adams ID : 8
I Marker  /s95/LBASE/MAR31
J Marker  /s95/UCL/MAR11
Length : None
Mode : Translational
Actiononly : False
Type : Bidirectional
Function : 
   -poly(VR(/s95/LBASE/MAR31, /s95/UCL/MAR11), 0.0, 87.82, 114.6)*
   step(VR(/s95/LBASE/MAR31, /s95/UCL/MAR11), -1.1, 0, -1.0, 1)*
   step(VR(/s95/LBASE/MAR31, /s95/UCL/MAR11), 1.6, 1, 1.63, 0)*
   step(DM(/s95/LBASE/MAR31, /s95/UCL/MAR11), 0.04, 0, 0.05, 1)

Object Name : /s95/Shock-Pos
Object Type : Single Component Force
Parent Type : Model
Adams ID : 9
I Marker : /s95/LBASE/MAR31
J Marker : /s95/UCL/MAR11
Length : None
Mode : Translational
Actiononly : False
Type : Bidirectional
Function : 
   -poly(VR(/s95/LBASE/MAR31, /s95/UCL/MAR11), 0.0, 251, 14.4)*
   step(VR(/s95/LBASE/MAR31, /s95/UCL/MAR11), 1.6, 0, 1.63, 1)*
   step(DM(/s95/LBASE/MAR31, /s95/UCL/MAR11), 0.04, 0, 0.05, 1)

Object Name : /s95/Lbump
Object Type : Single Component Force
Parent Type : Model
Adams ID : 6
I Marker : /s95/LBASE/MAR5
J Marker : /s95/UCL/MAR5
Length : None
Mode : Translational
Actiononly : False
Type : Bidirectional
Function : 
   poly(-ABS(DY(/s95/LBASE/MAR5, /s95/LBASE/LBUMP)), 0.0, 0.0, 763.263,
   -2117.02, 4074.6) * step(DY(/s95/LBASE/MAR5, /s95/LBASE/LBUMP), 0.0, 1, 0.05,
   0) * step(VR(/s95/LBASE/MAR5, /s95/LBASE/LBUMP), -0.05, 0, 0.0, 1)+ poly(-ABS
   (DY(/s95/LBASE/MAR5, /s95/LBASE/LBUMP)), 0.0, 0.0, 523.8) *
   step(DY(/s95/LBASE/MAR5, /s95/LBASE/LBUMP), 0.0, 1, 0.05, 0) *
   step(VR(/s95/LBASE/MAR5, /s95/LBASE/LBUMP), -0.05, 0, 0.0, 1)

**Linear Low Frequency COE Cab**

Object Name : /lfsh/Spring
Object Type : Single Component Force
Parent Type : Model
Adams ID : 1
I Marker : /lfsh/IN/SP
J Marker : /lfsh/FLOOR/MAR5
Length : None
Mode : Translational
Actiononly : False
Type : Bidirectional

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Function: poly(-DM(/lfst/IN/SP, /lfst/FLOOR/MAR5)+4, 0.0, 1206.38, 260)

Object Name: /lfst/Shock
Object Type: Single Component Force
Parent Type: Model
Adams ID: 2
I Marker: /lfst/FLOOR/MAR5
J Marker: /lfst/IN/SP
Length: None
Mode: Translational
Actiononly: False
Type: Bidirectional
Function: poly(-VR(/lfst/FLOOR/MAR5, /lfst/IN/SP), 0.0, 0.0, 6.5)

**Linear Low Frequency Conventional Cab**

Object Name: /lfst/Spring
Object Type: Single Component Force
Parent Type: Model
Adams ID: 1
I Marker: /lfst/IN/SP
J Marker: /lfst/FLOOR/MAR5
Length: None
Mode: Translational
Actiononly: False
Type: Bidirectional
Function: poly(-DM(/lfst/IN/SP, /lfst/FLOOR/MAR5)+4, 0.0, 1206.38, 260)

Object Name: /lfst/Shock
Object Type: Single Component Force
Parent Type: Model
Adams ID: 2
I Marker: /lfst/FLOOR/MAR5
J Marker: /lfst/IN/SP
Length: None
Mode: Translational
Actiononly: False
Type: Bidirectional
Function: poly(-VR(/lfst/FLOOR/MAR5, /lfst/IN/SP), 0.0, 0.0, 6.5)

**Linear High Frequency COE Cab**

Object Name: /hfsb/Spring
Object Type: Single Component Force
Parent Type: Model
Adams ID: 1
I Marker: /hfsb/IN/SP
J Marker: /hfsb/FLOOR/MAR5
Length: None
Mode: Translational
Actiononly: False
Type: Bidirectional
Function: poly(-DM(/hfsb/IN/SP, /hfsb/FLOOR/MAR5)+4, 0.0, 1101.34, 1300)
Object Name : /hfsb/Shock
Object Type : Single Component Force
Parent Type : Model
Adams ID : 2
I Marker : /hfsb/FLOOR/MAR5
J Marker : /hfsb/IN/SH
Length : None
Mode : Translational
Actiononly : False
Type : Bidirectional
Function : poly(-VR(/hfsb/FLOOR/MAR5, /hfsb/IN/SP), 0.0, 0.0, 14.5)

Linear High Frequency Conventional Cab
Object Name : /hfst/Spring
Object Type : Single Component Force
Parent Type : Model
Adams ID : 1
I Marker : /hfst/IN/SP
J Marker : /hfst/FLOOR/MAR5
Length : None
Mode : Translational
Actiononly : False
Type : Bidirectional
Function : poly(-DM(/hfst/IN/SP, /hfst/FLOOR/MAR5)+4, 0.0, 0.0, 1206.38, 1175)

Object Name : /hfst/Shock
Object Type : Single Component Force
Parent Type : Model
Adams ID : 2
I Marker : /hfst/FLOOR/MAR5
J Marker : /hfst/IN/SH
Length : None
Mode : Translational
Actiononly : False
Type : Bidirectional
Function : poly(-VR(/hfst/FLOOR/MAR5, /hfst/IN/SP), 0.0, 0.0, 13.5)

Nonlinear Low Frequency COE Cab
Object Name : /lfsb/Spring
Object Type : Single Component Force
Parent Type : Model
Adams ID : 1
I Marker : /lfsb/IN/SP
J Marker : /lfsb/FLOOR/MAR5
Length : None
Mode : Translational
Actiononly : False
Type : Bidirectional
Function : poly(-DM(/lfsb/IN/SP, /lfsb/FLOOR/MAR5)+4, 0.0, 0.0, 1101.34, 126.4, 41, 1.67, -4.28, 2.98)

Object Name : /lfsb/Shock

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<table>
<thead>
<tr>
<th>Object Name</th>
<th>Object Type</th>
<th>Parent Type</th>
<th>Adams ID</th>
<th>I Marker</th>
<th>J Marker</th>
<th>Length</th>
<th>Mode</th>
<th>Actiononly</th>
<th>Type</th>
<th>Function</th>
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</thead>
<tbody>
<tr>
<td>/lfst/Spring</td>
<td>Single Component Force</td>
<td>Model</td>
<td>1</td>
<td>/lfst/IN/SP</td>
<td>/lfst/FLOOR/MAR5</td>
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<td>poly(-DM(/lfst/IN/SP, /lfst/FLOOR/MAR5)+4, 0.0, 1206.38, 126.4, 41, 1.67, -4.28, 2.98)</td>
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<td>Model</td>
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<td>/lfst/IN/SP</td>
<td>/lfst/FLOOR/MAR5</td>
<td>None</td>
<td>Translational</td>
<td>False</td>
<td>Bidirectional</td>
<td>poly(-VR(/lfst/IN/SP), 0.0, 0.0, 6.5)</td>
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<tr>
<td>/hfsb/Spring</td>
<td>Single Component Force</td>
<td>Model</td>
<td>1</td>
<td>/hfsb/IN/SP</td>
<td>/hfsb/FLOOR/MAR5</td>
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<td>poly(-DM(/hfsb/IN/SP, /hfsb/FLOOR/MAR5)+4, 0.0, 1101.34, 617, 200.5, 8.1, -20.9, 14.5)</td>
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<td>Bidirectional</td>
<td>poly(-VR(/hfsb/IN/SP), 0.0, 0.0, 6.5)</td>
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**Nonlinear Low Frequency Conventional Cab**

**Nonlinear High Frequency COE Cab**
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<thead>
<tr>
<th>Parent Type</th>
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<tr>
<td>Adams ID</td>
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<td>I Marker</td>
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<tr>
<td>J Marker</td>
<td>/hfst/IN/SH</td>
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<tr>
<td>Type</td>
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<td>Function</td>
<td>poly(-VR(/hfst/FL00R/MAR5, /hfst/IN/SP), 0.0, 0.0, 14.5)</td>
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**Nonlinear High Frequency Conventional Cab**

<table>
<thead>
<tr>
<th>Object Name</th>
<th>/hfst/Spring</th>
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<tbody>
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<td>Object Type</td>
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<td>Parent Type</td>
<td>Model</td>
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</tr>
<tr>
<td>I Marker</td>
<td>/hfst/IN/SP</td>
</tr>
<tr>
<td>J Marker</td>
<td>/hfst/FL00R/MAR5</td>
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<tr>
<td>Length</td>
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<td>Actiononly</td>
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<tr>
<td>Type</td>
<td>Bidirectional</td>
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<tr>
<td>Function</td>
<td>poly(-DM(/hfst/IN/SP, /hfst/FL00R/MAR5)+4, 0.0, 1206.38, 560.9, 182.3, 7.4, -19, 13.2)</td>
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<table>
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<tr>
<th>Object Name</th>
<th>/hfst/Shock</th>
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<td>Object Type</td>
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<td>Model</td>
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<td>Type</td>
<td>Bidirectional</td>
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<tr>
<td>Function</td>
<td>poly(-VR(/hfst/FL00R/MAR5, /hfst/IN/SP), 0.0, 0.0, 14)</td>
</tr>
</tbody>
</table>
VITA

Cynthia Lynn McCoy was born in Aliquippa, Pennsylvania on December 10, 1968. She attended Ragsdale High School in Jamestown, North Carolina. She participated in many societies, including the National Honor Society. During her senior year, she was a member of the high IQ bowl team. She graduated in May, 1986. She studied one year at North Carolina State University before transferring to the University of Tennessee, Knoxville in September, 1987. She graduated with a Bachelor of Science in Engineering Science in May 1991. Starting in August 1991, she entered the graduate program at the University of Tennessee, Knoxville in the Engineering Science and Mechanics department. She concentrated her effort in the field of vibrations. In August 1994, she received her Master of Science degree in Engineering Science.