ENGINEERING DYNAMICS COMPANY

SIMON

HVE Human Vehicle Environment

Markdown Edition — revised to describe the current version of the software

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SIMON — Simulation Model Non-Linear

Engineering Dynamics Company, LLC SIMON (SIMulation mOdel Non-linear), Version 2

Updated Markdown edition, converted from the Fifth Edition printed manual (January 2006) and brought up to date with the current version of HVE, covering the SIMON physics engine, the DyMESH collision model and the HVE user-interface dialogs. Passages that differ from the printed edition because the software has changed are marked with italic

(updated: ...)

notes. Figures from the printed manual are referenced by placeholder.

How to Use This Manual

This manual is the user's manual for the SIMON simulation model. The manual describes the model features and use. Input parameters and output reports are described at a high level. This manual is intended to be used in conjunction with the HVE User's Manual. The HVE User's Manual provides in-depth coverage of the human, vehicle and environment models, as well as the HVE Event Editor and Playback Editor. For specific model definitions, the user is referred to the HVE User's Manual.

If You Have Questions

Engineering Dynamics Company, LLC provides technical assistance for supported users. The questions may relate to general usage or to specific engineering applications. When you call, you should have the following information available:

- Your User ID number
- Your System Hardware Profile report (available from EDC)
- HVE Version Information (available from the HVE Help System)
- SIMON Version Information (available from the SIMON Program Data report)

Contents

1. **Chapter 1 — SIMON Program Description** Overview; General Features and Capabilities; Model Inputs; Model Outputs; Validation; Basic Procedure
2. **Chapter 2 — SIMON Program Input** Objects Overview; Humans; Vehicles (Sprung Mass, Unsprung Mass, Exterior, Steering/Brake/Drivetrain Data); Environment; Event (Set-up, Position/Velocity, Driver Controls, Vehicle Mesh, Payload, Wheels, Accelerometers); Event Calculation Options *(incl. Hydroplaning Model, Connection Model, Solid Axle Inertia, Accident History Basis, Include Free Space)*; DyMESH Options; Simulation Controls; Executing an Event
3. **Chapter 3 — SIMON Program Output** Overview; Alpha-Numeric Reports (Accident History, Damage Data, Driver Data, Environment Data, Event Data, Human Data, Messages, Program Data, Vehicle Data); Graphic Reports; Variable Output Table; Trajectory Simulations; Damage Profiles
4. **Chapter 4 — Calculation Method** Basis for Analysis; Equations of Motion (Sprung Mass, Unsprung Mass, Wheel Spin); Steering System; Payloads; Human Occupants; Driver Controls; Wheel Torque; Wheel Position; Wheel Orientation; Tire Contact Patch Velocity; Tire-Road Contact Patch; Tire Radial Deflection and Force; Tire Model; Suspension Force; Inter-Vehicle Connection Model; Dollys; Aerodynamics Model; Sprung Mass Impact Model (DyMESH); Software Implementation
5. **Chapter 5 — SIMON Tutorial** ISO 3888 lane-change maneuver with and without ABS: creating the vehicle and environment, setting up and executing the events, and viewing/printing all output reports
6. **Chapter 6 — Messages** Complete catalog of SIMON run-time messages (fatal, diagnostic and informative)
7. **Chapter 7 — Technical References** Published references for the SIMON, DyMESH, tire, brake and driver models
8. **Chapter 8 — Index** Topic index adapted for this Markdown edition
9. **Electronic Stability Systems (TCS and YSC)** Supplementary reference for the Traction Control and Yaw Stability Control models: sensor models, control laws, user parameters, outputs, and limitations

Notable Differences from the Fifth Edition (2006)

- **Calculation Options** have grown from three options (Tire Model Method, Steer DOF, Collision Acceleration Threshold) to include: three semi-empirical tire model versions, an AutoStart Steer DOF mode, a Hydroplaning Model (NASA / NASA-TTI / Gallaway), a Connection Model selector, a Connection Failure Start Time, a Solid Axle Inertia selector, an Accident History Basis (Impact Force / Acceleration) selector, and an Include Free Space damage-report option (see Chapter 2).
- **DyMESH Options** are now configured in a dedicated dialog: Use DyMESH, Include Environment, Force to x-y Plane, Tow Vehicle / Trailer Contact, DyMESH Version 3/4, vehicle and environment start times, contact search (Automatic / Set Box Size) and mesh smoothing model (see Chapter 2).
- **DyMESH wheel contact and damage**: wheels may carry collision meshes; collision forces beyond user-set thresholds displace and reorient wheels, and the displacement is now capped by the actual wheel-mesh contact motion each timestep (see Chapter 4, Sprung Mass Impact Model).
- **Message catalog** extended with collision/damage-related messages and a roll/pitch limit message (see Chapter 6).

Chapter 1 — SIMON Program Description

Overview

SIMON (**SI**mulation **MO**del **N**on-linear) is a dynamic simulation of the response of one or more vehicles to driver inputs, inter-vehicle collision(s) and factors related to the environment (e.g., terrain, atmosphere). SIMON is a simulation model built on a general-purpose 3-D vehicle dynamics engine developed by Engineering Dynamics Company, LLC. The dynamics engine allows a sprung mass with six degrees of freedom and multiple axles with up to five degrees of freedom per axle. SIMON was specifically designed to take advantage of the rich feature set available in the HVE simulation environment, including the HVE Brake Designer, Driver Model and Tire Blow-out Model.

General Features and Capabilities

Each SIMON vehicle may have up to three axles (six axles for full trailers) and single or dual tires. Independent and solid axle suspensions are allowed. SIMON may be used for numerous types of simulation studies. Applications for SIMON include:

- **Single Vehicle Simulation** — SIMON's 3-D simulation model incorporates robust tire and suspension models required for vehicle handling simulation studies, including rollover.
- **Articulated Vehicle Simulation** — SIMON also simulates the dynamics of articulated multi-vehicle trains. The model is general and allows the user to model virtually any vehicle/trailer configuration.
- **Collision Simulation** — SIMON incorporates DyMESH, a general-purpose, 3-D collision model for simulating vehicle-to-vehicle and vehicle-to-barrier collisions.
- **Vehicle Design** — SIMON may be used for vehicle design projects involving suspension, brake (including ABS), tire and steering systems. Using SIMON, these systems may be optimized on the computer before expensive prototyping and testing begins. Parametric studies involving in-use factors, such as weight distribution changes due to occupant loading and payload location, may also be performed.
- **Safety Research** — Safety researchers may use SIMON for reconstructing most crashes involving single or multiple vehicles. Human, vehicle and environment factors may be simulated and evaluated to assist in performing detailed crash studies.

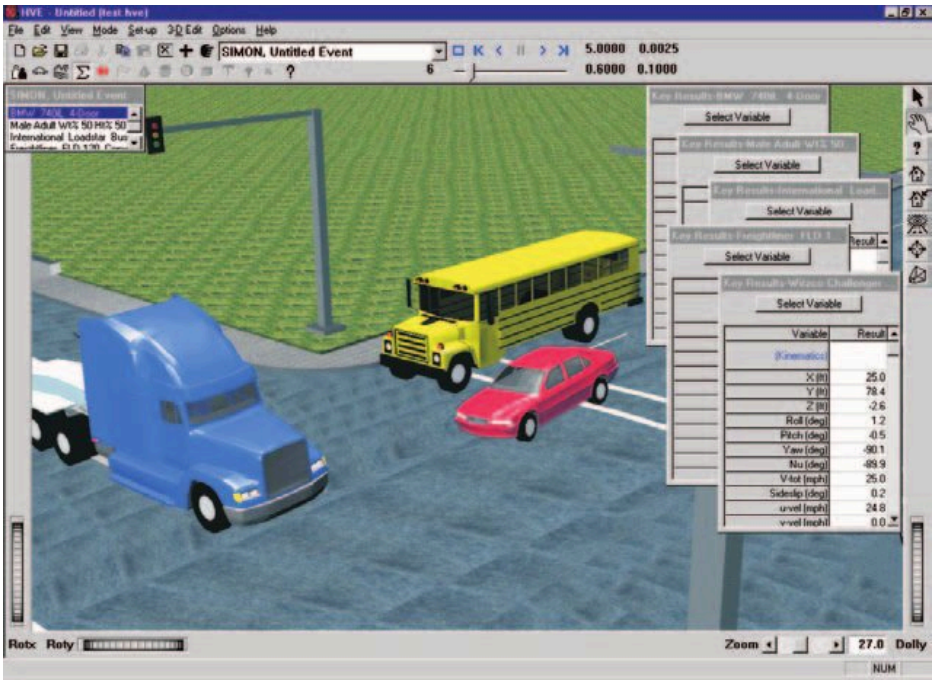


Figure 1-1: SIMON Event.

Model Inputs

SIMON inputs include one or more vehicles. Articulated vehicles are created from individual unit vehicles using compatible front and rear connections. Full trailers are created using a semi-trailer and a front dolly.

Human occupants may also be included in SIMON. Placing a human in a vehicle affects the vehicle's inertial properties and weight distribution; however, human motion relative to the vehicle is not modeled.

An environment is optional. If supplied, the environment may be a simple flat surface or complex 3-D digital terrain map (DTM) from a total station survey. An unlimited number of friction zones may be supplied.

Model Outputs

SIMON output reports include the following HVE Output Reports:

- Accident History
- Damage Data
- Damage Profiles
- Driver Controls
- Environment Data
- Event Data
- Human Data
- Messages
- Program Data
- Trajectory Simulation
- Variable Output
- Vehicle Data

Validation

The validation of SIMON for unit vehicle (UV), articulated vehicle (AV) and vehicle collision (VC) simulation has been completed and is available in Reference 16. A summary of the validation test matrix is shown in Table 1-1.

This test matrix represents a cross-section of the types of events for which SIMON was intended. It is anticipated that additional validations will be published as they are performed.

Table 1-1 SIMON Validation Study. Matrix of test configurations.

Test	Description	Validation By
UV-1	Passenger Car Combined Steering and Braking [11]	Experiment, EDVSM
UV-2	Passenger Car Alternate Ramp Traversal [11]	Experiment, EDVSM
UV-3	Light Truck Rollover [12]	Experiment, EDVSM
UV-4	Straight 3-Axle Truck (30 mph), Combined Braking and Steering [13]	Experiment, EDVDS
AV-1	Chevrolet Pickup (55 mph) towing Open-wheel Utility Trailer, ISO Lane-change with braking	EDSMAC4
AV-2	Tractor-trailer (27 mph), Combined Braking and Steering (U of M) [13]	Experiment, EDVDS
VC-1	1992 Ford Explorer (45.9 mph) vs. 1984 Ford F-150 Pickup (46.1 mph), 55 Degree Angled Impact [14]	Experiment, EDSMAC4
VC-3	1974 Ford Torino (21.23 mph) vs. 1974 Ford Pinto (0 mph), 10 Degree Offset Rear-end Impact (Calspan, RICSAC 3) [15]	Experiment, EDSMAC4
VC-7	1974 Chev Chevelle Malibu (31.53 mph) vs. 1974 Ford Pinto (31.53 mph), 60 degree Angled Impact (Calspan, RICSAC 2) [15]	Experiment, EDSMAC4
VC-8	1974 Chev Chevelle Malibu (21.47 mph) vs. 1975 Volkswagen Rabbit (21.47 mph), 60 degree Angled Impact (Calspan, RICSAC 6) [15]	Experiment, EDSMAC4
VC-11	1974 Honda Civic CVCC (31.35 mph) vs. 1974 Ford Torino (31.35 mph), 90 Degree Angled Impact (Calspan, RICSAC 10) [15]	Experiment, EDSMAC4

Basic Procedure

The procedure for using SIMON is essentially the same as using any simulation model in the HVE environment:

- Use the HVE Human Editor to add one or more human occupants or pedestrians to the case (this step is optional).
- Use the HVE Vehicle Editor to add one or more vehicles to the case. Optionally, edit any of the default vehicle parameters.
- Optionally, use the HVE Environment Editor to create an environment. A 2- or 3-dimensional terrain may also be added.
- Use the HVE Event Editor to set up and execute the SIMON simulation model by performing the following steps:
- Choose the humans and vehicle(s) in the desired order.

> **NOTE:** The order determines how articulated vehicles are connected: a vehicle without a front connection is assumed to be a unit vehicle or the first vehicle in a vehicle train.
>
> **NOTE:** The order in which humans are selected determines the vehicles in which the human(s) are riding: a human is riding in the first vehicle preceding it in the selection list.

- Choose the SIMON simulation model.
- The next step is to set up the event. For each human and vehicle in the Event Humans and Vehicles list:

- Position the object in the environment (or vehicle, in the case of human occupants) and supply a velocity.
- Optionally, assign Driver Controls (Steering, Braking, Throttle, Gear Selection).
- Optionally, assign a collision pulse.
- Optionally, assign a payload.
- Optionally, assign tire blow-out, wheel displacement and/or brake temperature and adjustment data.
- Optionally, assign locations for one or more accelerometers.
- Execute the simulation event.
- Modify the initial conditions, driver controls and other parameters as required to achieve the desired match between simulation and test conditions.
- Use the HVE Playback Editor to view the various reports.
- If desired, produce a simulation movie of the simulation.

Chapter 2 — SIMON Program Input

This chapter defines the objects (humans, vehicles and environment) and the event set-up parameters (positions, velocities, driver controls, and so forth) used by the SIMON analysis. In general, the chapter is divided into the following sections:

- **Objects** — The number of humans and vehicles, and the specific human, vehicle, and environment parameters actually used by SIMON.
- **Events** — The various HVE options available for setting up and executing a SIMON event.

Objects Overview

The objects used by the SIMON model are:

- **Humans** — Human occupants may be placed in vehicles in SIMON, and human pedestrians may be placed in environments in SIMON.
- **Vehicles** — An infinite number of vehicles may be studied by SIMON.
- **Environment** — Like the *real* world, SIMON has exactly one environment.

> **NOTE:** The environment is used in any HVE-compatible reconstruction or simulation model.

The following sections describe how the humans, vehicles and environment provide the required inputs to the SIMON calculation model.

Humans

SIMON uses humans created using the HVE Human Editor (Figure 2-1). Humans are selected from the Human Database by choosing the following attributes:

- **Location** — SIMON supports all occupant locations (pedestrians will appear in the event but are not used in SIMON calculations)
- **Sex** — SIMON supports both male and female
- **Age** — Adult or child (specified by years of age)
- **Weight Percentile** — SIMON supports all available weight percentiles
- **Height Percentile** — SIMON supports all available height percentiles

To add a human to the current HVE case, perform the following steps:

1. Choose Human mode. The Human Editor is displayed.
2. Choose *Add New Object*. The Human Information dialog is displayed.
3. Click on the *Location, Sex, Age, Weight Percentile,* and *Height Percentile* option buttons to select a human from the database.
4. Enter a name for the current human. A default name is supplied for each selected human. The name is user-editable, and does not affect calculations.

> **NOTE:** Duplicate human names are not allowed in the same case.

1. Click *OK* to add the human to the current case.

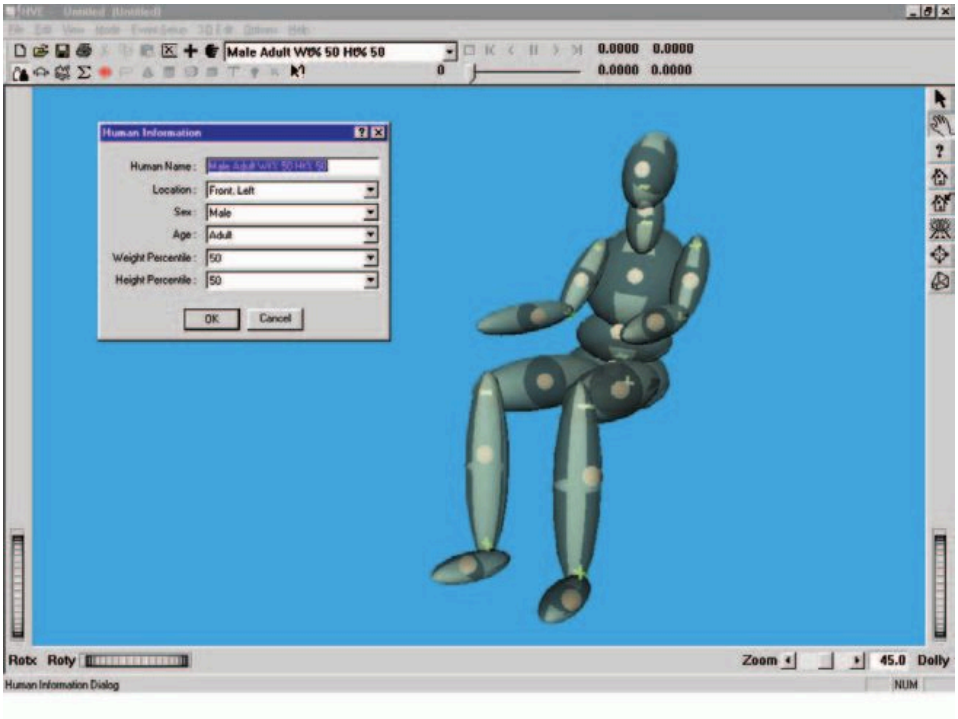


Figure 2-1: HVE Human Editor.

> **NOTE:** SIMON is not a human simulator. During simulation, the human occupants do not move relative to the vehicle.

The following human parameters are used by SIMON:

- Inertia
- Weight

Inertia

Weight

SIMON uses the total weight of the human in order to adjust the inertial properties of the vehicle sprung mass. The weights of all ellipsoids are added together to compute the total human weight. The total weight is applied to the vehicle at the vehicle-fixed coordinates of the pelvis CG. Neither the rotational segment inertias nor the body configuration is used by SIMON.

Vehicles

SIMON uses vehicles created using the HVE Vehicle Editor (Figure 2-2). Vehicles are selected from the Vehicle Database by choosing the following attributes:

- **Type** — SIMON supports all vehicle types including: *Passenger Car, Pickup, Van, Sport-Utility, Truck, Trailer, Dolly, Fixed Barrier* and *Movable Barrier*.
- **Make** — SIMON supports all available vehicle makes.
- **Model** — SIMON supports all available vehicle models, within the limits defined by number of axles; see below.
- **Year** — SIMON supports all available vehicle years.
- **Body Style** — SIMON supports all available vehicle body styles.

Each vehicle also has the following additional user-editable parameters:

- **Number of Axles** — SIMON supports 0-, 1-, 2- and 3-axled vehicles.
- **Driver Location** — The *Driver Location* is not used by SIMON. However, if *Driver Location* is *None*, the Driver Controls (Steering, Brakes, Throttle, Gears) will not be available during Event mode.
- **Engine Location** — The Engine Location is not used by SIMON. However, if *Engine Location is None*, the Throttle and Gear Tables will not be available during Event mode.
- **Drive Axle(s)** — SIMON supports all drive axles.

To add a vehicle to the current HVE case, perform the following steps:

1. Choose Vehicle mode. The Vehicle Editor is displayed.
2. Choose *Add New Object*. The Vehicle Information dialog is displayed.

3. Click on the *Type, Make, Model, Year* and *Body Style* option buttons to select a vehicle from the database.
4. If desired, modify the *Number of Axles, Driver Location, Engine Location* and *Drive Axle(s)* for the current vehicle.
5. Enter a name for the current vehicle. A default name is supplied for each selected vehicle. Its name is user-editable, and does not affect calculations.

> **NOTE:** Duplicate vehicle names are not allowed in the same case.

1. Click *OK* to add the vehicle to the current case.

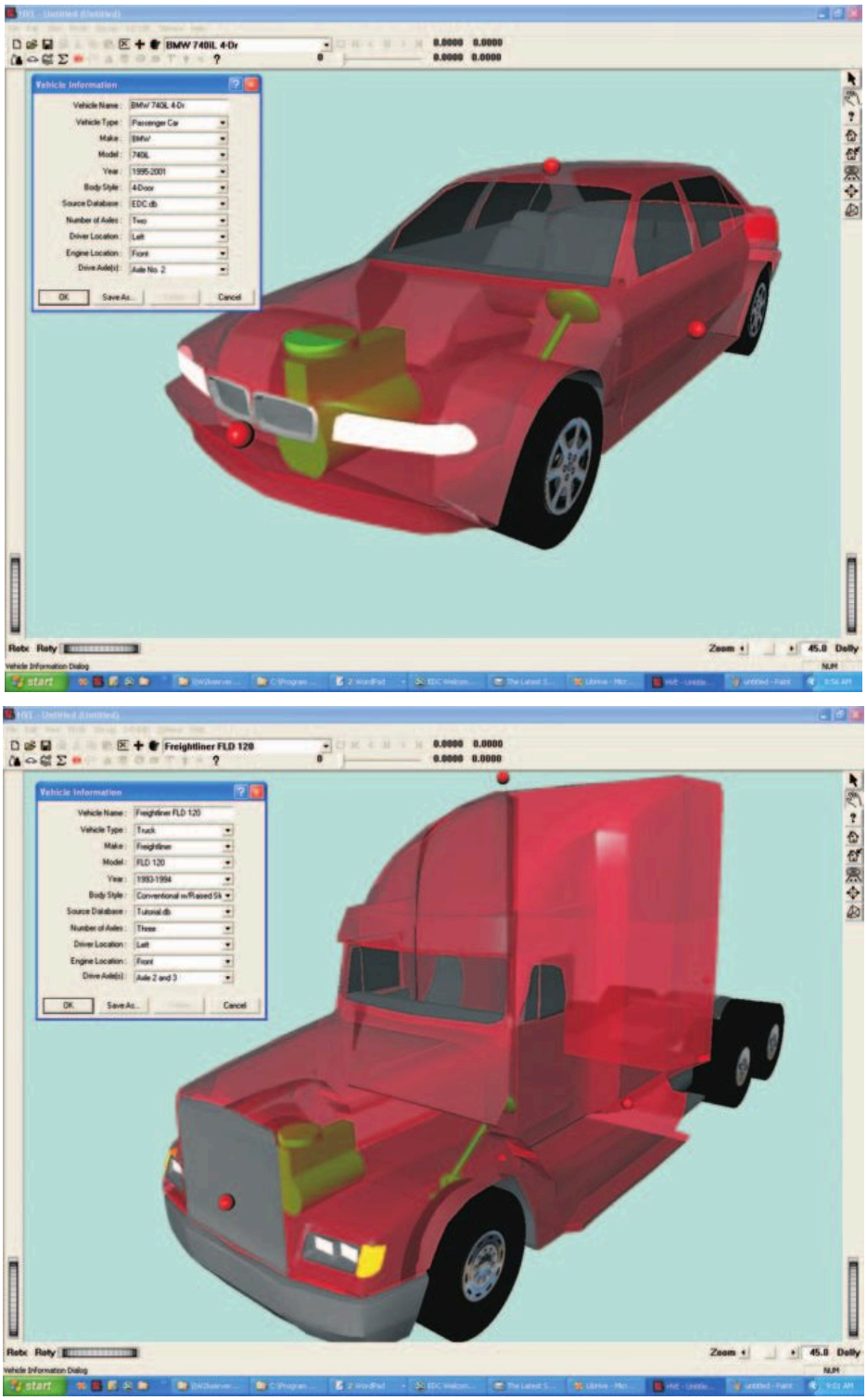


Figure 2-2: HVE Vehicle Editor showing a passenger car and a truck.

The following Vehicle Parameter groups are used by SIMON:

- Sprung Mass
- Inertias
- Move CG
- Connections
- Aerodynamic Drag
- Body Torsion
- Geometry File
- Unsprung Mass
- Suspension Data
- Brake Data
- Tire Data
- Wheel Location
- Steering System
- Braking System
- Drivetrain
- Engine
- Transmission
- Differential
- Inertia
- Exterior Stiffness

The specific data used in each of the above parameter groups are defined in Tables 2-1 through 2-5.

Sprung Mass

The Sprung Mass parameters are shown in Table 2-1. Information on each group is provided below.

Table 2-1 Vehicle Sprung Mass Parameters Used By SIMON.

Parameter	Description
Sprung Weight	Weight of vehicle sprung mass
Sprung Mass Roll, Pitch and Yaw Inertia, and x-z Product of Inertia	Rotational inertia of sprung mass about the vehicle-fixed x, y and z axes, respectively, and x-z product of inertia
Move CG x, y, z	Relocates the CG in the vehicle-fixed x, y and z directions. This entry causes an automatic adjustment of all vehicle coordinate-related parameters (e.g., contact surface, belt anchor points).
Geometry File	Vehicle exterior 3-D mesh and dimensions used by DyMESH
Inter-vehicle Connections	Type, location, and properties of front and rear connections
Aerodynamic Drag	Drag coefficient, projected surface area, and center of pressure x, y, z coordinates for each surface or device that is defined as installed

Inertias

SIMON uses the sprung vehicle weight (converted to mass according to the current gravitational constant; see Environment), as well as the roll, pitch, and yaw rotational inertia and x-z product of inertia of the sprung mass.

The total weight is often entered because it is the value traditionally obtained from data sources. However, the equations of motion require that the sprung mass be known. To obtain this value, SIMON simply deducts the weight of the suspensions and wheels from the total weight.

Move CG

Move CG is not directly used by SIMON, i.e., the current values do not show up in the SIMON results. However, the Move CG fields may be used in the Vehicle Editor to quickly move the vehicle's center of gravity; the x, y, z coordinates for the wheels and other vehicle-fixed components are updated to reflect the new CG location.

Geometry File

The Geometry File supplies the vehicle exterior vertex data used by DyMESH.

Contact Surfaces

Contact Surface parameters are not used by SIMON.

Belt Restraints

Belt Restraint parameters are not used by SIMON.

Airbag Restraints

Airbag Restraint parameters are not used by SIMON.

Inter-vehicle Connections

Inter-vehicle Connection parameters define the type, location and mechanical properties of the front and rear vehicle connections. The x, y, and z location is specified relative to the vehicle CG. The z coordinates for each vehicle determine the elevation at the connection point, and thus establish the pitch angle of the trailer.

Dollys may be either *fixed* or *converter* dollys. A fixed dolly is permanently attached to the front of the trailer and normally has a hinged drawbar. A converter dolly is removable. It normally has a hinged fifth wheel and a fixed drawbar. See Chapter 4 for more information about drawbar types and their characteristics.

The *Connection Friction, Radius* and *Maximum Articulation Angle* are only supplied for the rear connections.

> **NOTE:** Every trailer will have a pre-defined front connection. However, not every tow vehicle has a pre-defined rear connection. Use the Vehicle Editor to confirm the vehicle has a connection.

Drag Forces

Aerodynamic drag forces are allowed to act on up to six surfaces (front, rear, right, left, top, bottom) and two user-defined devices (e.g. front or rear wing). These forces are functions of projected area,

Aerodynamic Drag Coefficient

, wind speed and air density. By default, the front surface is assigned aerodynamic properties. Assignment of aerodynamic properties for other surfaces is left up to the user. See Chapter 4 for more information about aerodynamic drag forces.

Unsprung Mass

The Unsprung Mass parameters used by SIMON are shown in Tables 2-2 and 2-3. Information on each group is provided below.

Wheel Location

SIMON allows for the definition of x, y, z coordinates for each wheel center without any assumption of vehicle symmetry. The specific parameters used by SIMON are shown in Table 2-2.

> **NOTE:** SIMON uses z_{wheel} and the tire radius (see Tire Parameters) to calculate static CG elevation above ground.
>
> **NOTE:** It is possible, therefore, that one tire, on a four wheeled vehicle, could be out of contact with the ground in both static and dynamic conditions.

Brake

SIMON uses Time Lag and Time Rise in controlling brake torque application. Each wheel's brake assembly optionally can be described using the HVE Brake Designer (see HVE User's Manual, Section Eleven, Chapters 29 and 30, and reference 1). SIMON does not use the wheel brake system Antilock Effectiveness. Instead, SIMON uses the HVE ABS Model to simulate ABS. This is a robust, algorithm-driven ABS simulation model. See also Brake System (later in this chapter), HVE User's Manual, Chapter 31,

Anti-lock Braking Systems

, and reference 10.

Table 2-2 Vehicle Unsprung Mass, Wheel and Brake Parameters Used By SIMON.

Parameter	Description
Wheel Location	Vehicle-fixed x, y, z coordinates for each wheel
Brake Time Lag, Brake Time Rise	Time delay from initiation of pedal force to initiation of brake pressure at the wheel, and amount of time necessary to build up brake torque after pressure reaches the wheel
Brake Assembly Type, Brake Push-out Pressure, Brake Torque Ratio	Type of generic, disc, or drum brake (Brake Designer), and physical parameters of the brake system
Brake Proportioning Ratio, Brake Pstart	Ratio for splitting brake pressure between front and rear brakes, and pressure at which proportioning begins
ABS Wheel Data Parameters	Algorithm-dependent parameters used by the selected ABS model (see Brake System)

Tire

SIMON uses a modified version of the EDC Semi-empirical Tire Model developed for EDVDS [2, 3]. The HVE tire parameters provide the following data groups:

- Physical Data
- Friction Table
- Cornering Stiffness Table
- Camber Stiffness Table
- Slip vs. Rolloff Table

SIMON's use of these parameters is described below and in Table 2-3.

Physical Data

The Tire Physical Data used by SIMON are shown in Table 2-3. The tire's unloaded radius is used to establish the vehicle's static CG elevation. Aligning torque is not used.

Friction Table

The Friction Data used by SIMON are shown in Table 2-3. At each time-step SIMON interpolates between the data in the Friction vs. Longitudinal Slip table to obtain the peak friction, slide friction, slip at peak friction and *Longitudinal Stiffness* for the tire's current load and speed. The interpolated peak and slide friction values are then multiplied by the friction multiplier of the terrain beneath the tire; the longitudinal stiffness is not. *(updated: earlier editions stated that the Longitudinal Stiffness value was used only when there was no slippage in the contact patch. It is interpolated and used at every time-step — see Chapter 4, Tire Model.)*

The

In-use Factor

is a convenient way to reduce or increase the dependent friction values (peak and slide friction) for all loads and speeds by making just one adjustment.

Cornering Stiffness Table

The Cornering Stiffness data used by SIMON are shown in Table 2-3. SIMON interpolates between the data in the F_y vs. Slip Angle table to calculate cornering stiffness for the correct load and speed at each time-step.

The

In-use Factor

is a convenient way to reduce or increase the dependent cornering stiffness values for all loads and speeds by making just one adjustment.

> **NOTE:** If you are simulating a vehicle with a flat tire, you'll probably want to use the HVE Tire Blow-out Model (see Event).

Table 2-3 Vehicle Unsprung Mass, Tire Parameters Used By SIMON.

Parameter	Description
Tire Unloaded Radius	Tire radius before any load is applied
Tire Initial and Secondary Deflection Rates, Deflection at Secondary Rate and Maximum Deflection	The initial spring rate of the tire, and the spring rate after a specified amount of tire deflection. The maximum deflection is used to determine when tire deflection exceeds an allowable value.
Pneumatic Trail	In the tire-fixed coordinate system, the x' coordinate of the center of pressure
Weight, Spin Inertia, Steer Inertia	The weight and rotational inertia (about the spin and steer axes) of the tire/wheel/brake assembly
Linear and Velocity Dependent Rolling Resistance Constant	Constants accounting for rolling resistance due to tire and terrain properties
Tire Friction Table, Test Load/Speed	The load and speed for a given set of friction results
Peak Longitudinal Friction, Peak Lateral Friction, Slide Friction, Slip at Peak Friction and Longitudinal Stiffness	Tire frictional and longitudinal stiffness properties at the specified load and speed
Friction In-use Factor	Multiplier for Longitudinal and Slide Friction
Tire Cornering Stiffness Table, Test Load and Speed	The load and speed for a given cornering stiffness value
Cornering Stiffness	Tire lateral force per unit of tire lateral slip for small amounts of lateral slip
Cornering Stiffness In-use Factor	Multiplier for Cornering Stiffness values
Tire Camber Stiffness Table, Test Load and Speed	The load and speed for a given camber stiffness value
Camber Stiffness	Tire lateral force per unit of inclination angle for small amounts of inclination angle
Camber Stiffness In-use Factor	Multiplier for Camber Stiffness values

Camber Stiffness Table

The Camber Stiffness data used by SIMON are shown in Table 2-3. SIMON interpolates between the data in the F_y vs. Inclination Angle table to calculate camber stiffness for the correct load and speed at each time-step.

The

In-use Factor

is a convenient way to reduce or increase the dependent camber stiffness values for all loads and speeds by making just one adjustment.

Slip vs. Rolloff Table

SIMON does not use either the longitudinal or lateral Slip vs. Rolloff Tables.

Suspension

The HVE Suspension Model provides the following data groups:

- Springs and Shocks

- Inertia
- Jounce/Rebound Stops
- Spindle Axis
- Camber Table
- Anti-pitch Table
- Roll Steer Table

SIMON supports all suspension types, including those with tandem axles. Independent suspension types do not use inertial properties (the weight of the suspension arms is ignored; the weight of the brake and wheel are included with the tire). Independent and solid axle suspension types use different camber and roll-steer parameters. Spindle axis is only available for steerable wheels (see Steering System). SIMON's use of these parameters is described below.

Inter-tandem Axle Load Transfer

Inter-tandem Axle Load Transfer is not used by SIMON.

Springs and Shocks

SIMON uses the individual spring rates for the right and left springs on an axle. They need not be equal.

Roll Center Height and Lateral Spring Spacing apply for each axle.

> **NOTE:** Remember that roll center height is measured from the sprung mass CG, and is positive downwards.

Damping properties, like spring rates, are applied to the right and left sides of each axle. They need not be equal.

The Spring and Shock data used by SIMON are shown in Table 2-4.

Table 2-4 Suspension Mechanical and Geometry Parameters Used By SIMON.

Parameter	Description
Wheel Ride Rate	Linear force vs. Deflection characteristic measured at the wheel
Auxiliary Roll Stiffness	Moment per unit of body roll produced by the addition of an anti-sway bar
Roll Center Height	Vertical distance from the vehicle CG to the roll axis measured at the suspension
Lateral Spring Spacing	Lateral distance between right and left side suspensions
Shock Damping	Damping at wheel
Coulomb Friction and Friction Null Band	Coulomb friction and friction null band at wheel
Suspension Inertias	Weight and roll/yaw rotational inertia of the suspension, excluding wheels and brakes
Maximum Deflection at Jounce/Rebound Stop	Maximum suspension travel in jounce/rebound
Linear and Cubic Deflection Rates at Jounce/Rebound Stop	Linear and cubic stiffness rates of jounce/rebound stop
Energy Ratio of Jounce/Rebound Stop	Ratio of stop relaxation force to stop compression force
Caster	Rearward tilt of steering axis
Kingpin Inclination Angle	Inward tilt of steering axis
Stub Axle Length	Distance from steering axis to wheel center
Steering Offset (Scrub Radius)	Distance from wheel plane to steering axis at ground plane
Steering Stop Angles	Steer angle for right and left steering stops
Steering Stop Stiffness	Mechanical stiffness of steering stops
Camber	Camber and Half-track change as a function of jounce/rebound (constant camber value used for solid axle suspension)
Roll Steer	Constant, linear, quadratic, and cubic coefficient of roll steer (constant roll steer value used for solid axle suspension)

Inertias

Inertial mass parameters do not include the weight of the brake assemblies or wheels (brake and wheel inertias are included with the tire). The Suspension Inertia data used by SIMON are shown in Table 2-4.

Jounce and Rebound Stops

The linear and cubic deflection rate of the suspension stops is the effective value at the wheel. Left and right side suspension stops are not assumed to be the same. The Jounce and Rebound Stop data used by SIMON are shown in Table 2-4.

Spindle Axis

Caster, kingpin inclination angle, stub axle length and steering offset are used to orient the wheel (wheel camber and inclination angles change as the wheel jounces and steers).

> **NOTE:** Wheel Camber (below) also affects wheel orientation.

Steer stop data (stop angles and stop stiffness) are used when the Steer Degree of Freedom option is selected.

Camber and Half-track Change Tables

A constant Camber value may be assigned at each wheel for a solid axle suspension.

Camber and half-track tables apply only to independent suspension types. The default properties for the right side and left side tires are the same, but they may be edited individually.

Anti-pitch

Anti-pitch force vs. Suspension jounce and rebound are not used by SIMON.

Roll Steer

A Roll Steer vs. Jounce/Rebound Table is used for independent suspension types; a single value is used for solid axles. The roll steer properties for the right and left side tires are not assumed to be the same.

Exterior

The Vehicle Exterior Stiffness Data are used by the DyMESH collision routine within the SIMON physics model; see DyMESH manual.

Steering System Data

SIMON uses the steering gear ratio if the

At Driver

Steering Table option is used during Event set-up (see Table 2-5). SIMON also uses steering column friction and friction null band, and steering column inertia if the Steer Degree of Freedom option is selected.

If the Steer Degree of Freedom Normal option is selected, SIMON ignores user-entered steering system data and computes steer angles based on driving surface inputs. If the Append option is selected, SIMON uses the user-entered steering system data until the last timestep for which data are entered; after that, the Steer Degree of Freedom model is used.

The steering system data used by SIMON are shown in Table 2-5.

Table 2-5 Steering System, Brake System, and Drivetrain Parameters Used By SIMON.

Parameter	Description
Steering Gear Ratio	Ratio of steering wheel angle to wheel angle
Steering Column Friction and Friction Null Band	Friction and friction null band of steering column
Steering Column Inertia	Rotational inertia of the steering column
Brake Pedal Ratio	Ratio of brake system pressure to pedal force
Engine Power vs. Speed Table	Tables of engine power vs. engine speed for open and closed throttle positions
Transmission Ratio	Ratio of engine to drive shaft rotation
Differential Ratio	Ratio of drive shaft to axle rotation
Drivetrain Inertia	Rotational inertia of the drivetrain

Brake System Data

The Brake System pedal ratio (system pressure per unit of pedal force) is used if the

Pedal Force

Brake Table option is used during Event set-up (see Table 2-5).

The Brake System Data also include the HVE ABS System Model option. If this model is invoked, SIMON uses the ABS System Data parameters and ABS Wheel Data parameters. See also HVE User's Manual, Chapter 31,

Anti-lock Braking Systems

, and reference 10.

> **NOTE:** See also Wheel Brake Data (Unsprung Mass Parameters) for related information.

Drivetrain Data

Engine power as a function of engine speed is defined for both wide-open throttle and closed throttle positions. The Drivetrain data (engine power vs. speed, transmission ratio, differential ratio, drivetrain inertia) are used if the

Percent Wide-open Throttle

option is used during Event set-up (see Table 2-7). Rotational inertia includes the engine, transmission, driveline, and differential.

The drivetrain data used by SIMON are shown in Table 2-5.

The user-entered Power vs. Engine Speed Table defines the valid range of engine speeds for a given event. If the maximum engine speed in the table is exceeded, the simulation terminates with an error message (see Messages).

> **NOTE:** The typical solution to this problem is to shift sooner.

The transmission may have up to 12 forward ratios, plus reverse and neutral. The differential may have up to three ratios.

The transmission may be either manual or automatic. A manual transmission takes its gear from the driver's Gear table. An automatic transmission selects its own gear from the current engine speed and throttle position, using the vehicle's shift data — a minimum and maximum engine speed for shifting, and the upshift and downshift throttle positions at each. See Chapter 4,

Drive Torque — Gear selection

.

(updated: automatic transmission support was not described in the Fifth Edition manual.)

> **NOTE:** The *Use Clutch/Torque Converter* option is part of the event's Driver Controls rather than the vehicle's Drivetrain data. When it is selected, the engine is allowed to turn faster than the road speed implies while the transmission is in reverse or first gear. See Chapter 4, *Drive Torque — Clutch*.

Environment

SIMON uses the environment created by the HVE Environment Editor (Figure 2-3). The environment is created by defining the following groups of attributes:

- Visual Data
- Physical Data

Creating an Environment

To add an environment to the current HVE case, perform the following steps:

1. Choose Environment mode. The Environment Editor is displayed.
2. Choose *Add New Object*. The Environment Information dialog is displayed.
3. Click on the *Location* combo box to select the desired city, state and country, and associated *Latitude*, *Longitude* and *GMT*.
4. Enter the *Time* and *Date* for the event.
5. Enter the *Angle of the X axis*, *Wind Speed* and *Direction*, *Barometric Pressure* and *Temperature* for the event.
6. Enter the *Gravity Constant* for the event.
7. Enter an environment name. A default name is supplied for the current environment. The name is user-editable, and does not affect calculations.
8. Click *OK* to add the environment to the current case.

The Visual and Physical attribute groups are defined below.

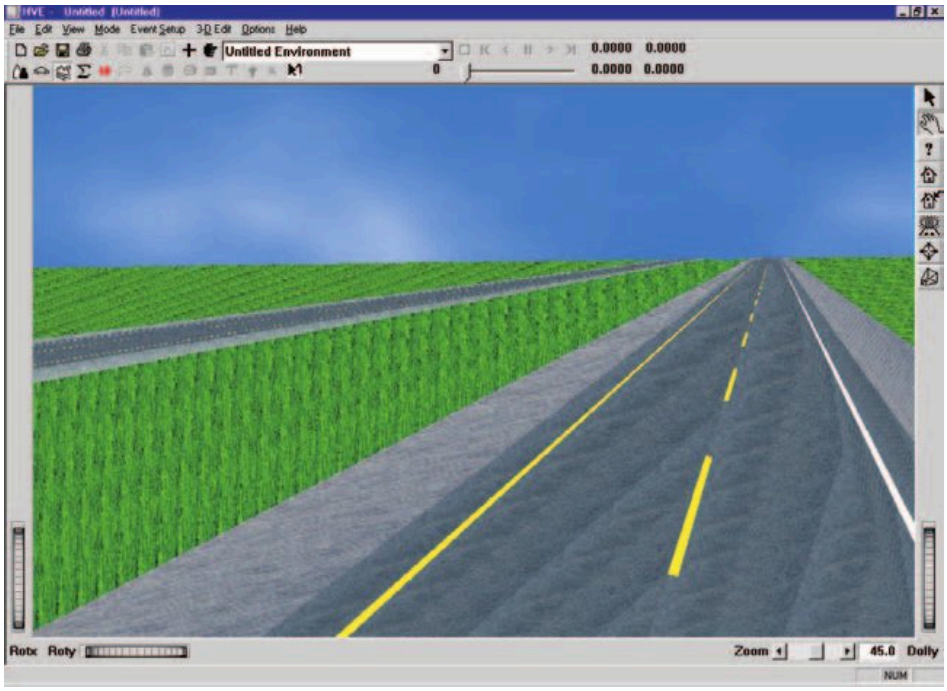


Figure 2-3: HVE Environment Editor.

Visual Data

The following visual parameters may be edited:

- **Environment Location** — A database containing the name (City/State/Country), Latitude and Longitude and GMT for the selected location.
- **Time and Date** — The local standard time and date for the event.

The visual data are not used by the event; they are provided for studies related to visibility at the time of an event (e.g., avoidability of an accident).

> **NOTE:** The visual data (Location, Time, Date and Angle of earth-fixed X axis) affect the lighting of the event! Depending on your view (Camera Position) the scene may be shaded and difficult to see. If the time is after sundown, the view will be dark.

Physical Data

The Physical Data groups are:

- Angle of X Axis
- Wind Speed and Direction
- Atmospheric Temperature and Pressure
- Gravity Constant
- 3-D Surface Geometry

The specific physical environment data used by SIMON are described below and in Table 2-6.

Table 2-6 Environment Model Parameters Used By SIMON.

Parameter	Description
Angle of X Axis	The angle between the earth-fixed X axis and true North
Wind Speed and Direction	Local wind speed and direction relative to earth-fixed X axis
Atmospheric Temperature and Pressure	Local atmospheric temperature and pressure
Gravitational Constant	Local gravitational constant
3-D Surface Geometry (Elevation, Friction Factor)	The polygon database used to create the 3-D environment describing the physical characteristics of the driving surface

Angle Of X Axis

The angle of the X axis is used to position the earth-fixed coordinate system on the surface of the earth.

> **NOTE:** The angle is specified relative to true North. If you are using a compass to determine direction at the scene of an accident, you should provide a correction factor before entering this angle.
>
> **NOTE:** The angle of the X axis affects how you visualize a SIMON event because it affects the location of the sun.

Wind Speed and Direction

SIMON uses Wind Speed and Direction together with vehicle velocity and exterior to calculate aerodynamic forces acting on the vehicle (see Chapter 4, Calculation Method).

> **NOTE:** Enter the wind direction relative to the earth-fixed coordinate system.

Atmospheric Temperature and Pressures

Atmospheric temperature and pressure are used to calculate the density of the air. The density, in turn, is used in the aerodynamic calculations.

Gravity Constant

The gravity constant converts mass to force. An object's mass and rotational inertias are properties that are the same throughout the universe; however, the weight of an object is dependent on the local gravitational constant.

3-D Surface Geometry

The 3-D Surface Geometry is used by the tire model in SIMON to calculate the elevation, slope and friction multiplier for the current X, Y position of the tire.

Rapid changes in surface elevation, such as curbs, during one integration timestep may cause severe tire force causing the run to terminate. If this occurs, you should try reducing the integration timestep or smoothing, to the degree possible, the surface.

Event

SIMON uses the HVE Event Editor (Figure 2-4) to create, set up and execute an event. Each of these topics is described below.

Creating An Event

A SIMON event is created using the Event Information dialog.

To create a SIMON event:

1. Choose Event mode. The Event Editor is displayed.
2. Choose *Add New Object*. The Event Information dialog is displayed.
3. Optional: Select humans from the Active Humans list.
4. Select vehicles from the Active Vehicles list.

> **NOTE:** The order in which humans and vehicles are selected is important in SIMON, as in all HVE-compatible physics models. Humans are assigned to be "riding" in the vehicle that is chosen directly before them. For combination vehicles, trailers and dollys are also assigned to be attached to the vehicle chosen immediately before them.

1. Select the calculation model, *SIMON*, from the Calculation Model options list.
2. Enter an event name. A default name is supplied for the selected event. The name is user-editable, and does not affect calculations.

> **NOTE:** Duplicate event names are not allowed in the same case.

1. Click *OK* to create the SIMON event.

> **NOTE:** If you choose a vehicle that is not compatible with SIMON, a message will be displayed describing the problem. You will not be allowed to proceed until SIMON-compatible objects are selected.

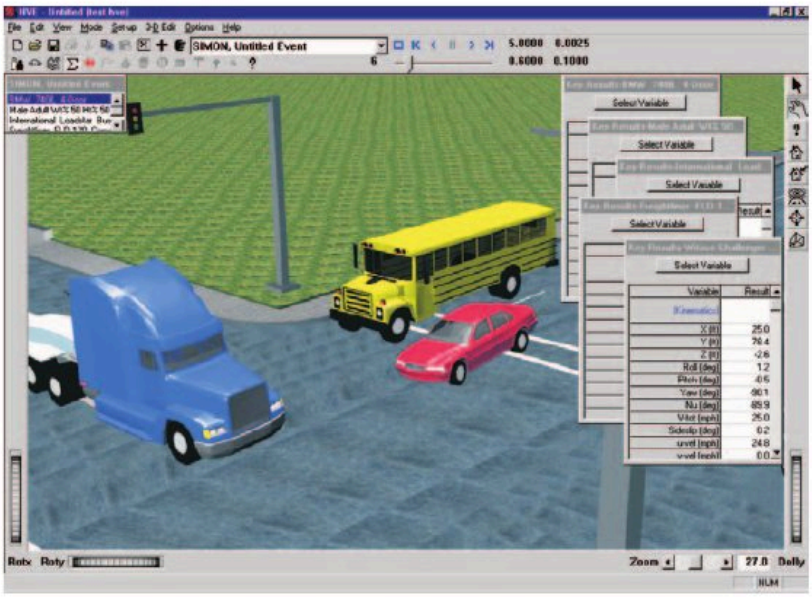


Figure 2-4 HVE Event Editor setting up and executing a SIMON event.

Figure 2-4: HVE Event Editor setting up and executing a SIMON event.

Setting Up an Event

SIMON uses the following event set-up options:

- Position/Velocity — Vehicles
- Position/Velocity — Humans
- Driver Controls
- Collision Pulse
- Vehicle Mesh
- Vehicle Payload
- Vehicle Wheels
- Accelerometers

The specific Event Set-up data used by SIMON are defined in Table 2-7.

Table 2-7 Event Set-up Parameters Used By SIMON.

Parameter	Description
Vehicle Initial Position	Earth-fixed X, Y, Z coordinates and roll, pitch, yaw angles of the vehicle sprung mass at the start of the simulation
Vehicle Initial Velocity	Total velocity and sideslip angle, and roll, pitch and yaw angular velocities at the start of the simulation
Human Initial Position	For occupants, the vehicle-fixed x, y, z coordinates and roll, pitch and yaw angles of the human pelvis segment; for pedestrians, the positions are defined in the earth-fixed coordinate system
Human Initial Velocity	Any initial velocity is ignored by SIMON
Driver Controls, Steer Table	Steer Table Option: Steering Wheel Angle vs. time, or Tire Steer Angle vs. time
Driver Controls, Brake Table	Brake Table Option: Brake Pedal Force vs. time, or Wheel Brake Force vs. time, or Percent Available Friction vs. time
Driver Controls, Throttle Table	Throttle Table Option: Percent Wide-open Throttle vs. time, or Tractive Force vs. time, or Percent Available Friction vs. time
Driver Controls, Gear Table	Gear Table Option: Transmission Gear vs. time, or Differential vs. time
Driver Controls, HVE Driver Model	Variable Steering control parameters, Neuro-muscular Filter
Collision Pulse	Pulse Table Type: vehicle-fixed force and moment vs. time, velocity vs. time, or position vs. time
Vehicle Mesh (DyMESH Only)	Weld Distance, Tessellation Length, Water-tight Status, Inter-vehicle Friction, Restitution
Payload	Location, Weight and roll, pitch and yaw moments of inertia of payload
Accelerometers	Vehicle-fixed x, y, z coordinates for up to five accelerometers
Wheels, Blow-out	Start time and Duration, stiffness factor, and rolling resistance factor for tire blow-out model
Wheels, Damage	Start time and Duration, Vehicle-fixed change in Wheel Coordinates and Camber, Percent of Lock-up Torque
Wheels, Brake	Initial Lining/Drum Temperatures, Stroke, and Start Time, Duration and Percent of Brake Failure
Wheels, Tire-Terrain	Tire-Terrain Model, Radial Spring Parameters, Sidewall Impact Parameters

Position/Velocity

Like all simulations, SIMON requires initial positions and velocities to be supplied by the user.

Unit vehicles and tow vehicles are positioned relative to the earth-fixed coordinate system by supplying the X, Y, Z coordinates of the CG, and roll (Φ), pitch (Θ) and yaw (Ψ) angles about vehicle x, y and z axes, respectively.

> **NOTE:** Z, roll and pitch are supplied automatically when AutoPosition is selected from the Options menu.

Trailers and dollys are positioned relative to their respective tow vehicle by supplying the initial roll, pitch and yaw articulation angles. Roll and pitch are determined by the environment if AutoPosition is selected. The default yaw articulation angle is 0.0 degrees.

The velocities of unit and tow vehicles are supplied in the form of a total velocity and sideslip angle.

> **NOTE:** The vehicle-fixed u (forward) and v (side) velocity components are calculated using the total velocity and sideslip angle. Vertical velocity is initially set to zero.

Trailer and dolly articulation angular velocities are assigned relative to the tow vehicle. The default angular velocities are 0.0 degrees/second.

For human occupants, positions are specified relative to the original, unloaded vehicle CG. However, HVE stores the positions relative to the CG altered to reflect the addition of humans and any payloads. Any initial velocity entered by the user is ignored by SIMON. The positions are used with the occupant body weights to adjust the weight and CG location of the sprung mass.

> **NOTE:** Although human pedestrians are not used by SIMON, they can be placed in events to add to the scene. Pedestrian positions are defined relative to the earth-fixed coordinate system.

Driver Controls

SIMON uses the following Driver Controls:

- **Steering** — A table of steering inputs as a function of time. All input options (*At Driver* and *At Axle*) are supported.

> **NOTE:** If selected, the Steer Degree of Freedom Model will override any user-entered steering inputs, unless the Append Option is selected.

- **Brakes** — A table of braking inputs as a function of time. All input options (*Pedal Force, Wheel Force* and *Percent Available Friction*) are supported.

> **NOTE:** The Pedal Force option is recommended (see Chapter 4, Calculation Method).

- **Throttle** — A table of throttle inputs as a function of time. All input options (*Percent Wide Open Throttle (%WOT), Wheel Force* and *Percent Available Friction*) are supported.

> **NOTE:** The %WOT option is recommended (see Chapter 4, Calculation Method).

- **Gears** — A table of instructions for shifting gears as a function of time.
- **HVE Driver Model** — Options for using the Path Follower, Speed Follower, and Neuro-muscular Filter. The HVE Driver Model is a closed-loop driver simulator which calculates driver input as a function of time (see Chapter 4 and [4]).

The Driver Controls Data used by SIMON are shown in Table 2-7.

Damage Profiles

The Damage Profiles data are not used by SIMON.

Collision Pulse

SIMON supports a user-entered collision pulse table, available by selecting Set-up, Collision Pulse for the selected vehicle. Three table types are available:

- **Force and moment vs. time** — vehicle-fixed force and moment components are applied at a user-specified vehicle-fixed location and added to the forces and moments acting on the sprung mass. The effect of non-modeled external forces (e.g., wind) may be studied using this option.
- **Velocity vs. time** — the vehicle's six velocity components are prescribed directly from the table.
- **Position vs. time** — the vehicle's earth-fixed position and orientation are prescribed directly from the table.

(updated: the Fifth Edition manual described only the force and moment table type.)

> **NOTE:** The velocity and position table types override the vehicle's equations of motion while the table runs, so results that depend on the computed forces should not be read as physical during that interval. See Chapter 4, *Collision Pulse Table*.

Vehicle Mesh

The Vehicle Mesh options used by SIMON include the ability to tessellate the vehicle mesh, assign inter-vehicle friction and relaxation length values between the selected object and target objects. These options are used in vehicle collision simulations by the DyMESH 3-D collision algorithm.

Vehicle Payload

The Payload Data used by SIMON include the payload x, y, z coordinates relative to the vehicle's CG, and the payload rotational inertias. One payload may be entered per vehicle. SIMON computes a new sprung mass CG location and new sprung mass rotational inertias based on these data.

Vehicle Wheels

SIMON supports the Tire Blow-out Model, Wheel Damage option, HVE Brake Designer (HVE User's Manual, See Section Eleven and References 1 and 10) and the Tire-Terrain Model options. These options allow the user to study event-related phenomena at each wheel. The individual parameters are shown in Table 2-7. Also, refer to the HVE User's Manual, Menu Reference, Edit Menu, for detailed information regarding these options.

Accelerometers

SIMON allows the user to place virtual accelerometers in the vehicle at up to five vehicle-fixed locations. These accelerometers can be used to track the total acceleration and acceleration components at various locations within the vehicle (e.g., specific seat locations). The accelerometer outputs are included in the Variable Output, Accelerometer group (see Chapter 3, Variable Output).

Contacts

The Contacts data are not used by SIMON.

Restraints

The Restraints data are not used by SIMON.

Event Calculation Options

(updated: the SIMON Calculation Options dialog has grown considerably since the Fifth Edition manual, which described only three options — Tire Model Method, Steer DOF and Collision Acceleration Threshold.)

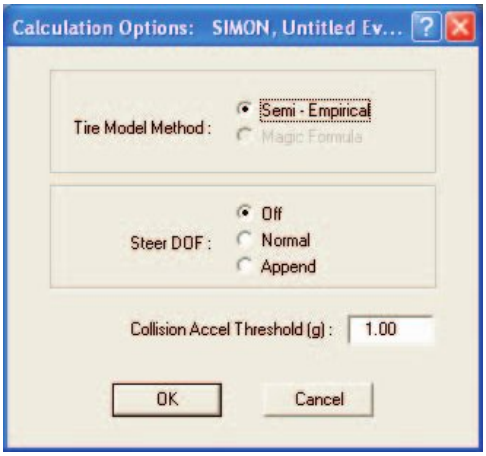


Figure 2-5: SIMON Calculation Options Dialog (legacy version).

SIMON's Calculation Options include:

- **Tire Model Method** — A radio button defining the version of the EDC semi-empirical tire model used for the event. The current dialog offers *Semi-Empirical, Vers. 1*, *Semi-Empirical, Vers. 2* and *Semi-Empirical, Vers. 3*. **Vers. 3 is the default for a new event** and is the recommended choice; Vers. 1 and Vers. 2 are retained so that results from earlier releases can be reproduced. The three versions share the same structure and differ only in the treatment of the slip-angle function, the direction of the lateral force when a tire travels rearward, the limit placed on effective friction, and the handling of cornering and camber stiffness at very low wheel speeds — see Chapter 4, *Tire Model*, for the details. The selected version affects only the tire shear-force calculation; it does not change the radial force, rolling resistance or hydroplaning models. *(updated: the Fifth Edition manual showed only "Semi-Empirical" and a disabled "Magic Formula" option; the current dialog instead offers three versions of the semi-empirical tire model.)*
- **Steer DOF** (Off/Normal/Append/AutoStart) — Selecting this option allows tire forces to affect wheel heading. This is equivalent to the driver taking their hands off the steering wheel. With *AutoStart*, the wheels follow the steer table until the vehicle is involved in a collision and respond to tire forces thereafter; both vehicles in the collision start together. See Chapter 4, *Steering System*, for what each setting does and for the moments acting on the steering system. *(updated: the AutoStart option has been added since the Fifth Edition manual.)*

> **NOTE:** If Normal Steer DOF is selected, any user-entered driver control inputs will be ignored. If Append Steer DOF is selected, SIMON switches from user-entered driver control input to the Steer DOF at the time of the last entry in the Driver Steering Table.
>
> **NOTE:** Using the Append option provides a powerful tool for simulating post-collision vehicle behavior during which the driver's hands come off the steering wheel.

- **Hydroplaning Model** (Off / NASA / NASA-TTI / Gallaway) — Selects the model used to compute tire hydroplaning on flooded road surfaces. The default is Off. Hydroplaning is evaluated only for tires in contact with an environment polygon whose object type is **Water**, and it requires the tire's *Nominal Pressure*, *Tread Width* and *Tread Depth* (Tire Physical Data); the Gallaway model additionally uses the water depth and the pavement macrotexture. Above the model's threshold speed the tire's friction switches to the water polygon's friction value. Hydroplaning cannot be combined with the Soft Soil tire-terrain model. See Chapter 4 — Hydroplaning Model for the threshold equations and reporting. *(updated: option added since the Fifth Edition manual.)*
- **Connection Model** (Use Both Vehicles / Use Heavier Vehicle / Use Tow Veh Properties) — Selects which vehicle's properties govern the stiffness of an inter-vehicle connection (e.g., tractor-trailer articulation). The default is Use Heavier Vehicle. *(updated: option added since the Fifth Edition manual.)*
- **Connection Failure Start Time** — The earliest simulation time (sec) at which an inter-vehicle connection is permitted to fail. *(updated: option added since the Fifth Edition manual.)*
- **Solid Axle Inertia** (Use Axle Only / Use Axle + Wheels) — Selects whether solid-axle roll inertia is computed from the axle alone or from the combined axle and wheel inertias. The default is Use Axle + Wheels. *(updated: option added since the Fifth Edition manual.)*
- **Accident History Basis** (*Use Impact Force* / *Use Impact Acceleration*) — Selects the basis used by SIMON to detect the separation time reported in the Accident History report. With *Use Impact Force* (the default), the collision phase ends when collision contact between the two vehicles is lost. With *Use Impact Acceleration*, it ends either when contact is lost or when both vehicles' total acceleration falls below the Collision Acceleration Threshold, whichever comes first. The impact time is detected from collision contact in both cases. See Chapter 4, *Sprung Mass Impact Model — Impact and separation*. *(updated: option added since the Fifth Edition manual. Earlier descriptions stated that this option governs the detection of impact as well as separation; it affects separation only.)*
- **Collision Acceleration Threshold** — The total acceleration below which both vehicles must fall for the collision phase to be considered over. *(updated: in the current dialog this field is active only when the Accident History Basis is set to Use Impact Acceleration; its default value is the environment's gravity constant, i.e., 1 g. It is used to end the collision phase, not to begin it, and it does not gate the delta-V calculation.)*
- **Include Free Space** — A checkbox controlling how crush depth is reported in the Damage Data report. When unchecked (the default), the crush tables report crush depth excluding free space (*Crush Depth, Excl. Free Space*); when checked, the reported crush depth includes the free space between the exterior surface and the underlying crushable structure (*Crush Depth, Incl. Free Space*). *(updated: option added since the Fifth Edition manual.)*

DyMESH Options

(updated: this section is new. In the current version of HVE, SIMON events include a separate DyMESH Options dialog, which controls the DyMESH 3-D collision algorithm. SIMON is the HVE simulation model that uses DyMESH.)

The DyMESH Options dialog contains the following options:

- **Use DyMESH** — Main checkbox enabling the DyMESH 3-D collision algorithm for the event. This option requires a DyMESH license; all remaining options are available only when Use DyMESH is checked.
- **Include Environment** — Extends DyMESH collision detection to include vehicle-environment mesh contact (e.g., striking a rigid barrier or terrain feature), in addition to vehicle-vehicle contact.
- **Force To x-y Plane** — Keeps the DyMESH deformation (pushback) direction horizontal, which helps prevent a severe over-ride in very high-speed collisions. It is not applied to contact with the environment.
- **Tow Vehicle / Trailer Contact** — Enables DyMESH mesh contact between vehicles joined by an inter-vehicle connection (e.g., jackknife contact between a tractor and its trailer).
- **DyMESH Version No** (Version 3 / Version 4) — Selects the version of the DyMESH collision algorithm. Version 4 is the default for new events.
- **DyMESH Start Time** (sec) — Simulation time at which DyMESH vehicle-vehicle collision detection begins.
- **DyMESH Environment Start Time** (sec) — Simulation time at which DyMESH vehicle-environment collision detection begins.
- **Search Option** (Automatic / Set Box Size) — Selects the contact search method. Automatic searches the neighbors of the penetrated polygon; Set Box Size uses the user-entered box size (in) to define the search region around each vertex.
- **Force Smoothing** (Version 1 and 2 / Version 3 and Later) — Selects how collision forces are averaged between timesteps as DyMESH alternates the primary and secondary roles (see Chapter 4 — Primary and secondary vehicles). Version 3 and Later is the default; Version 1 and 2 reproduces results from earlier releases.

(updated: earlier descriptions of this dialog listed General and Advanced pages and described Force Smoothing as a mesh smoothing model. The current dialog is a single page, and the setting smooths the collision force, not the mesh. Settings that earlier releases offered on an Advanced page are fixed in the current release; see the HVE User's Manual, Options Menu, DyMESH.)

Simulation Controls

SIMON uses the current simulation control parameters in the Simulation Controls dialog (see Options Menu, Simulation Controls). The Simulation parameters used by SIMON are shown in Table 2-8.

> **NOTE:** The output time interval should be an even multiple of the vehicle trajectory integration timestep. For example, if the integration timestep is 0.02, the output interval might be 5 × 0.02, or 0.10.

Table 2-8 Simulation Control Parameters Used By SIMON.

Parameter	Description
Vehicle Trajectory Integration Timestep	The integration timestep used by the numerical integration routine while no collision is in progress
Collision Integration Timestep	The integration timestep used while any pair of objects in the event is in contact. Collision forces change far more rapidly than trajectory forces, so a shorter timestep is used during the collision phase. See Chapter 4, *Sprung Mass Impact Model — Collision integration timestep*
Output Interval	The timestep used to send output results back to HVE
Maximum Simulation Time	Maximum length of the run
Min Linear Velocity, Min Angular Velocity	The linear and angular velocities used to terminate the run (if the linear and angular velocities for all vehicles are simultaneously less than these termination velocities, the run terminates)

Get Surface Information Options

SIMON uses a surface-information search to determine on which environment polygon each tire is riding. This search looks below the X, Y coordinates of each tire contact patch (in the case of the Point Contact Tire-Terrain Model) or in the direction of the radial and sidewall springs (in the case of the Radial Spring and Sidewall Impact Tire-Terrain Models) to determine the friction and other characteristics of the terrain in the vicinity of the tire.

The default search method is

From Previous Polygon

. This results in fastest execution, but does not work for all environments. Refer to the User's Manual for further information about how the Get Surface Information options work.

When the Sidewall Impact Tire-Terrain Model is used, the Search Direction Z component may need to be set to a small, positive value, say 0.001, to account for small rounding errors in the curb geometry.

Executing An Event

To execute a SIMON event, use the Event Controller, shown in Figure 2-4 as a component of the Event Editor, and again in Figure 2-6. The Event Controller's buttons have the following functions:

- **Reset** — Reinitialize the calculation model for re-execution
- **Rewind to Start** — Return to the start of the simulation
- **Reverse** — Play the simulation backwards
- **Pause** — Pause the simulation
- **Play** — Execute the event or play the simulation forwards
- **Advance to End** — Advance to the end of the simulation



Figure 2-6: Event Controller.

> **NOTE:** If you make any changes to an event (humans, vehicles, environments, or event set-up options), those changes will have no effect unless you press Reset before pressing Play.
>
> **NOTE:** Remember to use HVE's Options Menu to choose useful options, such as Key Results, Axes, Velocity Vectors and Skidmarks.

Chapter 3 — SIMON Program Output

This chapter defines the outputs available from a SIMON event. The reports produced by SIMON are available in the HVE Playback Editor.

Overview

SIMON produces three types of output reports:

- **Alpha-Numeric Reports** — Reports containing text and numeric information, such as vehicle dimensional parameters
- **Variable Output Tables** — Reports containing tabular simulation results as a function of time
- **Trajectory Simulations** — Viewers containing dynamic, 3-D visual simulations

> **NOTE:** Each of these reports may be printed on the system printer. To print a report, click on the menu bar of the desired output report (the menu bar will change colors indicating that it is selected), then either choose Print from HVE's Files menu or click on the Print icon in the toolbar. Refer to the HVE User's Manual for further details.

To view any of these reports, perform the following steps:

1. Choose Playback mode. The Playback Editor is displayed.
2. Choose *Add New Object*. The Report Window Information dialog is displayed, showing a list of all the current events in the case.
3. Select a SIMON event from the Active Events List. Once an event is selected, the Report Type list is displayed, containing all the available reports for the selected event.
4. Select the desired report in the Report Type list. To open several reports at once, select more than one (*Select All* selects every report); each opens in its own Report Window. The available output reports are:
5. Accident History
6. Damage Data
7. Damage Profiles
8. Driver Controls
9. Environment Data
10. Event Data
11. Human Data
12. Messages
13. Program Data
14. Trajectory Simulation
15. Variable Output
16. Vehicle Data
17. If desired, edit the Name. A default name is supplied for the report window. The name is user-editable, and does not affect calculations.

> **NOTE:** Duplicate Report Window names are not allowed. Because HVE truncates the name to 30 characters, you should ensure that two truncated names are not the same.

1. Click *OK* to display the report.

Alpha-Numeric Reports

SIMON produces the following alpha-numeric reports:

- **Accident History** — A table of initial, impact, separation and final positions and velocities
- **Damage Data** — A table of vehicle collision data, such as delta-V
- **Driver Data** — A series of tables containing driver input data (steering, brakes, throttle) used by SIMON
- **Environment Data** — A series of tables containing the environment data used by SIMON
- **Event Data** — A series of tables containing the event data used by SIMON
- **Human Data** — A series of tables containing the human locations and weights
- **Messages** — A list of messages produced by the current run
- **Program Data** — A table containing program control information for the current run
- **Vehicle Data** — A series of tables containing the vehicle data used by SIMON

An example of each of these numeric output reports from SIMON is shown on the following pages.

Accident History

The Accident History Report displays a table of initial, impact, separation and final positions and velocities for the vehicle. A typical Accident History Report is shown in Figure 3-1.

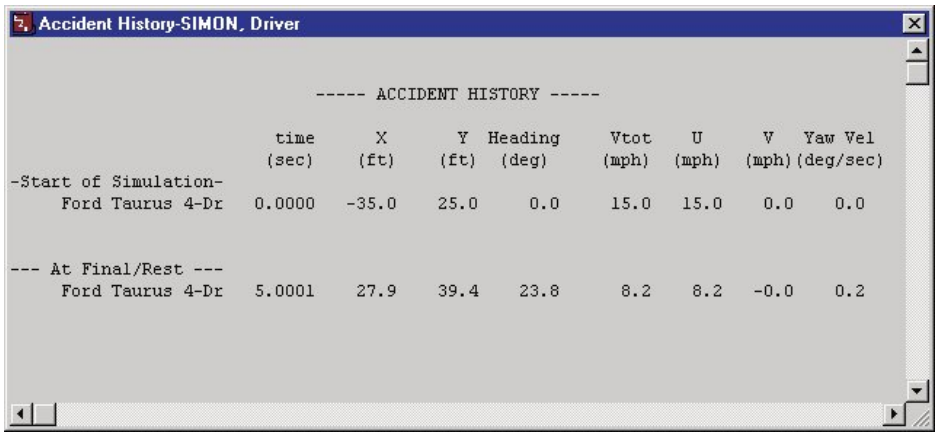


Figure 3-1: Typical Accident History Output Report issued by SIMON.

*(updated: the current SIMON Calculation Options dialog includes an **Accident History Basis** option. Impact and separation rows in the Accident History report may be detected either from **Impact Force** or from vehicle **Acceleration** exceeding a user-set threshold; see Chapter 2, Event Calculation Options.)*

Damage Data

The Damage Data Report displays a table containing vehicle collision information, such as delta-V. A typical Damage Data Report is shown in Figure 3-2.

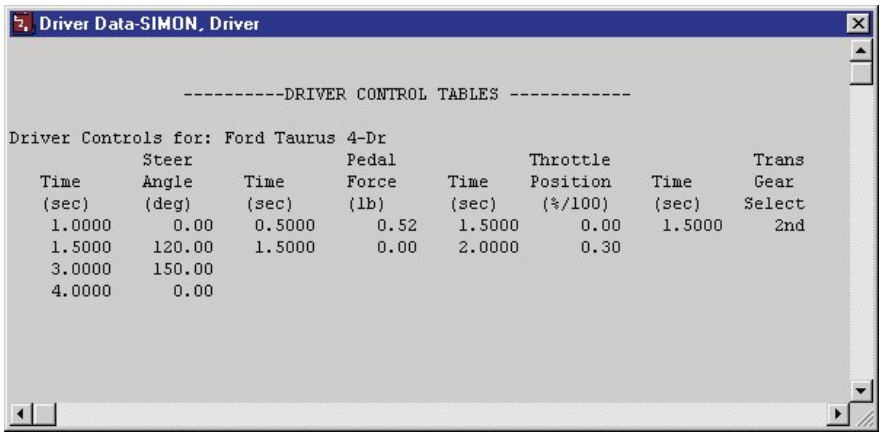


Figure 3-2: Typical Damage Data Report issued by SIMON.

*(updated: the current Damage Data report also includes CDC(s), collision impulse data — up to 10 impulses per vehicle — and crush depth tables. When the **Include Free Space** calculation option is changed, the crush tables switch between "Excl. Free Space" and "Incl. Free Space" measurements; see Chapter 2, Event Calculation Options.)*

Driver Data

Individual Driver Control tables for steering, braking, throttle and gear selection used by SIMON in the current event. A typical Driver Controls Output Report is shown in Figure 3-3.

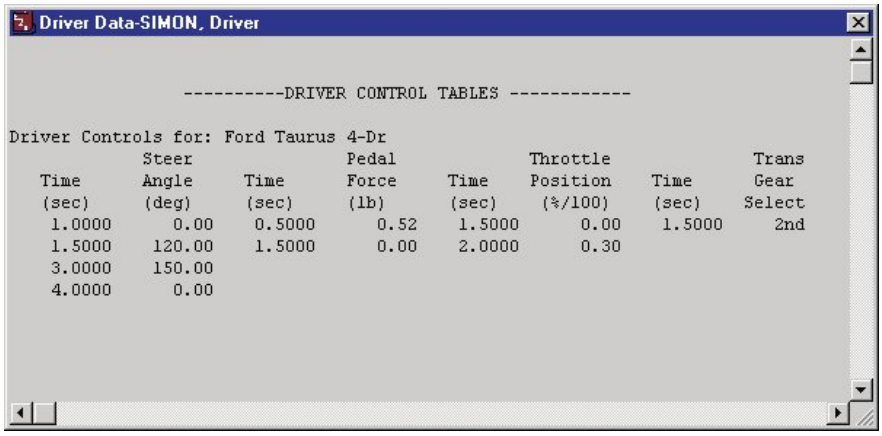


Figure 3-3: Typical Driver Controls Output Report issued by SIMON.

Environment Data

The Environment Data Output Report consists of two sections of data:

- **General Environment Data** — Environmental parameters used by SIMON including air temperature, pressure and density; wind speed and direction; and gravity constant
- **3-D Environment Terrain Data** — Name of and information about the environment geometry file

A typical Environment Data Output Report is shown in Figure 3-4.

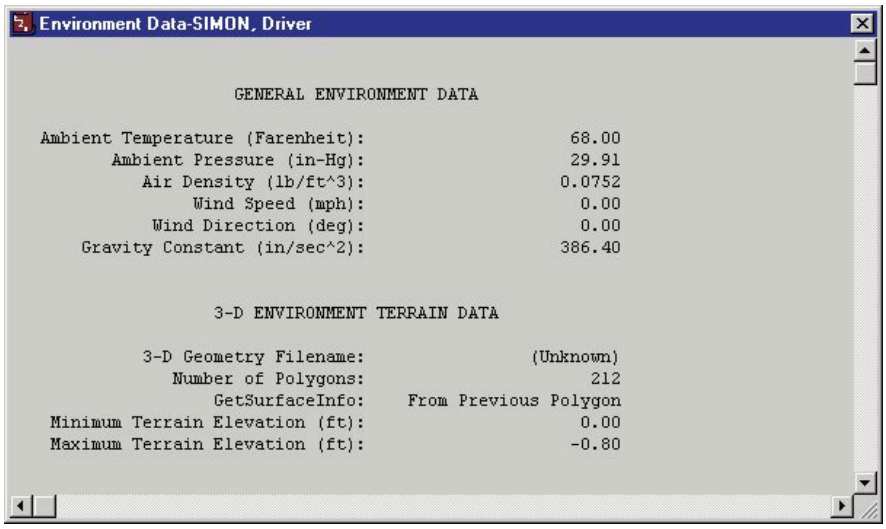


Figure 3-4: Typical Environment Data Output Report issued by SIMON.

Event Data

The Event Data Output Report lists event-specific data for each vehicle. These data include:

- **Payload Information** — Mass and location of optional payload
- **Accelerometer Information** — Location of optional accelerometers
- **Wheel Data** — Information regarding optional wheel displacements, brake assembly, and tire blow-outs for each wheel location

A typical Event Data Output Report is shown in Figure 3-5.

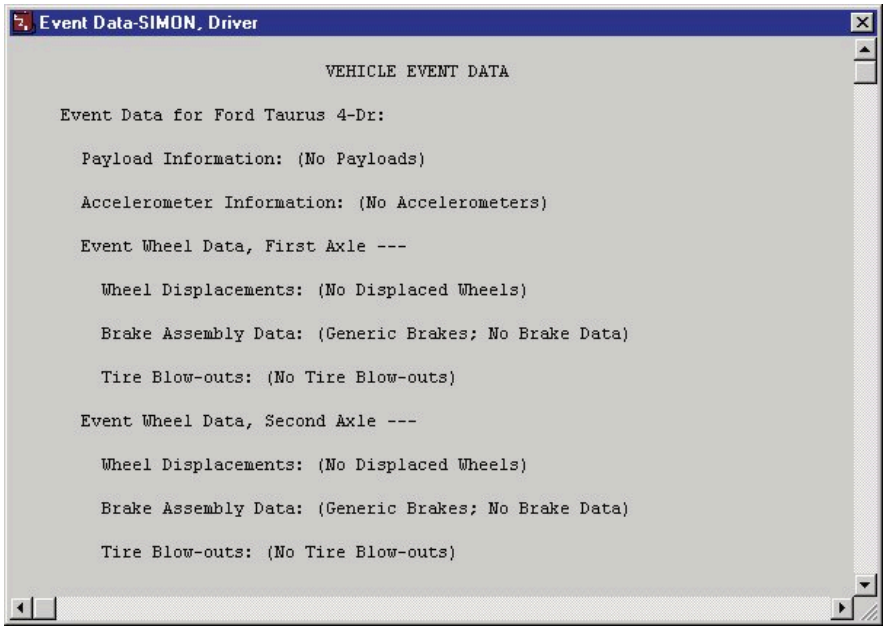


Figure 3-5: A portion of a typical Event Data Output Report issued by SIMON.

Human Data

A typical Human Data Output Report is shown in Figure 3-6. Human data includes total weight, vehicle-fixed x, y, z location, and seat position for occupants, and earth-fixed X, Y, Z location for pedestrians.

> **NOTE:** Coordinates are for the pelvis CG.

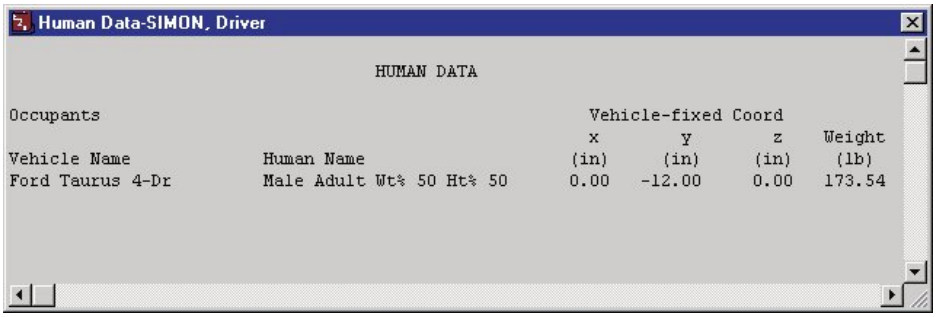


Figure 3-6: Typical Human Data Output Report issued by SIMON.

Messages

A typical Messages Report is shown in Figure 3-7. For a complete listing of messages issued by SIMON, see Chapter 6.



Figure 3-7: Typical Messages Output Report issued by SIMON.

Program Data

The Program Data Report includes the following information:

- **Simulation Controls** — Integration parameters used for the current event (integration method, maximum simulation time, integration timestep, output interval, linear and angular termination velocities)
- **Calculation Options** — Get Surface Information option, Tire Model Method, Steer Degree of Freedom, Articulation Option, DyMESH Option

- **DyMESH Options** — When DyMESH is used, the report lists the DyMESH options in effect (e.g., Restitution Model, Search Option, Pushback Option, Inter-vehicle Friction, Crush Height For Stiffness)

A typical Program Data Report is shown in Figure 3-8.

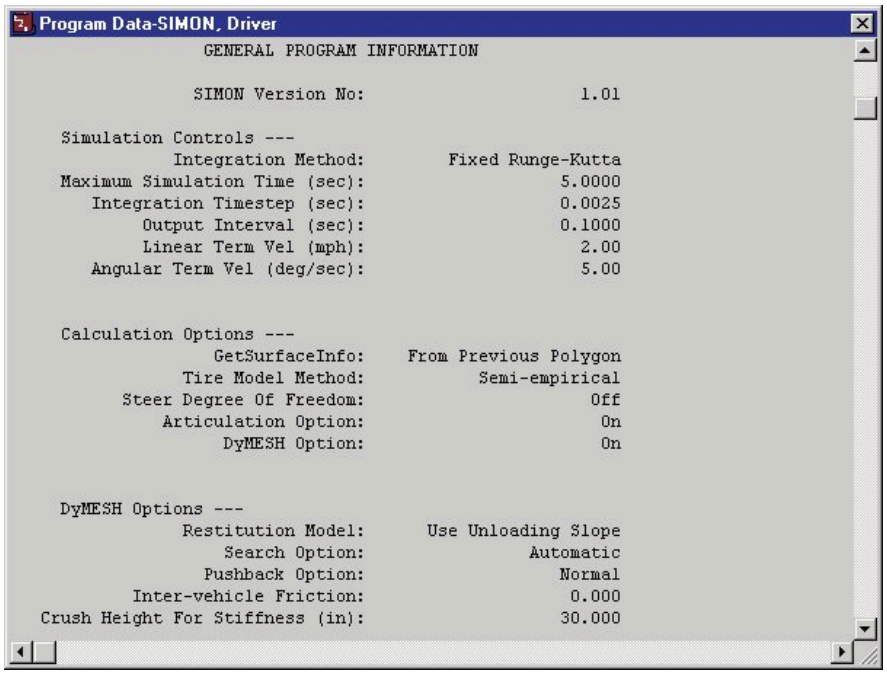


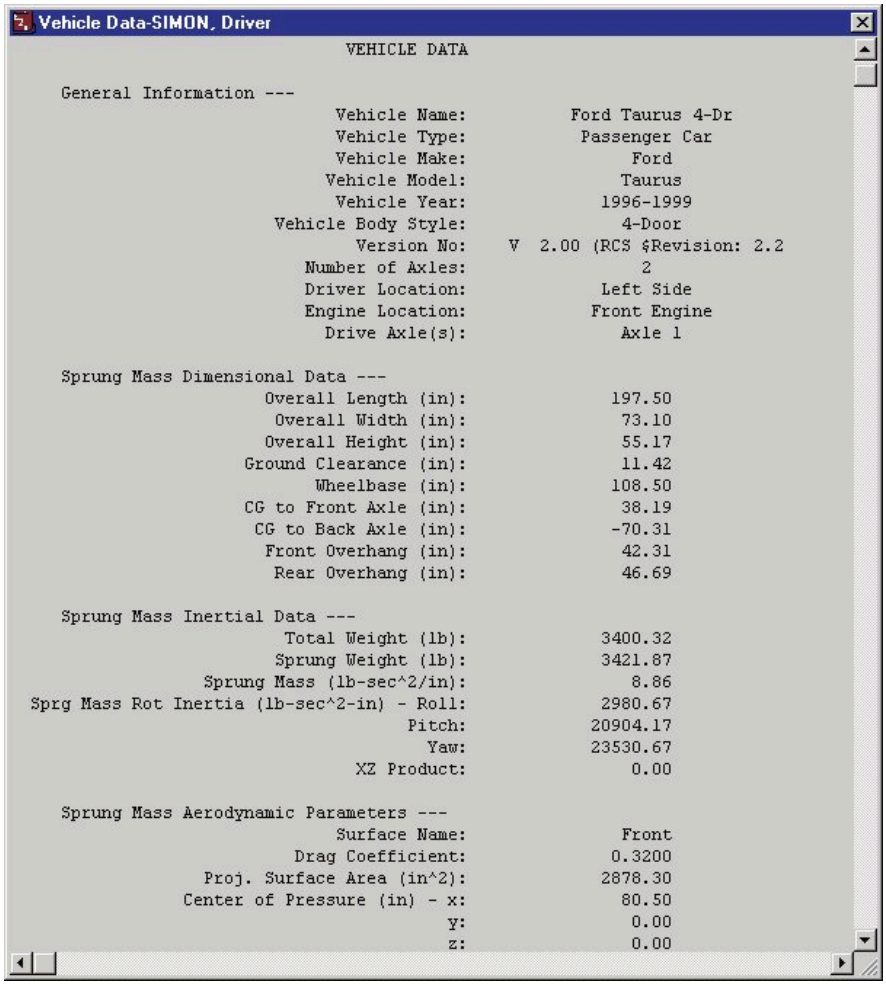
Figure 3-8: Typical Program Data Output Report issued by SIMON.

Vehicle Data

The Vehicle Data Report includes the following information:

- **Vehicle General Information** — Includes make, model, year, body style, basic vehicle configuration, and specific EDC vehicle version number
- **Sprung Mass Dimensional and Inertial Properties** — The dimensional and inertial parameters used by SIMON in the current event.
- **Connection Properties** — The connection parameters used by SIMON in the current event.
- **Aerodynamic Properties** — Sprung mass aerodynamic parameters used by SIMON in the current event.
- **Brake, Steering, and Drivetrain Properties** — Brake, steering, and drivetrain parameters (including engine, transmission and differential) used by SIMON in the current event.
- **Tire and Wheel Properties** — The tire and wheel parameters used by SIMON in the current event
- **Suspension Properties** — The suspension parameters used by SIMON in the current event.

A portion of a typical Vehicle Data Output Report is shown in Figure 3-9.



Vehicle Data-SIMON, Driver

```
                          VEHICLE DATA

   General Information ---
                              Vehicle Name:          Ford Taurus 4-Dr
                              Vehicle Type:           Passenger Car
                              Vehicle Make:                   Ford
                             Vehicle Model:                 Taurus
                              Vehicle Year:              1996-1999
                         Vehicle Body Style:                 4-Door
                                Version No:    V  2.00 (RCS $Revision: 2.2
                           Number of Axles:                      2
                           Driver Location:              Left Side
                           Engine Location:             Front Engine
                             Drive Axle(s):                 Axle 1

   Sprung Mass Dimensional Data ---
                         Overall Length (in):               197.50
                          Overall Width (in):                73.10
                         Overall Height (in):                55.17
                       Ground Clearance (in):                11.42
                              Wheelbase (in):               108.50
                       CG to Front Axle (in):                38.19
                        CG to Back Axle (in):               -70.31
                         Front Overhang (in):                42.31
                          Rear Overhang (in):                46.69

   Sprung Mass Inertial Data ---
                           Total Weight (lb):              3400.32
                          Sprung Weight (lb):              3421.87
                    Sprung Mass (lb-sec^2/in):                 8.86
Sprg Mass Rot Inertia (lb-sec^2-in) - Roll:                2980.67
                                      Pitch:               20904.17
                                        Yaw:               23530.67
                                 XZ Product:                   0.00

   Sprung Mass Aerodynamic Parameters ---
                                Surface Name:                Front
                            Drag Coefficient:               0.3200
                  Proj. Surface Area (in^2):               2878.30
                Center of Pressure (in) - x:                 80.50
                                          y:                  0.00
                                          z:                  0.00
```

Figure 3-9: A portion of a typical Vehicle Data Output Report issued by SIMON.

Graphic Reports

SIMON produces no Graphic Output Reports.

> **NOTE:** Graphs of simulation results vs time may be produced using the Variable Output window (see next section).

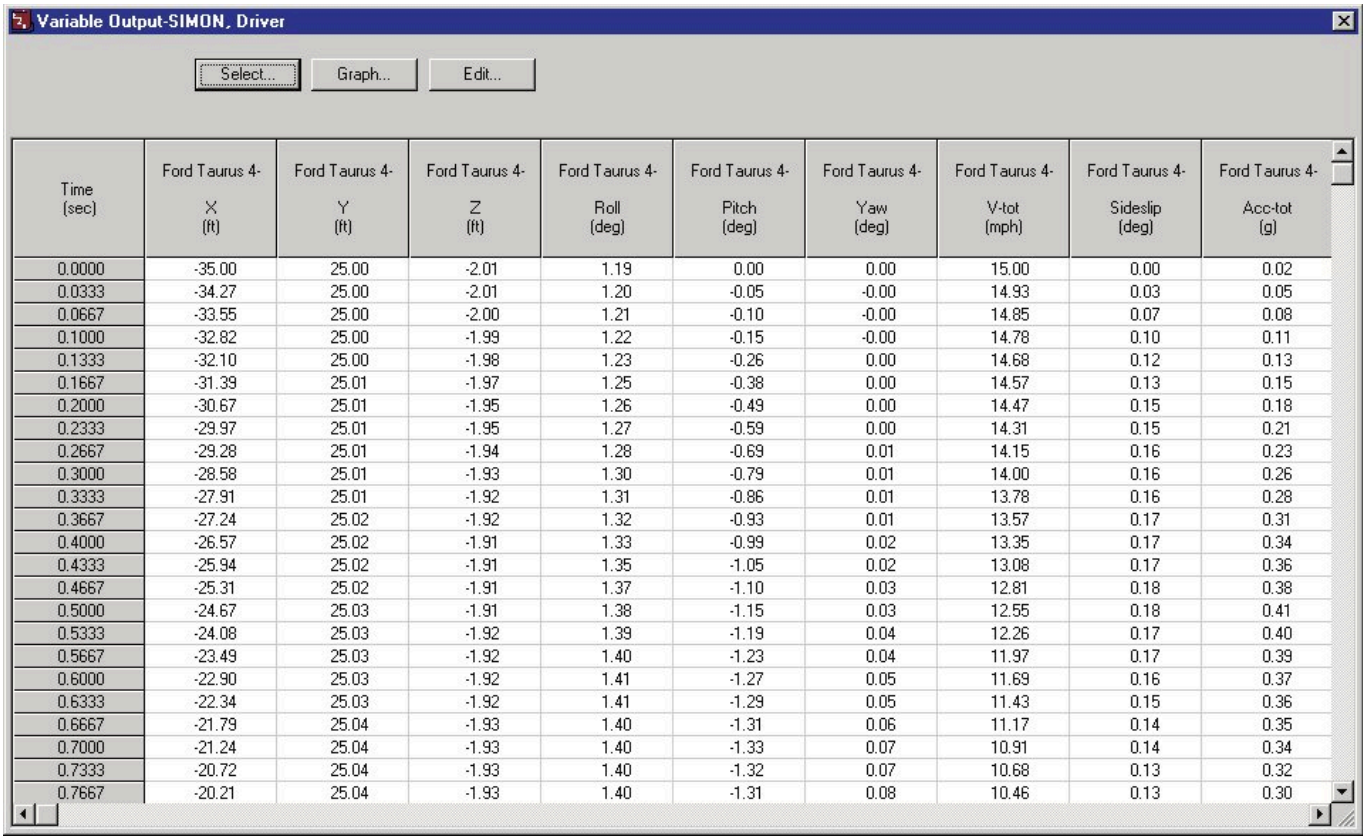
Variable Output Table

SIMON produces a Variable Output table containing the time-based simulation results. The Variable Output groups produced by SIMON are as follows:

Vehicle Output Groups

- **Kinematics** — Position, velocity and acceleration for the vehicle
- **Kinetics** — Summation of tire, drag and collision forces and moments acting at the CG of each vehicle
- **Accelerometers** — Linear accelerations at user-specified locations
- **Tires** — The tire output parameters existing at the tire contact patch (compare with Wheel Output, below)
- **Wheels** — The wheel output parameters existing at the wheel's hub (compare with Tire output, above)
- **Connections** — Articulation angles and inter-vehicle connection forces
- **Drivetrain** — Engine and transmission parameters used by SIMON
- **Driver** — Current levels of driver inputs (steering, braking and throttle)

An example of a Variable Output table is shown in Figure 3-10. Detailed listings of each Variable Output parameter produced by SIMON are found in Tables 3-1, 3-2, and 3-3. For more information about HVE Variable Output parameters, refer to the HVE User's Manual, Chapter 16, Event Model.



Time (sec)	Ford Taurus 4- X (ft)	Ford Taurus 4- Y (ft)	Ford Taurus 4- Z (ft)	Ford Taurus 4- Roll (deg)	Ford Taurus 4- Pitch (deg)	Ford Taurus 4- Yaw (deg)	Ford Taurus 4- V-tot (mph)	Ford Taurus 4- Sideslip (deg)	Ford Taurus 4- Acc-tot (g)
0.0000	-35.00	25.00	-2.01	1.19	0.00	0.00	15.00	0.00	0.02
0.0333	-34.27	25.00	-2.01	1.20	-0.05	-0.00	14.93	0.03	0.05
0.0667	-33.55	25.00	-2.00	1.21	-0.10	-0.00	14.85	0.07	0.08
0.1000	-32.82	25.00	-1.99	1.22	-0.15	-0.00	14.78	0.10	0.11
0.1333	-32.10	25.00	-1.98	1.23	-0.26	0.00	14.68	0.12	0.13
0.1667	-31.39	25.01	-1.97	1.25	-0.38	0.00	14.57	0.13	0.15
0.2000	-30.67	25.01	-1.95	1.26	-0.49	0.00	14.47	0.15	0.18
0.2333	-29.97	25.01	-1.95	1.27	-0.59	0.00	14.31	0.15	0.21
0.2667	-29.28	25.01	-1.94	1.28	-0.69	0.01	14.15	0.16	0.23
0.3000	-28.58	25.01	-1.93	1.30	-0.79	0.01	14.00	0.16	0.26
0.3333	-27.91	25.01	-1.92	1.31	-0.86	0.01	13.78	0.16	0.28
0.3667	-27.24	25.02	-1.92	1.32	-0.93	0.01	13.57	0.17	0.31
0.4000	-26.57	25.02	-1.91	1.33	-0.99	0.02	13.35	0.17	0.34
0.4333	-25.94	25.02	-1.91	1.35	-1.05	0.02	13.08	0.17	0.36
0.4667	-25.31	25.02	-1.91	1.37	-1.10	0.03	12.81	0.18	0.38
0.5000	-24.67	25.03	-1.91	1.38	-1.15	0.03	12.55	0.18	0.41
0.5333	-24.08	25.03	-1.92	1.39	-1.19	0.04	12.26	0.17	0.40
0.5667	-23.49	25.03	-1.92	1.40	-1.23	0.04	11.97	0.17	0.39
0.6000	-22.90	25.03	-1.92	1.41	-1.27	0.05	11.69	0.16	0.37
0.6333	-22.34	25.03	-1.92	1.41	-1.29	0.05	11.43	0.15	0.36
0.6667	-21.79	25.04	-1.93	1.40	-1.31	0.06	11.17	0.14	0.35
0.7000	-21.24	25.04	-1.93	1.40	-1.33	0.07	10.91	0.14	0.34
0.7333	-20.72	25.04	-1.93	1.40	-1.32	0.07	10.68	0.13	0.32
0.7667	-20.21	25.04	-1.93	1.40	-1.31	0.08	10.46	0.13	0.30

Figure 3-10: Typical Variable Output Table from SIMON.

Table 3-1 General Vehicle Variable Output Table

Parameter	Description
Vehicle Kinematic Data	X, Y, Z position of CG; Φ, Θ, Ψ orientation; ν (Course Angle); Total linear velocity; u, v, w components; Sideslip angle; p, q, r angular velocities; Total linear accel; forward, lat, vert components; u-dot, v-dot, w-dot linear components; p-dot, q-dot, r-dot angular accelerations
Vehicle Kinetic Data	F_x, F_y, F_z Suspension; M_x, M_y, M_z Suspension; F_x, F_y, F_z Impact; M_x, M_y, M_z Impact; F_x, F_y, F_z Aerodynamic; M_x, M_y, M_z Aerodynamic; F_x, F_y, F_z Connection; M_x, M_y, M_z Connection
Accelerometer Data	Total linear accel; forward, lat, vert components for each accelerometer
Damage Data *	Vertex is changed (Flag); Vertex ID; x, y, z vertex coordinates; F_x, F_y, F_z vertex force components

* Not selectable by user.

Table 3-2 Vehicle Component Variable Output Table

Parameter	Description
Tire Data	X, Y, Z position of tire contact patch; F'_x, F'_y, F'_z; Tire radius, Tire Deflection; Longitudinal slip; Slip angle; Inclination angle; Skid flag; Friction multiplier; u_x, u_y, u_z (unit vector for surface normal); Soil Constants N, Kphi, Kc (Soft Soil Tire-Terrain Model); F'_x, F'_y Flow Force (Soft Soil Model); Fx Sidewall, Fy Sidewall, Fz Sidewall; Water Depth, Macrotexture, Mu Mult (water) (hydroplaning models)
Wheel Data	x, y, z location of each wheel center; Axle z deflection from equilibrium (solid axle); Axle roll (solid axle); Vehicle-fixed wheel camber; Wheel spin displacement; Steer angle (δ) at each steerable wheel; Axle z-dot deflection rate (solid axle); Axle roll rate (solid axle); Vehicle-fixed wheel camber rate; Wheel spin velocity; Steer velocity at each steerable wheel; Axle z-dot-dot deflection acceleration (solid axle); Axle roll acceleration (solid axle); Wheel spin acceleration; Steer acceleration at each steerable wheel (steer DOF only); F_x, F_y, F_z (wheel); Steer axis moment at each steerable wheel (steer DOF only); Spring deflection; Spring deflection rate; Total suspension force; Individual spring and damper suspension forces; Drive, brake torque; Auxiliary stiffness from sway bar; Brake stroke, pressure; Brake interface, drum, lining temperature
Connection Data	Φ, Θ, Ψ articulation; F_x, F_y, F_z; M_x, M_y, M_z
Drivetrain Data	Engine speed; Engine power; Engine torque; Transmission ratio; Differential ratio

(updated: when a hydroplaning model is active, the Water Depth, Macrotexture and water friction-multiplier tire tracks are populated. The Rolling Resist Mult, Fx' (water) and Fy' (water) tracks are not used. See Chapter 4 — Hydroplaning Model.)

Table 3-3 Vehicle Driver Variable Output Table

Parameter	Description
Driver Data (for "At Driver" Driver Table options only)	Throttle position; Brake pedal; Brake system; Transmission, differential gear; Steer angle (all options)
Driver Data (for HVE Driver Model)	Steering Wheel Angular Velocity; CG Path Error; Preview Distance; Path error at preview distance; X, Y at preview distance

Trajectory Simulations

SIMON produces a trajectory simulation and a damage profile (if a vehicle collision has been simulated using DyMESH) of the current event. The trajectory simulation is a 3-D visualization of the data displayed in the Variable Output table (see previous section). An example of a trajectory simulation is shown in Figure 3-11. The damage profile is a time-domain 3-D visualization of the deformation of the vehicle mesh calculated by DyMESH during the vehicle collision.

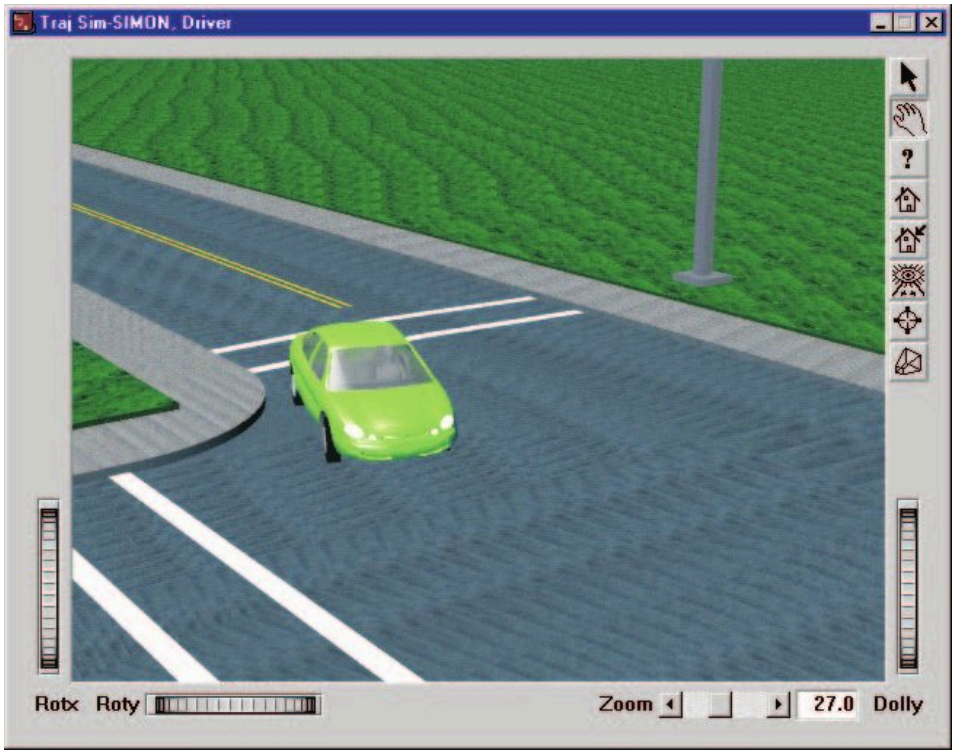


Figure 3-11: Typical Trajectory Simulation Report issued by SIMON.

Displaying a Trajectory Simulation

The Trajectory Simulation is controlled using the Playback Controller (Figure 3-12).



Figure 3-12: Playback Controller.

The Playback Controller's buttons have the following functions:

- **Reset** — Returns to the start of the simulation and reinitializes the video output device (this applies a hardware reset and is otherwise the same as the *Rewind to Start* button, below).
- **Rewind to Start** — Return to the start of the simulation
- **Reverse** — Play the simulation backwards
- **Pause** — Pause the simulation
- **Play** — Execute the event or play the simulation forwards
- **Advance to End** — Advance to the end of the simulation

> **NOTE:** The Playback Controller also includes additional features used for creating video. Refer to the HVE User's Manual, Playback Editor and Video Output sections, for further details.

Displaying a Damage Profile

You must have a Trajectory Simulation open in order to play the Damage Profile and, thus, view the vehicle damage. With both the Damage Profile and Trajectory Simulation viewers open, the Damage Profile viewer is controlled with the Playback Controller. In fact, you will notice the Trajectory Simulation playing at the same time.

Chapter 4 — Calculation Method

Basis For Analysis

SIMON is a dynamic simulation analysis of one or more vehicles. SIMON is a simulation model incorporating a general purpose 3-D dynamics engine developed by Engineering Dynamics Company, LLC. SIMON includes a sprung mass with six degrees of freedom (x, y, z, roll, pitch, yaw) and unsprung masses with three degrees of freedom per axle (axle roll, z and steer for solid axle suspensions; axle steer and right and left wheel z for independent suspensions).

The vehicle model includes the following basic components for each vehicle unit:

- Sprung mass
- Up to three axles (full trailers may have up to six axles)
- Independent or solid axle suspension for each axle
- Single or dual tires at each wheel location
- Front and rear inter-vehicle connections
- Exterior geometry
- Aerodynamic surface(s)
- Payload(s)
- Human occupant(s)
- Steering system
- Braking system
- Drivetrain

The basic vehicle model is shown in Figure 4-1.

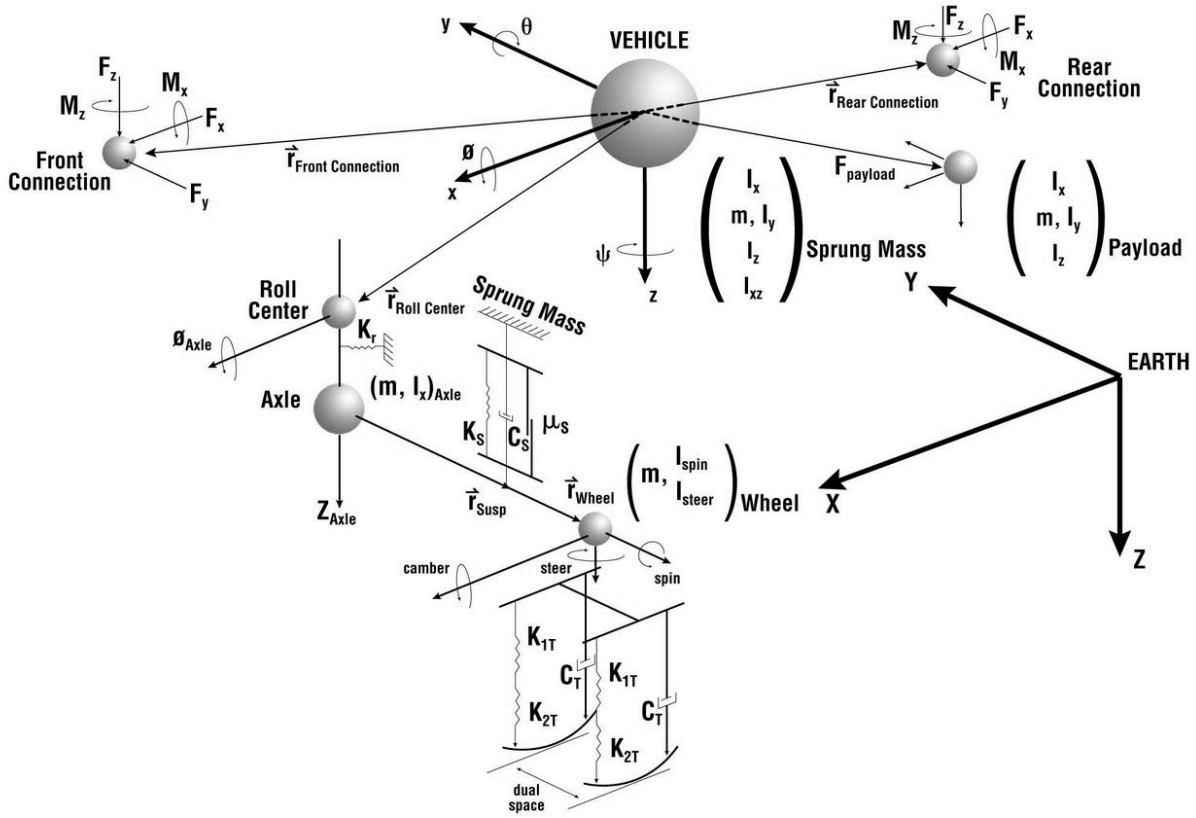


Figure 4-1: SIMON Vehicle Model.

Equations of Motion

The system of equations requires three coordinate systems:

- **Inertial reference system** — This is the earth-fixed coordinate system defined by an orthogonal (right-handed) X, Y, Z coordinate axis system. The Z-axis is parallel to the gravity vector, hence it is pointed downward. The X and Y directions are arbitrary (within the requirement of orthogonality).

- **Vehicle reference system** — This is the vehicle-fixed coordinate system defined by an orthogonal (right-handed) x, y, z coordinate system attached to the sprung mass. Its origin is coincident with the sprung mass center of gravity (CG). The x-axis defines the forward vehicle direction, the y-axis points to the vehicle's right side and the z-axis points towards the bottom of the vehicle.
- **Tire reference system** — This is the tire-fixed coordinate system defined by an orthogonal, right-handed coordinate system fixed to the tire. It has as its origin the perpendicular intersection between the tire plane, the road plane (i.e., terrain) and a plane through the wheel center. The x′-axis lies in the road plane and points in the tire-fixed forward direction, the y′-axis points to the right and the z′-axis is a radial vector in the tire plane normal to the road plane.

These coordinate systems follow SAE Recommended Practice J-670e [5].

The equations that govern the general, non-linear, unsteady motion of the vehicle sprung mass are based on Euler's equations of motion written with respect to the (moving) vehicle reference system. The choice of the moving reference frame greatly simplifies the calculation of external forces and avoids the appearance of the moments and products of inertia in the equations of motion. The derivation for these equations may be found in reference 6. Much of the following development may also be found in reference 9 of this manual.

The equations of motion take the general form:

$$\begin{aligned}\sum F_x &= m(\dot{u} + wq - vr)\\ \sum F_y &= m(\dot{v} + ur - wp)\\ \sum F_z &= m(\dot{w} + vp - uq)\\ \sum M_x &= \dot{L}_x + qL_z - rL_y\\ \sum M_y &= \dot{L}_y + rL_x - pL_z\\ \sum M_z &= \dot{L}_z + pL_y - qL_x\end{aligned}$$

where

Symbol	Definition
u, v, w	vehicle-fixed components of linear velocity
p, q, r	vehicle-fixed components of angular velocity
F_x, F_y, F_z	vehicle-fixed components of external forces
M_x, M_y, M_z	vehicle-fixed components of external moments
L_x, L_y, L_z	vehicle-fixed components of angular momentum

Assuming symmetry in the xy and yz planes, the xy and yz products of inertia are zero. However, symmetry is not assumed in the xz plane, so I_{xz} is included in the angular equations, thus

$$\begin{aligned}M_x &= I_x\dot{p} - I_{xz}\dot{r} - qr(I_y - I_z) - qpI_{xz}\\ M_y &= I_y\dot{q} - (r^2 - p^2)I_{xz} - rp(I_z - I_x)\\ M_z &= I_z\dot{r} - I_{xz}\dot{p} - pq(I_x - I_y) + qrI_{xz}\end{aligned}$$

Sprung Mass

In the derivation that follows, the Euler equations of motion include the inertial coupling effects of the unsprung masses. When combined with the other applied external forces acting on the sprung mass,

$$\sum \vec{F}_{Total} = \sum \vec{F}_{Suspension} + \sum \vec{F}_{Connection} + \sum \vec{F}_{Collision} + \sum \vec{F}_{Aerodynamic} + \sum \vec{F}_{Pulse} + \sum \vec{F}_{Unsprung\,Mass} \qquad \text{(Eq. 3)}$$

where

Term	Definition
$\sum \vec{F}_{Suspension}$	Vehicle-fixed components of suspension forces. This term also carries the tire shear forces reaching the sprung mass through the suspension, the tire sidewall contact forces produced by the Sidewall Impact Tire-Terrain Model, and the collision forces acting on any wheel carrying its own collision mesh
$\sum \vec{F}_{Connection}$	Vehicle-fixed components of connection forces
$\sum \vec{F}_{Collision}$	Vehicle-fixed components of collision forces
$\sum \vec{F}_{Aerodynamic}$	Vehicle-fixed components of aerodynamic forces
$\sum \vec{F}_{Pulse}$	Vehicle-fixed components of forces from a user-entered collision pulse table (see *Collision Pulse Table*, below)
$\sum \vec{F}_{Unsprung\,Mass}$	Vehicle-fixed components of inertial forces from unsprung masses (described in the next section)

(updated: earlier editions omitted the collision pulse table forces, and did not state what the suspension term contains.)

In the vehicle-fixed x-direction, each unsprung mass is assumed to act as a point mass. Thus, Newton's 2nd law yields

$$m_{Sprung}(\dot{u} - vr + wq) + (m_{Sprung} + \sum m_{Unsprung})g\sin\theta = \sum F_{x_{Total}} \qquad \text{(Eq. 4)}$$

where

Symbol	Definition
m_{Sprung}	Sprung mass
$\sum m_{Unsprung}$	Total unsprung mass (axles + wheels)
g	Gravitational constant
θ	Vehicle-fixed pitch angle

In the vehicle-fixed y-direction, Newton's 2nd law yields

$$m_{Sprung}(\dot{v} + ur - wp) - (m_{Sprung} + \sum m_{Unsprung})g\cos\theta\sin\phi = \sum F_{y_{Total}} \qquad \text{(Eq. 5)}$$

where ϕ = vehicle-fixed roll angle.

In the vehicle-fixed z-direction, Newton's 2nd law yields

$$m_{Sprung}(\dot{w} + vp - uq) - m_{Sprung}\, g\cos\theta\cos\phi = \sum F_{z_{Total}} \qquad \text{(Eq. 6)}$$

Rotational motion is handled in a similar manner. The external forces acting on the sprung mass are

$$\sum \vec{M}_{Total} = \sum \vec{M}_{Suspension} + \sum \vec{M}_{Connection} + \sum \vec{M}_{Collision} + \sum \vec{M}_{Aerodynamic} + \sum \vec{M}_{Pulse} + \sum \vec{M}_{Unsprung\,Mass} \qquad \text{(Eq. 7)}$$

where the external moments from suspensions, connections, and so forth, are defined in the vehicle-fixed axis system, and each force term of Eq. 3 contributes a moment about the sprung mass CG through its point of application. As in Eq. 3, the suspension moment term also carries the tire, tire sidewall and wheel collision contributions.

The following equation defines roll motion about the vehicle-fixed x-axis:

$$I_x\dot{p} - I_{xz}\dot{r} - qr(I_y - I_z) - qpI_{xz} + (\sum mz)_{Unsprung}\, g\cos\theta\sin\phi = \sum M_{x_{Total}} \qquad \text{(Eq. 8)}$$

For pitch rotation about the vehicle-fixed y-axis, the equation is

$$I_y\dot{q} - (r^2 - p^2)I_{xz} - rp(I_z - I_x) + (\sum mz)_{Unsprung}\, g\sin\theta = \sum M_{y_{Total}} \qquad \text{(Eq. 9)}$$

For motion about the vehicle-fixed z-axis, the equation is

$$(I_z + \sum I_{z_{Unsprung}})\dot{r} - I_{xz}\dot{p} - pq(I_x - I_y) + qrI_{xz} + (\sum mx)_{Unsprung}\, g\cos\theta\sin\phi - (\sum my)_{Unsprung}\, g\sin\theta = \sum M_{z_{Total}} \qquad \text{(Eq. 10)}$$

where x, y = vehicle-fixed components for location of unsprung mass CGs (refer to the next section for more information regarding the exact definitions for x, y coordinate locations).

Unsprung Mass

Calculation of the motion of the sprung mass requires the contributions of the inertial coupling of the unsprung masses in the vehicle-fixed x- and y-directions; thus, the calculation of the vehicle-fixed components of acceleration of the unsprung masses is necessary. Assuming the unsprung masses are point masses moving with respect to the (moving) vehicle, the vehicle-fixed accelerations for an unsprung mass are

$$a_x = \dot{u} - vr + wq + \ddot{x} + 2q\dot{z} - 2r\dot{y} - x(q^2 + r^2) + y(pq - \dot{r}) + z(pr + \dot{q}) \qquad \text{(Eq. 11)}$$

$$a_y = \dot{v} + ur - wp + \ddot{y} + 2r\dot{x} - 2p\dot{z} + x(pq + \dot{r}) - y(p^2 + r^2) + z(qr - \dot{p}) \qquad \text{(Eq. 12)}$$

where x, y, z = vehicle-fixed components for location of unsprung mass CGs.

> **NOTE:** The acceleration in the z-direction is not required because the unsprung masses are not directly coupled in the z-direction.

The constraints placed on the motion of the unsprung masses by the connections between the sprung and unsprung masses are such that there is no relative motion between the unsprung and sprung masses in the vehicle-fixed x-direction. Thus, $\ddot{x} = 0$. Therefore, in Newton's 2nd law form, the inertial coupling forces for unsprung mass i may be rewritten as

$$F_{x_i} = m_i a_{x_i} = m_i(\dot{u} - vr + wq + 2\dot{z}_i + 2r\dot{y}_i - x_i(q^2 + r^2) + y_i(pq + \dot{r}) + z_i(pr + \dot{q})) \qquad \text{(Eq. 13)}$$

and

$$F_{y_i} = m_i a_{y_i} = m_i(\dot{v} + ur - wp - 2p\dot{z}_i + x_i(pq + \dot{r}) - y_i(p^2 + r^2) + z_i(qr - \dot{p})) \qquad \text{(Eq. 14)}$$

The unsprung mass positions and their derivatives appear in the above inertial coupling equations. Thus, the vehicle-fixed positions, velocities and accelerations for the unsprung masses are required. These kinematics are different for independent and solid axle suspension types.

Independent Suspensions

An independent suspension includes two degrees of freedom that are inertially coupled to the sprung mass: the vertical motion (vehicle-fixed z-direction) for each wheel (see Figure 4-2). Wheel lateral velocity is considered negligible, thus, the $2r\dot{y}$ term vanishes from Eq. 13. The wheel x-coordinate relative to the vehicle center of gravity is equal to the user-assigned wheel coordinate, possibly modified by occupants and payloads and event-related wheel x-displacement. Thus,

$$x_{Wheel} = x_{Initial} + \Delta x_{Payload} + \Delta x_{Occupants} + \Delta x_{Displ} \qquad \text{(Eq. 15)}$$

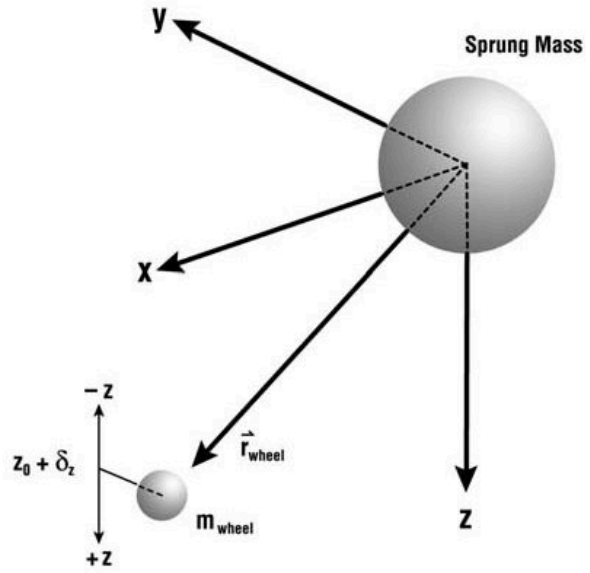


Figure 4-2: Degrees of freedom for an independent suspension. Dynamic motion of each wheel is defined in the vehicle-fixed z-direction (spin and steer angular dynamic motion may also be defined as an option; see Wheel Spin and Steering System sections).

The wheel y-coordinate is defined by the user-assigned wheel y-coordinate, possibly modified by occupants and payloads and event-related wheel displacement. In addition, the wheel lateral location changes for an independent suspension due to the effects of the half-track change during wheel jounce and rebound. Thus,

$$y_{Wheel} = y_{Initial} + \Delta y_{Payload} + \Delta y_{Occupants} + \Delta y_{Displ} + \frac{dy_w}{dz} \qquad \text{(Eq. 16)}$$

where $\frac{dy_w}{dz}$ = half-track change interpolated from the user-entered Half-track vs. Jounce/Rebound table.

The wheel z-coordinate is initially defined by the user-assigned z-coordinate, again, possibly modified by payloads, occupants and event-related displacement. However, the z-wheel position is a degree of freedom, thus the current wheel position is governed by its equation of motion in the z-direction. Thus,

$$z_{Wheel} = z_{Initial} + \Delta z_{Payload} + \Delta z_{Occupants} + \Delta z_{Displ} + \delta z_{Wheel} \qquad \text{(Eq. 17)}$$

where δz_{Wheel} = current wheel z-displacement from initial (equilibrium) position.

Newton's 2nd law applied to the equation for vertical wheel motion results in

$$\sum F_{z_{Unsprung}} = F_{z_{Tire}} - F_{Suspension} + m_{Unsprung}(uq - vp + g\cos\theta\cos\phi - x_{Wheel}\,pr - y_{Wheel}\,qr + z_{Wheel}(p^2 + r^2)) \qquad \text{(Eq. 18)}$$

Solid Axle Suspensions

A solid axle suspension also includes two degrees of freedom inertially coupled to the sprung mass: axle vertical motion (vehicle-fixed z-direction) and axle roll about the roll center (see Figure 4-3).

Unlike independent suspensions, for solid axle suspensions the equations of motion describe the motion of the solid axle roll center, not the individual wheels.

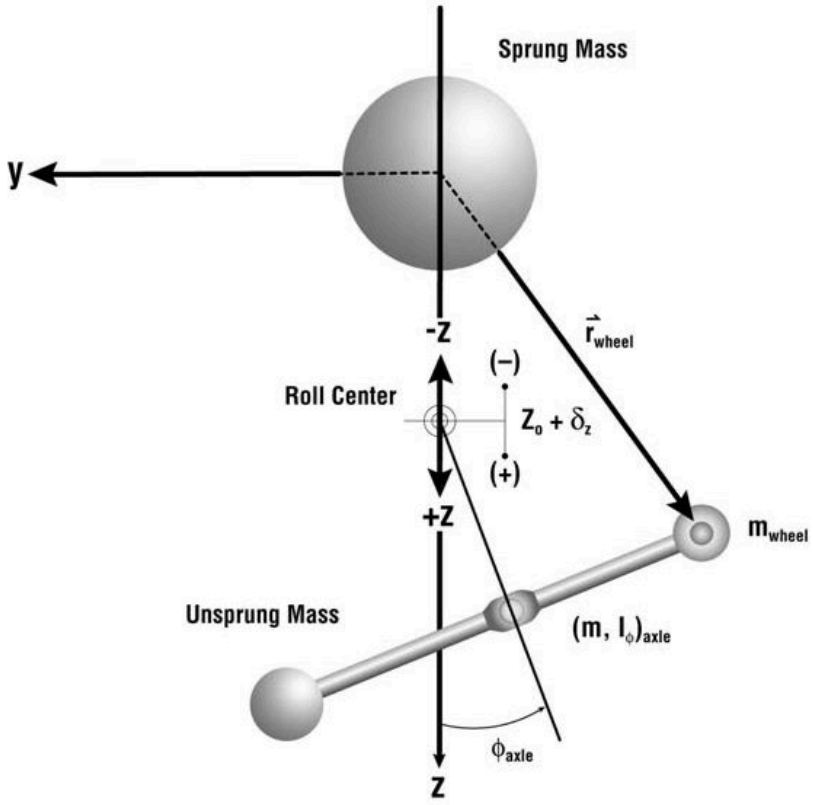


Figure 4-3: Degrees of freedom for a solid axle suspension.

The vehicle-fixed x-coordinate of the unsprung mass is equal to the average of the current wheel x-coordinates (see Eq. 15).

The unsprung mass y- and z-coordinates have the following kinematic relationships (refer to Figure 4-3):

$$y = -\rho\sin\phi \qquad \text{(Eq. 19)}$$

$$\dot{y} = -\rho\dot{\phi}\cos\phi \qquad \text{(Eq. 20)}$$

$$\ddot{y} = \rho(\dot{\phi}^2\sin\phi - \ddot{\phi}\cos\phi) \qquad \text{(Eq. 21)}$$

$$z = z_{Static} + \delta_{Axle} + \rho\cos\phi \qquad \text{(Eq. 22)}$$

$$\dot{z} = \dot{\delta}_{Axle} - \rho\dot{\phi}\cos\phi \qquad \text{(Eq. 23)}$$

$$\ddot{z} = \ddot{\delta}_{Axle} - \rho(\dot{\phi}^2\cos\phi + \ddot{\phi}\sin\phi) \qquad \text{(Eq. 24)}$$

where

Symbol	Definition
ρ	Fixed distance from roll center to sprung mass (downward +)
ϕ	Vehicle-fixed axle roll angle
z_{Static}	Initial distance from the vehicle CG to the roll center
δ_{Axle}	Current vertical deflection of roll center from initial (equilibrium) position

Newton's 2nd law applied to axle vertical motion is

$$\sum F_{z_{Unsprung}} = F_{z_{Tire}} - F_{z_{Suspension}} + m_{Unsprung}(uq - vp + g\cos\theta\cos\phi + \rho\dot{\phi}^2 + 2p\rho\dot{\phi} - x_{Axle}\,pr + \rho\phi qr + (\rho + z_{Axle})(p^2 + q^2)) \qquad \text{(Eq. 25)}$$

For solid axle roll-axis rotation, application of Newton's 2nd law yields

$$\sum M_{\phi_{Unsprung}} = M_{\phi_{Tire}} - M_{\phi_{Suspension}}$$
$$+m_{Unsprung}\,\rho(ur - wp - 2p\dot{z} + 2p\rho\dot{\phi}\dot{\phi} + x_{Axle}\,pq + \rho\dot{\phi}(p^2 + r^2)$$
$$+(\rho + z_{Axle})qr - g\cos\theta\sin(\phi_{Sprung} + \phi_{Axle}))$$
$$+m_{Unsprung}\,\rho\dot{\phi}(vp - uq - 2p\rho\dot{\phi} + x_{Axle}\,pr - (\rho + z_{Axle})(p^2 + q^2))$$
$$-I_{Axle,\phi}\,\dot{\phi}(r^2 - q^2) - I_{Axle,\phi}\,qr$$

Wheel Spin

Calculation of wheel spin velocity, Ω, requires the wheel spin acceleration, $\dot{\Omega}$. Spin acceleration arises from drive and brake torques and from tire rolling resistance, as shown in the free-body diagram in Figure 4-4 (for purposes of clarification, only torque-producing forces are shown in the figure).

For driven axles, the model is complicated by the introduction of the coupling effect of the differential and from the introduction of drivetrain rotational inertia. The differential equation for the spin degree of freedom at each wheel includes these effects:

$$(I_{Wheel,Rt} + \frac{I_{Drivetrain}\,\eta_{Diff}^2}{4})\frac{d}{dt}\Omega_{Wheel,Rt} + (\frac{I_{Drivetrain}\,\eta_{Diff}^2}{4})\frac{d}{dt}\Omega_{Wheel,Lt} = [-F_{x'}\,r + M_{Rolling} + T_b + (\zeta T_d)]_{Right\ Tires} \qquad \text{(Eq. 27)}$$

$$(I_{Wheel,Lt} + \frac{I_{Drivetrain}\,\eta_{Diff}^2}{4})\frac{d}{dt}\Omega_{Wheel,Lt} + (\frac{I_{Drivetrain}\,\eta_{Diff}^2}{4})\frac{d}{dt}\Omega_{Wheel,Rt} = [-F_{x'}\,r + M_{Rolling} + T_b + (\zeta T_d)]_{Left\ Tires} \qquad \text{(Eq. 28)}$$

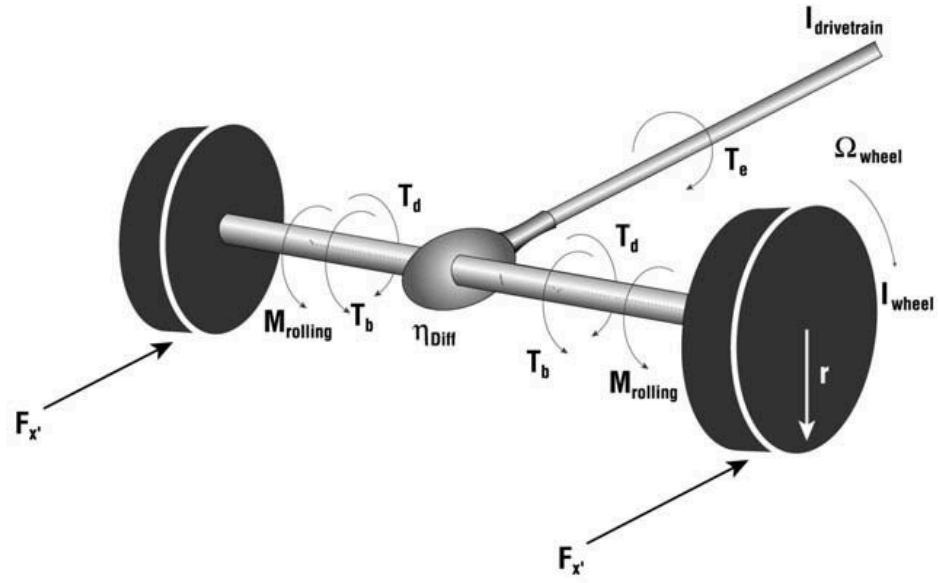


Figure 4-4: Wheel spin degree of freedom moments and forces.

Collecting the applied moments at each wheel as

$$M = T_d + T_b + M_{Collision} - F_{x'}\,r + M_{Rolling} \qquad \text{(Eq. 28a)}$$

Eqs. 27 and 28 form a pair of simultaneous equations in the two spin accelerations, whose exact solution is

$$\dot{\Omega}_{Wheel,Rt} = \frac{M_{Rt}\,I_{Wheel,Lt} + (M_{Rt} - M_{Lt})A}{I_{Wheel,Rt}\,I_{Wheel,Lt} + A(I_{Wheel,Rt} + I_{Wheel,Lt})} \qquad \text{(Eq. 29)}$$

and

$$\dot{\Omega}_{Wheel,Lt} = \frac{M_{Lt}\,I_{Wheel,Rt} + (M_{Lt} - M_{Rt})A}{I_{Wheel,Rt}\,I_{Wheel,Lt} + A(I_{Wheel,Rt} + I_{Wheel,Lt})} \qquad \text{(Eq. 30)}$$

where

$$A = \frac{I_{Drivetrain}\,\eta_{Diff}^2}{4}$$

For a non-driven axle, $I_{Drivetrain}$ is zero, A vanishes, and Eqs. 29 and 30 reduce to $\dot{\Omega} = M / I_{Wheel}$ at each wheel independently.

(updated: earlier editions solved Eqs. 27 and 28 using a pair of "wheel inertial factors", ξ. That method was replaced by the direct solution given above, and the factors are no longer used. The moment term also now includes the collision moment, $M_{Collision}$, which is the wheel spin moment produced by the 3-D mesh collision model when a wheel is being contacted; it is zero otherwise.)

Once $\dot{\Omega}$ is determined for each wheel, the wheel spin velocity is integrated directly:

$$\Omega = \Omega_{Prev} + \int \dot{\Omega}\, dt \qquad \text{(Eq. 31)}$$

In the preceding development,

Symbol	Definition
Ω	Wheel spin velocity
Ω_{Prev}	Wheel spin velocity during previous integration timestep
T_d	Wheel drive torque $= T_e \times \eta_{Trans} \times \eta_{Diff} \times \zeta$ (Eq. 32)
T_e	Engine torque (see Eq. 51)
η_{Trans}	Transmission ratio
η_{Diff}	Differential ratio
ζ	Torque split (fraction of engine torque applied to specified wheel)
$F_{x'}$	Tire circumferential force
r	Tire effective rolling radius
$M_{Rolling}$	Tire rolling resistance moment
I_{Wheel}	Total wheel spin inertia: tire + rim (×2 if dual tires) + axle + any spinning portion of brake
$I_{Drivetrain}$	Total rotational inertia of drivetrain components: engine + transmission + driveline
$M_{Collision}$	Wheel spin moment from the 3-D mesh collision model (zero unless the wheel is being contacted)
T_b	Applied brake torque at wheel (see *Brake Torque*)

The spin velocity of Eq. 31 is advanced with a first-order update at each tire force timestep rather than by the vehicle's main integrator.

> **NOTE:** In the calculation of wheel drive torque, T_d (above), ζ is the "torque function" that determines how the drive torque is distributed between the drive wheels. By default, SIMON splits the drive torque equally between all drive wheels; if the vehicle is equipped with traction control the split is redistributed between the wheels. See *Drive Torque — Torque split* (Eq. 51g).

Steering System

The steering system includes the steering gear ratio (steer angle at the steering wheel divided by the steer angle at the axle). This ratio is used when the At Steering Wheel steer table option is selected.

SIMON also incorporates a Steer Degree of Freedom model. The Steer Degree of Freedom model is activated by selecting the appropriate option in SIMON's Calculation Options dialog.

(updated: the current Calculation Options dialog offers four Steer DOF settings — Off, Normal, Append and AutoStart; see Chapter 2.)

The engineering model used by the Steer Degree of Freedom option is shown in Figure 4-5. The linkage is assumed to be rigid, thus the angular acceleration about the steering axis is the same for right-side and left-side wheels. External steer forces are generated at the tire-road interface, thus producing moments about the steering axis. The moments are resisted by steer system inertia and internal coulomb friction. Steering is limited by right and left steering stops at each wheel.

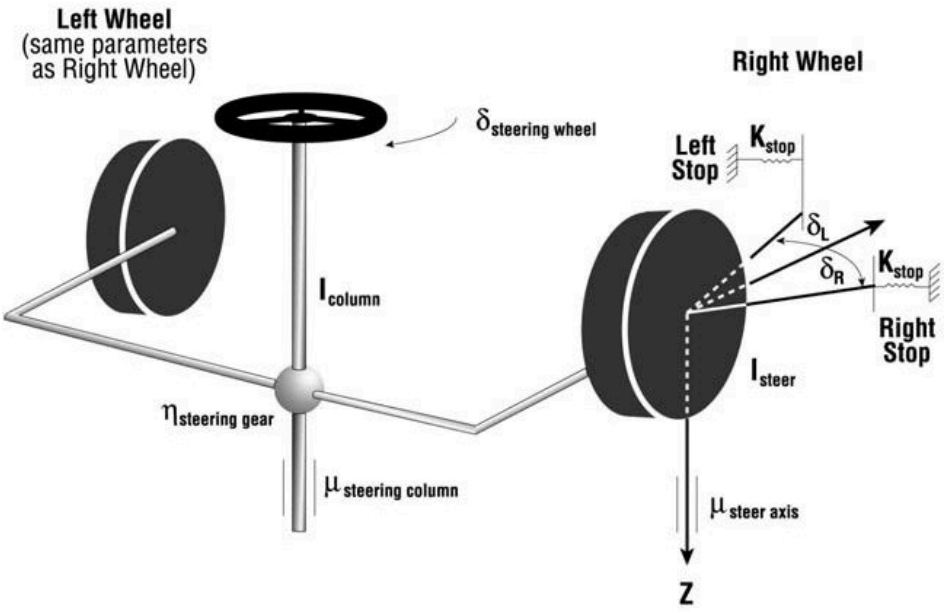


Figure 4-5: Steering system model used for Steer Degree of Freedom option.

Application of Newton's 2nd law to the steering system, ignoring inertial coupling effects, results in

$$\sum M_{Steering} = I_{Steering}\,\ddot{\delta}_{Steer} \qquad \text{(Eq. 33)}$$

where

Symbol	Definition
$\sum M_{Steering}$	Sum of external moments acting on steering system components
$I_{Steering}$	Total rotational inertia of steering system components
δ_{Steer}	Steer angle of each steerable wheel about its steering axis (thus, $\ddot{\delta}_{Steer}$ is the angular acceleration)

The sum of external moments, taken over both wheels of the axle, is

$$\sum M_{Steering} = \sum_{Sides} (M_{Stops} + M_{Steer\ Axis\ Friction} + M_{Precession} + M_{Tires} + M_{Sidewall} + M_{Collision}) + M_{Steering\ Column\ Friction} \qquad \text{(Eq. 34)}$$

where

Term	Definition
M_{Stops}	Moments about wheel steer axis produced by contact with steering stops
$M_{Steer\ Axis\ Friction}$	Moments about wheel steer axis produced by coulomb friction in the steering ball joints or king pin
$M_{Precession}$	Gyroscopic precession moment produced by the spinning wheel
M_{Tires}	Moments about the wheel steer axis produced by the tire forces acting through the trail and scrub radius at the tire-ground shear interface (contact patch)
$M_{Sidewall}$	Moment about the wheel steer axis produced by tire sidewall contact, when the Sidewall Impact Tire-Terrain Model is in use
$M_{Collision}$	Moment about the wheel steer axis produced by the 3-D mesh collision model when the wheel is being contacted
$M_{Steering\ Column\ Friction}$	Moment produced by coulomb friction between the steering shaft and its bushings or bearings, applied once for the axle

(updated: earlier editions listed only the stop, steer axis friction, steering column friction and tire terms. The gyroscopic precession, sidewall and collision moments are also part of the sum.)

Steer angle from the driver

While the Steer Degree of Freedom is not controlling the wheels, the steer angle at each wheel comes from the driver's steer table. For the At Axle option, each wheel takes its own table entry directly. For the At Steering Wheel option, the table entry is a steering wheel angle, and the wheel angle is

$$\delta = \frac{\delta_{SW}}{\eta} \qquad \text{(Eq. 33a)}$$

where η is the axle's steering gear ratio. Both wheels of the axle receive the same angle unless **Use Ackermann Steering** is selected in Driver Controls, in which case the front axle's wheel angles are spread so that the two wheels point at a common turn center:

$$\delta_i = \arctan\left(\frac{\tan\delta}{1 - y_i \frac{\tan\delta}{L}}\right) \qquad \text{(Eq. 33b)}$$

where y_i is the vehicle-fixed lateral coordinate of the wheel and L is the wheelbase. For a three-axle vehicle the wheelbase is measured from the front axle to the midpoint of the rear tandem. If the denominator becomes too small to evaluate — which happens only at steer angles well beyond any physical steering stop — the wheel angles are left unchanged for that timestep.

Ackermann steering applies to the front axle only, and only with the At Steering Wheel option.

(updated: Ackermann steering was not described in earlier editions.)

When the HVE Driver Model is following a path, it supplies the steer angle and steer velocity directly, overriding the steer table. Following a collision the driver model releases control and the steer table resumes.

Steer Degree of Freedom modes

The Steer DOF setting determines when the wheels stop following the driver's steer table and begin responding to the tire forces:

Setting	Behavior
Off	The wheels follow the steer table for the whole event.
Normal	The steer table is not used at all; the wheels respond to tire forces from the start of the event.
Append	The wheels follow the steer table until the time of its last entry, then respond to tire forces for the remainder of the event.
AutoStart	The wheels follow the steer table until the vehicle is involved in a collision, then respond to tire forces.

When AutoStart triggers, each steerable wheel's steer angle is set to the angle it had at that instant, its steer velocity is set to zero, and the numerical integration is reinitialized so the new state is carried forward smoothly. Both vehicles in the collision start together.

While the Steer Degree of Freedom is controlling the wheels, the steering wheel angle and velocity reported in the output are the average of the two wheel values multiplied by the steering gear ratio.

(updated: this subsection describes behavior not documented in earlier editions.)

Steering Stop Torque

Steer angles are limited by steering stops at the right- and left-side wheels. Each wheel's steering stops limit the steer angle for both right and left steering inputs; the right-steer and left-steer stop angles need not be equal.

> **NOTE:** For example, it is possible during a left turn for the stop at the left-side wheel to engage before the stop for the right-side wheel.

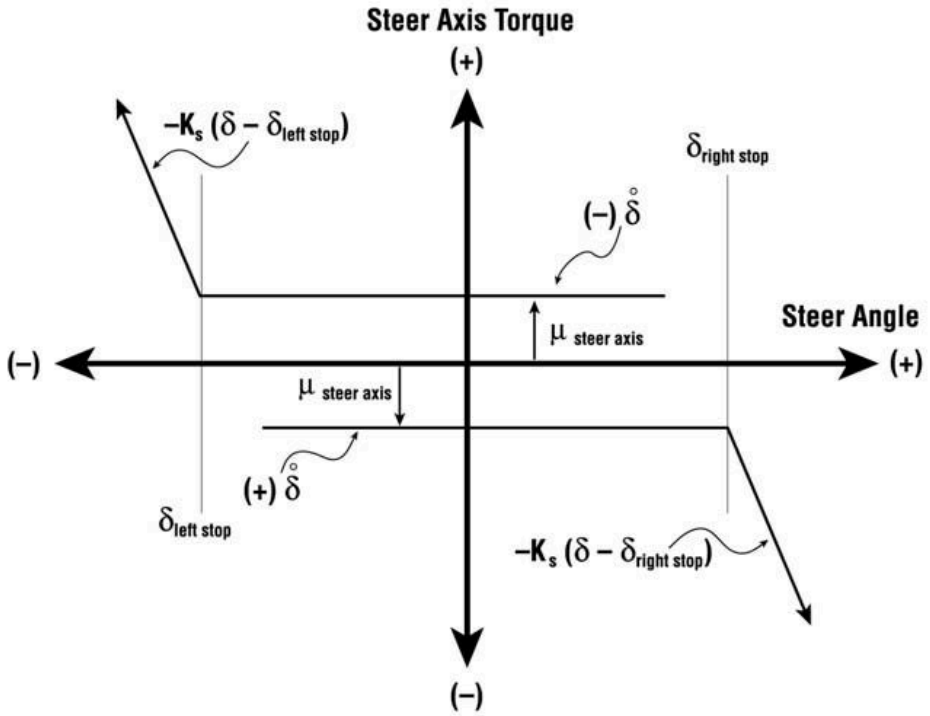


Figure 4-6: Steer axis friction and stop torque vs. steer angle.

A stop resists further travel only while the wheel is still turning deeper into it. The steering stop torque at each stop is therefore

$$M_{Stop} = \begin{cases} -K_{Stop}(\delta_{Steer} - \delta_{Stop}), & |\delta_{Steer}| > |\delta_{Stop}| \text{ and } \operatorname{sgn}\dot{\delta}_{Steer} = \operatorname{sgn}\delta_{Steer} \\ [6pt]0, & \text{otherwise} \end{cases} \qquad \text{(Eq. 35)}$$

where K_{Stop} is the mechanical stiffness and δ_{Stop} the travel limit of the stop being engaged. Each wheel has a stop for each steer direction, so four stop angles and four stiffnesses apply to a steerable axle; the stop selected is the one for the wheel and the direction of the current steer angle. The general characteristic for steer axis torque is shown in Figure 4-6.

(updated: earlier editions gave the stop torque as a positive multiple of the overtravel. The torque opposes the overtravel, and it is released as soon as the wheel begins to turn back out of the stop, regardless of how far into the stop it still is.)

Steer Axis Torque

Torque is also produced by steering rotation of the wheel about its steer axis. However, no torque is produced unless the steer velocity is non-zero. Thus, a minimum value of steer velocity is required to develop the assigned frictional torque. This minimum steer velocity is called a friction null band. The friction opposes the steer velocity:

$$M_{Steer\ Axis} = \begin{cases} -\mu_{Steer\ Axis}\dfrac{\dot{\delta}_{Steer}}{\varepsilon_{Wheel}}, & |\dot{\delta}_{Steer}| < \varepsilon_{Wheel} \\ [10pt] -\mu_{Steer\ Axis}\operatorname{sgn}\dot{\delta}_{Steer}, & |\dot{\delta}_{Steer}| \geq \varepsilon_{Wheel} \end{cases} \qquad \text{(Eq. 36)}$$

where

Symbol	Definition
$\mu_{Steer\ Axis}$	Steer axis friction torque for each wheel
ε_{Wheel}	Steering friction null band, referred to the wheel (Eq. 36a)

The friction null band is entered as a steering column velocity, while the equations of motion work at the wheel, so it is converted before use:

$$\varepsilon_{Wheel} = \frac{\varepsilon}{\eta} \qquad \text{(Eq. 36a)}$$

where ε is the null band as entered and η is the steering gear ratio. The right and left wheels may have different steer axis friction values, but they share the same steer velocity and therefore the same null band ratio.

(updated: earlier editions gave the steer axis friction with the sign of the steer velocity rather than opposing it, and did not describe the conversion of the null band from the steering column to the wheel.)

Steering Column Torque

The steering column and steering gear introduce additional friction torque. Like steer axis torque, described above, there is no steering column frictional torque unless the steer velocity is non-zero. Because the column turns faster than the wheels by the steering gear ratio, the column friction torque is multiplied by that ratio when it is referred to the wheel steer axis:

$$M_{Steering\ Column} = \begin{cases} -\mu_{Steering\ Column}\ \eta\ \dfrac{\dot{\delta}_{Steer}}{\varepsilon_{Wheel}}, & |\dot{\delta}_{Steer}| < \varepsilon_{Wheel} \\ [10pt] -\mu_{Steering\ Column}\ \eta\ \operatorname{sgn}\dot{\delta}_{Steer}, & |\dot{\delta}_{Steer}| \geq \varepsilon_{Wheel} \end{cases} \quad \text{(Eq. 37)}$$

where

Symbol	Definition
$\mu_{Steering\ Column}$	Steering column frictional torque
η	Steering gear ratio (column angle : wheel angle)

Unlike the steer axis friction, which acts at each wheel, the steering column friction is applied once for the axle.

(updated: earlier editions omitted the steering gear ratio that refers the column friction torque to the wheel steer axis, and gave the null band comparison in inconsistent units.)

Gyroscopic Precession Torque

A spinning wheel resists being tilted. When the vehicle rolls, or when the wheel's inclination angle changes as the suspension deflects, the wheel's spin momentum produces a moment about the steer axis:

$$M_{Precession} = I_{Spin}\ \Omega(p + \dot{\gamma})\cos\delta_{Steer} \quad \text{(Eq. 37a)}$$

where

Symbol	Definition
I_{Spin}	Wheel spin inertia
Ω	Wheel spin velocity
p	Vehicle roll velocity
$\dot{\gamma}$	Rate of change of the wheel's vehicle-fixed inclination angle

The $\cos\delta_{Steer}$ term projects the moment onto the steer axis. This torque is what produces the steering disturbance felt when a wheel is loaded or unloaded during a rapid maneuver.

(updated: the gyroscopic precession torque was not documented in earlier editions.)

Tire-Ground Torque

Forces at the tire-ground shear interface are the external input to the steering system. Because these forces do not act through the steer axis at its intersection with the ground plane (see Figure 4-7), an external moment is produced.

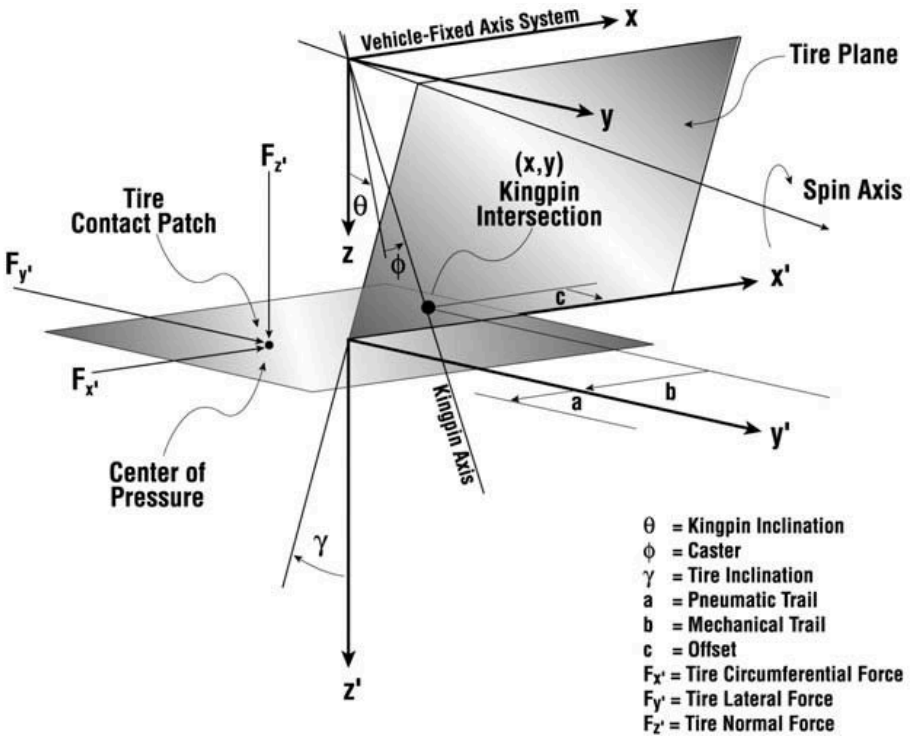


Figure 4-7: Close-up view of torque-producing mechanism at tire-ground shear interface.

The tire force acts at the contact patch, which is offset from the steer axis both fore-and-aft (by the trail) and laterally (by the scrub radius). The external moment produced at each tire is

$$M_{Tire} = F_y \ell_y - F_x \ell_x + F_z \ell_z \qquad \text{(Eq. 38)}$$

with moment arms

$$\ell_x = r_y + T\sin\delta_G + R_S\cos\delta_G \qquad \text{(Eq. 38a)}$$

$$\ell_y = r_x + T\cos\delta_G + R_S\sin\delta_G\cos\theta_x \qquad \text{(Eq. 38b)}$$

$$\ell_z = (r_x + T\cos\delta_G + R_S\sin\delta_G)\sin\gamma \qquad \text{(Eq. 38c)}$$

The total trail, T, is the pneumatic trail less the mechanical trail, and the mechanical trail follows from the caster angle:

$$T = T_P - T_M, \qquad T_M = r_z\sin\phi_C \qquad \text{(Eq. 38d)}$$

The scrub radius follows from the stub axle length and the kingpin inclination:

$$R_S = L_{Stub}\cos\gamma - r_z\sin(\phi_K + \gamma) \qquad \text{(Eq. 38e)}$$

where

Symbol	Definition
F_x, F_y, F_z	Vehicle-fixed tire force components
r_x, r_y, r_z	Vehicle-fixed components of distance from wheel center to tire-ground contact point
T_P	Tire pneumatic trail
T_M	Mechanical trail
R_S	Scrub radius
L_{Stub}	Stub axle length
ϕ_C	Steer axis caster angle
ϕ_K	Steer axis kingpin inclination
γ	Inclination angle (angle from tire z′ axis to ground surface normal)
δ_G	Wheel vehicle-fixed steer angle relative to ground plane (see Eq. 84)
θ_x	Angle from vehicle x-axis to ground plane (see Eq. 72)

The kingpin inclination and stub axle length are taken as positive on the right-side wheel and negative on the left, so that the scrub radius acts outboard on both sides.

(updated: earlier editions stated that the mechanical trail is assumed to be zero and gave no scrub radius term. Both are calculated from the vehicle's steer axis geometry, the mechanical trail subtracts from the pneumatic trail rather than adding to it, and the longitudinal force term is subtracted rather than added.)

> **NOTE:** Dual tires often have different tire force components, tire radii and tire-ground contact characteristics. Therefore, each dual tire must be handled separately.

When the Sidewall Impact Tire-Terrain Model is in use, the moment produced about the steer axis by sidewall contact — a tire striking the vertical face of a curb, for example — is added to Eq. 38 for each tire.

Steering System Rotational Inertia

The rotational inertia of the entire steering system is

$$I_{Steering} = I_{Steer,Rt} + I_{Steer,Lt} + I_{Column}\,\eta^2 \qquad \text{(Eq. 39)}$$

where

Symbol	Definition
$I_{Steer,Rt}$	Total steer rotational inertia for right-side wheel: tire + rim (×2 if dual tires) + any steering portion of brake
$I_{Steer,Lt}$	Total rotational inertia for left-side wheel
I_{Column}	Total rotational inertia of steering column, including steering gearbox
η	Steering gear ratio

The column inertia is referred to the wheel steer axis through the square of the steering gear ratio, as it is for any inertia reflected through a gear reduction. A high gear ratio therefore makes the steering system substantially harder to accelerate, which is the dominant contribution for most vehicles.

(updated: earlier editions multiplied the column inertia by the steering gear ratio rather than its square.)

Payloads

SIMON supports one payload on each unit vehicle. Note that this means the tow vehicle, trailer(s) and dolly(s) may each have an individual payload. The effect of the payload is simply to change the inertial properties and CG location of the vehicle's sprung mass.

Human occupants are handled by the same procedure; see

Human Occupants

, below. The payload adjustment is applied first, then each occupant in turn.

Given a payload with inertial properties m, I_x, I_y, I_z, placing the payload at a distance, $\vec{r}$, from the sprung mass center of gravity changes the sprung mass inertial properties as follows:

$$m_{Total} = m_{Sprung} + m_{Payload} \qquad \text{(Eq. 40)}$$

$$\eta = \frac{m_{Payload}}{m_{Total}} \qquad \text{(Eq. 41)}$$

$$\Delta_x = \eta r_x, \qquad \Delta_y = \eta r_y, \qquad \Delta_z = \eta r_z \qquad \text{(Eq. 42)}$$

where $\vec{\Delta}$ = payload adjustment.

Adding a payload to a vehicle does not cause the vehicle to be repositioned on the road (in the real world or in HVE!). Rather, adding a payload simply relocates the center of gravity within the vehicle. This effectively changes the x, y, z coordinates of all vehicle-fixed components (wheels, inter-vehicle connections, etc). For each component, the coordinates are changed

$$\vec{C}_{Adjusted} = \vec{C}_{Original} - \vec{\Delta} \qquad \text{(Eq. 43)}$$

where $\vec{C}_{Adjusted}$ = adjusted x, y, z coordinates for component C (e.g., the adjusted vehicle-fixed coordinates of the right front wheel).

The adjustment in center of gravity location and the additional rotational inertias of the payload affect rotational inertias of the sprung mass. This effect can be calculated using the parallel axis theorem.

Let

$$dV_x = \sqrt{\Delta_y^2 + \Delta_z^2}, \qquad dV_y = \sqrt{\Delta_x^2 + \Delta_z^2}, \qquad dV_z = \sqrt{\Delta_x^2 + \Delta_y^2} \qquad \text{(Eq. 44)}$$

and

$$dP_x = \sqrt{r_y^2 + r_z^2} - dV_x, \qquad dP_y = \sqrt{r_x^2 + r_z^2} - dV_y, \qquad dP_z = \sqrt{r_x^2 + r_y^2} - dV_z \qquad \text{(Eq. 45)}$$

(updated: earlier editions gave these moment arms as simple sums of the coordinate offsets. They are the perpendicular distances from each axis, so the two offsets combine as a root sum of squares. The simple sum overstates the inertia change substantially for any payload offset in more than one direction.)

The adjusted rotational inertias for the vehicle sprung mass are

$$I_{Adjusted,x} = I_{Original,x} + I_{Payload,x} + m_{Sprung}\, dV_x^2 + m_{Payload}\, dP_x^2$$
$$I_{Adjusted,y} = I_{Original,y} + I_{Payload,y} + m_{Sprung}\, dV_y^2 + m_{Payload}\, dP_y^2$$
$$I_{Adjusted,z} = I_{Original,z} + I_{Payload,z} + m_{Sprung}\, dV_z^2 + m_{Payload}\, dP_z^2$$

where

Symbol	Definition
$d\vec{V}$	Adjustment in vehicle CG location in vehicle-fixed roll, pitch and yaw planes
$d\vec{P}$	Payload distance from adjusted CG in vehicle-fixed roll, pitch and yaw planes

Human Occupants

Human occupants may be added to vehicles. Mathematically, the result is equivalent to adding multiple payloads, using the procedures described above. The effect is to change the inertial properties and weight distribution of the vehicle. Humans do not move relative to the vehicle; their motion is not simulated.

Driver Controls

SIMON provides user-entered driver controls for steering, braking, throttle and gear selection. These driver controls are entered in the form of open-loop tables of driver input vs. time. SIMON also provides a closed-loop option, called the HVE Driver Model.

Steering

Steering inputs may be provided for right and left side wheels at each steerable axle. At Steering Wheel and At Axle options are supported. If the At Steering Wheel option is selected, the right side and left side wheel steer angles are equal to the current table entry divided by the axle's steering gear ratio, unless Use Ackermann Steering is selected, in which case the two front wheels are given different angles so that they point at a common turn center. See

Steering System — Steer angle from the driver

(Eqs. 33a and 33b).

HVE Driver Model

SIMON supports the HVE Driver Model. The HVE Driver Model is a closed-loop driver control model that uses driver control attributes and the SIMON vehicle dynamics model to attempt to follow a user-specified path. This model is described in detail in HVE reference [4].

Braking

Braking inputs may be provided for each wheel. The At Pedal option is the most robust model and its use is recommended. Using the At Pedal option causes SIMON to use the brake torque ratio supplied for each individual wheel. Wheel spin inertias and coupling within the differential are also included when the At Pedal option is selected.

Braking Force and Percent Available Friction brake table options are also allowed. If either of these methods is used, spin inertias and differential coupling are irrelevant and are ignored.

HVE Brake Designer

If the At Pedal braking option is selected, SIMON uses the HVE Brake Designer. Detailed brake component designs can be created and edited in the Vehicle Editor. The resulting brake design is then used during SIMON simulations. The brake torque is calculated dynamically for each wheel according to the mechanical design of the brake assembly as well as the current brake pressure, lining/drum temperature and sliding speed. The current lining friction is determined from the lining temperature and sliding speed; thus, brake fade can be simulated.

The HVE Brake Designer is described in detail in the HVE Designer Manual. Additional information may be found in reference [1].

Throttle

Throttle inputs may be provided for each drive axle. The Percent Wide-open Throttle (% WOT) option is the most robust and its use is recommended. Using the % WOT option causes SIMON to use the entire drivetrain model (see next section). Wheel spin inertias and coupling within the differential are also included when the % WOT option is selected.

Tractive Effort and Percent Available Friction options are also allowed. If either of these methods is used, spin inertias and differential coupling are irrelevant and are ignored.

Gear Selection

If the % WOT throttle option (above) is selected, the Gear Selection table is used for setting the time and gear selections for each gear shift.

> **NOTE:** If no gear selection entries are made, the vehicle's transmission will be in neutral and no amount of throttle will cause the vehicle to accelerate!

Wheel Torque

Wheel torque arises from driver throttle and brake inputs from open-loop driver tables (see Driver Controls, above).

Brake Torque

SIMON uses three optional methods for computing the current level of attempted brake torque. These are:

- At Pedal
- Braking Force
- Percent Available Friction

These methods are described in the following section.

Whichever method is used, SIMON distinguishes between the **attempted** brake torque — what the brake system is trying to apply — and the **applied** brake torque, which is what actually reaches the wheel spin equation after the low-speed limits of Eqs. 47e through 47h are applied.

At Pedal Braking Option

For the At Pedal option, the attempted brake torque at each wheel is produced by a sequence of steps applied at every timestep. In order, these are:

1. Determine the brake system pressure at the source, from the driver's pedal input or from the HVE Driver Model (Eq. 47a).
2. Apply the brake time lag for the wheel (Eq. 47a).
3. Apply proportioning, if a proportioning valve is present (Eq. 48).
4. Apply brake failure, if the wheel's brake is failing (Eq. 47b).
5. Apply the brake rise time (Eq. 47c).
6. Apply yaw stability control modulation, if the vehicle is so equipped.
7. Apply ABS modulation, if the vehicle is so equipped.
8. Update the brake temperatures and calculate the attempted brake torque from the resulting wheel pressure (Eq. 47).

A wheel damaged during a collision bypasses this sequence entirely; see

Wheel damage lockup

, below.

The attempted brake torque, T_b, at each wheel is calculated as follows:

$$T_b = T_{Ratio} \max(0,\ p - p_0) \qquad \text{(Eq. 47)}$$

where

Symbol	Definition
T_b	Attempted brake torque at wheel
T_{Ratio}	Brake Torque Ratio, the attempted brake torque produced per unit of actuation pressure
p	Current application pressure at wheel cylinder or air chamber (after proportioning)
p_0	Pushout Pressure (psi)

Torque ratio for generic brake types is entered directly by the user for each wheel. For brakes created using the HVE Brake Designer, the torque ratio is calculated from the brake's mechanical design and material parameters and current operational characteristics (lining temperature and sliding speed).

Note that the brake produces no torque until the wheel pressure exceeds the pushout pressure, and none at all when the wheel pressure is zero.

(updated: earlier editions gave Eq. 47 without the limit at zero. A wheel pressure below the pushout pressure produces no brake torque; it does not produce a negative torque.)

If the HVE ABS Model is invoked, wheel brake pressures are modulated according to the selected ABS algorithm (see HVE User's Manual, Chapter 31).

Source pressure and brake time lag

The brake system pressure at the source is the product of the current pedal force and the vehicle brake pedal ratio. Each wheel has its own brake time lag, which delays the arrival of that pressure at the wheel:

$$p_{Table} = \begin{cases} 0, & t < t_{Lag} \\ [4\text{pt}] F_{Table}(t - t_{Lag}) \times R, & t \geq t_{Lag} \end{cases} \qquad \text{(Eq. 47a)}$$

where t_{Lag} is the wheel's brake time lag, F_{Table} is the At Pedal brake table interpolated at the lagged time, and R is the vehicle brake pedal ratio. Note that the table is evaluated at the lagged time, so the entire pedal force history is shifted later by the lag rather than merely being suppressed during it.

When the HVE Driver Model's speed follower is controlling the vehicle, the pedal force is supplied by the driver model instead of the brake table, and the brake time lag is not applied. A trailer whose tow vehicle is under speed follower control uses the tow vehicle's pedal force and pedal ratio. Following a collision, speed follower control is released and the brake table resumes, with the time lag applied.

(updated: the brake time lag and the speed follower interaction were not documented in earlier editions.)

Brake Pressure Proportioning

The presence of a proportioning valve will reduce the pressure at the wheel as follows:

$$p = \begin{cases} p_{Table}, & \text{for } p_{Table} \le p_{Proportion} \\ [4pt] p_{Proportion} + \eta(p_{Table} - p_{Proportion}), & \text{for } p_{Table} > p_{Proportion} \end{cases} \qquad \text{(Eq. 48)}$$

where

Symbol	Definition
p	Brake system pressure effective at wheel
p_{Table}	Brake system pressure effective at source (master cylinder or storage reservoir), Eq. 47a
$p_{Proportion}$	System pressure at which proportioning begins
η	Proportioning ratio (wheel pressure : source pressure)

Brake failure

A wheel brake may be set to fail during the event. The extent of the failure ramps linearly from none at the failure start time to the user-specified extent at the end of the failure duration, and the wheel pressure is reduced by that fraction:

$$p \leftarrow p(1 - \varepsilon(t)), \qquad \varepsilon(t) = \begin{cases} 0, & t \le t_{Fail} \\ [4pt] \varepsilon_{Fail} \dfrac{t - t_{Fail}}{\Delta t_{Fail}}, & t_{Fail} < t < t_{Fail} + \Delta t_{Fail} \\ [8pt] \varepsilon_{Fail}, & t \ge t_{Fail} + \Delta t_{Fail} \end{cases} \qquad \text{(Eq. 47b)}$$

where ε_{Fail} is the fraction of total brake failure, t_{Fail} the failure start time and Δt_{Fail} the failure duration. A failure extent of 1.0 removes the wheel's braking entirely.

(updated: the brake failure model was not documented in earlier editions.)

Brake rise time

The wheel pressure does not reach its commanded value instantaneously. If a brake rise time is specified for the wheel, the pressure approaches the commanded value exponentially:

$$p \leftarrow p_{Prev}\, e^{-\Delta t / \tau} + p(1 - e^{-\Delta t / \tau}) \qquad \text{(Eq. 47c)}$$

where p_{Prev} is the wheel pressure at the previous brake calculation, τ is the brake rise time and Δt is the interval since the previous brake calculation. A rise time of zero applies the commanded pressure directly.

(updated: the brake rise time was not documented in earlier editions.)

Brake temperature and fade

Before the torque of Eq. 47 is calculated, SIMON updates the brake temperatures at each wheel from the work done by the brake during the previous step, the wheel's rotational speed, the vehicle's total velocity and the ambient air temperature. Temperatures are tracked at five locations through the lining and drum cross-section. The lining temperature and the sliding speed then determine the current lining friction, and therefore the brake torque ratio of Eq. 47 — this is how brake fade is produced. Fade is available only for brakes created with the HVE Brake Designer, since a generic brake's torque ratio is a fixed user entry.

(updated: earlier editions noted that fade can be simulated but did not describe where the temperature calculation occurs in the sequence.)

Wheel damage lockup

If a wheel is damaged during the event and the Wheel Damage option specifies a lockup torque for it, the entire brake system calculation above is bypassed and the attempted brake torque is set directly from the damage:

$$T_b = \lambda(t)\, T_{Nominal}, \qquad T_{Nominal} = \mu_{p,max}\, r_{Tire,max}\, F_{z,Static} \qquad \text{(Eq. 47d)}$$

where $\lambda(t)$ is the wheel's damage lockup fraction, interpolated over the wheel damage start time and duration in the same way as the other wheel damage parameters, and $T_{Nominal}$ is a nominal wheel lockup torque formed from the largest peak friction value and the largest tire radius among the vehicle's tires, multiplied by the wheel's static vertical load. A lockup fraction of 1.0 therefore corresponds to approximately the torque needed to lock the wheel under its static load.

The bypass takes effect only when the lockup fraction is meaningfully greater than zero; below that, the normal brake system calculation applies.

(updated: the wheel damage lockup torque was not documented in earlier editions.)

Brake Force and Percent Available Friction Options

If the Brake Force or Percent Available Friction brake table options are used, the attempted brake torque, T_b, is calculated directly:

$$T_b = \begin{cases} F_{Wheel} \times \max(r_{Tire,Inner}, r_{Tire,Outer}), & \text{for the Brake Force method} \\ [6pt]\theta_{Wheel} \times \sum_{i=0}^{N} \mu_p F_R \, r_{Tire}, & \text{for the \% Available Friction method} \end{cases} \quad \text{(Eq. 49)}$$

where

Symbol	Definition
F_{Wheel}	Attempted brake force from Brake Force table
r_{Tire}	Current tire radius
θ_{Wheel}	Attempted brake input from % Available Friction table
N	Number of tires at wheel location
μ_P	Current peak coefficient of friction for the tire — the value interpolated from the friction table for the tire's current load and speed, scaled by the terrain friction multiplier and by the hydroplaning model if one is selected
F_R	Current radial tire force (normal to the terrain)

> **NOTE:** If the Brake Force method is used and the wheel location uses dual tires, T_b is based on the largest tire radius. The % Available Friction method sums the contribution of each tire at the wheel location.
>
> **NOTE:** Use of the Wheel Brake Force and Percent Available Friction options is discouraged because they bypass SIMON's brake system model and ignore the effects of the wheel spin degree of freedom. In addition, ABS simulation is not possible.

(updated: the % Available Friction method sums over the tires at a dual-tire wheel location; earlier versions used a single tire radius.)

Applied brake torque at low wheel speed

The attempted brake torque calculated above cannot always be applied. A brake can command more torque than the tire is able to react against the road, and near zero wheel speed a torque larger than the tire can support would drive the wheel spin solution unstable — it would spin the wheel backwards. SIMON therefore limits the attempted torque to the torque actually available at the tire before applying it to the wheel spin equation.

The torque available at a wheel is the moment its tire forces produce about the spin axis:

$$T_{Avail} = \sum_{i=0}^{N} F_{x_i} \, r_{Tire_i} \quad \text{(Eq. 47e)}$$

The limit is applied per axle, because the two wheels of a driven axle are coupled through the differential. Let Ω_{min} be the minimum wheel spin velocity, 0.1 rad/sec. Then:

- **Both wheels turning** ($|\Omega| > \Omega_{min}$ on both sides). The full attempted torque is applied at each wheel, directed opposite to that wheel's spin:

$$T_{b,Applied} = -|T_b|\text{sgn}(\Omega) \quad \text{(Eq. 47f)}$$

- **Both wheels at or below the threshold, and the available torque does not exceed the attempted torque on either side.** Both wheels are locked, and each wheel's applied torque is limited to its own available torque:

$$T_{b,Applied} = -|T_{Avail}|\text{sgn}(\Omega) \quad \text{(Eq. 47g)}$$

- **One wheel turning and one at or below the threshold, or one wheel locked and the other not.** The two wheels are treated separately. The turning (or unlocked) wheel receives its attempted torque as in Eq. 47f, except that if its partner is locked and its own available torque acts in the same direction as its spin, its brake is released and no torque is applied. The remaining wheel is limited by the torque available to it after the differential has transferred torque from the other side:

$$T_{Diff} = T_{Avail,2} - \rho(T_{Avail,1} - T_{b,Applied,1}), \qquad \rho = \frac{A}{I_{Wheel,1} + A} \quad \text{(Eq. 47h)}$$

where subscript 1 denotes the turning or unlocked wheel, subscript 2 the other wheel, I_{Wheel} is the wheel spin inertia and A is the drivetrain inertia term of Eq. 29. If T_{Diff} does not act to slow the wheel, the brake is released; if $|T_{Diff}|$ equals or exceeds the attempted brake torque, the attempted torque is applied; otherwise T_{Diff} is applied.

ρ is zero on a non-driven axle, so the two wheels of a non-driven axle are limited independently.

The result of this limiting is that the brake torque reported in the variable output can be smaller than the torque the brake system is attempting to produce whenever a wheel is near or at rest, and can momentarily fall to zero when a wheel is released. This is expected behavior at low speed and does not indicate a problem with the brake data.

(updated: this subsection describes behavior not documented in earlier editions. It applies to the At Pedal, Brake Force and % Available Friction methods alike.)

Drive Torque

SIMON uses three optional methods for computing the current level of attempted drive torque. These are:

- Percent Wide-open Throttle
- Tractive Effort
- Percent Available Friction

These methods are described in the following section. Drive torque is applied only at drive axles; a non-driven axle receives no drive torque by any method.

Percent Wide-Open Throttle Option

If the Percent Wide-open Throttle table option is used, the drivetrain model is used in full: the current engine speed is derived from the drive wheel speeds through the transmission and differential, the engine torque is interpolated from the engine tables at that speed and the current throttle position, and the result is distributed to the drive wheels.

The engine speed implied by the current road speed is

$$\dot{\theta}_{e,\,Road} = \eta_{Differential} \times \eta_{Transmission} \times \frac{\sum_{i=1}^{N} \Omega_{Wheel}}{N} \qquad \text{(Eq. 50)}$$

The engine cannot turn more slowly than its idle speed, so the speed used to look up engine torque is

$$\dot{\theta}_e = \max\left(\dot{\theta}_{e,\,Road},\ \dot{\theta}_{Idle}\right) \qquad \text{(Eq. 50a)}$$

If $\dot{\theta}_{e,\,Road}$ exceeds the highest speed in either engine table, the event terminates with an excessive engine speed message (see Chapter 6).

The engine torque is then

$$T_e = T_{CT} + \lambda(T_{WOT} - T_{CT}) \qquad \text{(Eq. 51)}$$

where

Symbol	Definition
$\eta_{Differential}$	Current numerical ratio of differential from driver controls, differential gear table
$\eta_{Transmission}$	Current numerical ratio of transmission (Eq. 51b or 51c)
$\dot{\theta}_{e,\,Road}$	Engine speed implied by the current drive wheel speeds
$\dot{\theta}_e$	Engine speed used for the engine table look-up
$\dot{\theta}_{Idle}$	Engine idle speed
Ω_{Wheel}	Current wheel spin velocity (see Eq. 31)
N	Number of drive wheels
T_e	Current engine torque (this result is used in Eq. 32)
T_{CT}	Closed throttle engine torque at $\dot{\theta}_e$ from engine torque vs. engine speed table (negative)
T_{WOT}	Wide-open-throttle engine torque at $\dot{\theta}_e$ from engine torque vs. engine speed table
λ	Current throttle position, interpolated from driver controls, throttle table

Below idle speed the engine is not allowed to retard the vehicle: if Eq. 51 returns a negative torque while the road speed implies an engine speed below idle, the engine torque is set to zero.

$$T_e = 0 \quad \text{when} \quad T_e < 0 \text{ and } \dot{\theta}_{e,\,Road} < \dot{\theta}_{Idle} \qquad \text{(Eq. 51a)}$$

(updated: the idle speed floor of Eq. 50a and the engine braking limit of Eq. 51a were not described in earlier editions.)

Gear selection

The differential ratio always comes from the driver's differential gear table. The transmission ratio comes from the driver's gear table for a manual transmission, or from the automatic shift logic for an automatic transmission.

A gear table is a step function of time, not an interpolation: the gear in effect is the one from the most recent table entry at or before the current time. Before the first table entry the transmission is in neutral and the ratio is zero, which is why a vehicle with no gear table entries will not accelerate no matter how much throttle is applied.

$$\eta_{Transmission} = \eta(g), \qquad g = \text{gear number from the most recent table entry} \qquad \text{(Eq. 51b)}$$

Gear numbers are ordered with neutral first, reverse second, and the forward gears after that. The transmission may have up to 12 forward ratios, plus reverse and neutral; the differential may have up to three ratios.

For an **automatic transmission**, the gear is selected from the current engine speed and throttle position rather than from a time table. If a gear table entry selects neutral or reverse, or the transmission is currently in neutral or reverse, that selection is used directly and no speed-dependent shift is made. Otherwise the transmission upshift and downshift points — expressed as throttle positions — are interpolated against engine speed from the vehicle's automatic shift data, and

$$\text{Upshift when} \quad \dot{\theta}_e > \dot{\theta}_{Max} \quad \text{or} \quad \left(\lambda < \lambda_{Up} \text{ and } \dot{\theta}_e \frac{\eta(g+1)}{\eta(g)} > \dot{\theta}_{Min}\right) \qquad \text{(Eq. 51c)}$$

$$\text{Downshift when} \quad \dot{\theta}_e < \dot{\theta}_{Min} \quad \text{or} \quad \left(\lambda > \lambda_{Down} \text{ and } \dot{\theta}_e \frac{\eta(g-1)}{\eta(g)} < \dot{\theta}_{Min} + 0.75(\dot{\theta}_{Max} - \dot{\theta}_{Min})\right) \qquad \text{(Eq. 51d)}$$

where

Symbol	Definition
g	Current gear number
$\eta(g)$	Transmission ratio for gear g
$\dot{\theta}_{Min}, \dot{\theta}_{Max}$	Minimum and maximum engine speeds for shifting
$\lambda_{Up}, \lambda_{Down}$	Upshift and downshift throttle positions, interpolated against engine speed

The second condition in each case tests where the engine would land after the shift, so that a shift is not made if it would leave the engine outside its usable speed range. At most one shift is made per timestep, and the engine speed of Eq. 50 is recomputed with the new ratio. Downshifts are permitted only out of the second forward gear or higher.

(updated: automatic transmission gear selection was not described in earlier editions, which described only the driver's gear table.)

Clutch

If **Use Clutch/Torque Converter** is selected in Driver Controls and the transmission is in reverse or first gear, the engine is allowed to turn faster than the road speed would imply, as it does when a driver slips the clutch to pull away. The target engine speed rises with throttle position from idle to the speed at which the engine develops its peak wide-open-throttle torque:

$$\dot{\theta}_{e,Clutch} = \dot{\theta}_{Idle} + \lambda(\dot{\theta}_{Peak} - \dot{\theta}_{Idle}) \qquad \text{(Eq. 51e)}$$

and the clutch slip is the fraction by which the engine is outrunning the driveline:

$$s_{Clutch} = 1 - \frac{\dot{\theta}_{e,Road}}{\dot{\theta}_{e,Clutch}}, \qquad 0 \le s_{Clutch} \le 1 \qquad \text{(Eq. 51f)}$$

While the clutch is slipping, the engine torque of Eq. 51 is looked up at $\dot{\theta}_{e,Clutch}$ instead of $\dot{\theta}_e$. Once the driveline catches up, the slip falls to zero and the engine speed reverts to Eq. 50a. Clutch slip is reported in the variable output.

The clutch is active only in reverse and first gear; in any higher gear the engine speed follows the road speed directly. The transmission type — manual or automatic — is part of the vehicle's Drivetrain data, while the clutch setting belongs to the event's Driver Controls.

(updated: the clutch model was not described in earlier editions.)

Torque split

The engine torque, multiplied by the transmission and differential ratios, is divided among the drive wheels by the torque split, ζ. By default the split is equal across every drive wheel:

$$\zeta = \frac{0.5}{N_{Drive\ Axles}} \quad \text{at each wheel of a drive axle;} \qquad \zeta = 0 \quad \text{at a non-driven axle} \qquad \text{(Eq. 51g)}$$

If the vehicle is equipped with traction control, the traction control model redistributes the split between the wheels according to the total drive torque and the vehicle's yaw velocity error, and the split is then no longer equal. Traction control activity is reported in the variable output.

(updated: the redistribution of the torque split by traction control was not described in earlier editions.)

Tractive Effort and Percent Available Friction Options

If the Tractive Effort or Percent Available Friction throttle table options are used, the attempted drive torque, T_d, is calculated directly, bypassing the drivetrain model:

$$T_d = \begin{cases} F_{Wheel} \times \max(r_{Tire,Inner}, r_{Tire,Outer}), & \text{for the Tractive Effort method} \\ [6pt]\theta_{Wheel} \times \max_{i=1}^{N}(\mu_p F_R r_{Tire}), & \text{for the \% Available Friction method} \end{cases} \qquad \text{(Eq. 52)}$$

(Definitions are the same as for the Braking Tables; see above.)

> **NOTE:** At a wheel location with dual tires, both drive torque methods use the largest single tire value rather than the sum over the tires. This differs from the % Available Friction brake method, which sums over the tires at the wheel location.

(updated: earlier editions gave the % Available Friction drive torque as a sum over the tires at the wheel location. It is the largest value among them.)

> **NOTE:** Use of the Tractive Effort and Percent Available Friction options is discouraged because they bypass SIMON's drivetrain model and ignore the effects of the wheel spin degree of freedom.

Wheel Position

Calculation of wheel center earth-fixed coordinates requires a transformation from the vehicle-fixed reference frame to the inertial reference frame. This transformation matrix, A, is given by

$$A = \begin{bmatrix} \cos\psi\cos\theta & \sin\psi\cos\theta & -\sin\theta \\ -\sin\psi\cos\phi + \cos\psi\sin\phi\sin\theta & \cos\psi\cos\phi + \sin\phi\sin\psi\sin\theta & \sin\phi\cos\theta \\ \sin\psi\sin\phi + \cos\psi\sin\theta\cos\phi & -\cos\psi\sin\phi + \sin\psi\sin\theta\cos\phi & \cos\theta\cos\phi \end{bmatrix} \quad \text{(Eq. 53)}$$

where

Symbol	Definition
ψ	Vehicle yaw angle
θ	Vehicle pitch angle
ϕ	Vehicle roll angle

> **NOTE:** The order of rotations is yaw, pitch, roll.

Calculation of these Euler angles requires integration of the rotational velocity components, p, q, r (see Eqs. 8 through 10). The integrated values are

$$\dot{\phi} = p + (q\sin\phi + r\cos\phi)\tan\theta \quad \text{(Eq. 54)}$$

$$\dot{\theta} = q\cos\phi - r\sin\phi \quad \text{(Eq. 55)}$$

$$\dot{\psi} = (q\sin\phi + r\cos\phi)\sec\theta \quad \text{(Eq. 56)}$$

Note that the presence of $\tan\theta$ and $\sec\theta$ results in a singularity at 90 degree pitch angle. If necessary, an axis indexing scheme can be implemented to eliminate this singularity.

Given the current vehicle sprung mass orientation in space, the earth-fixed (inertial) wheel center coordinates are

$$\begin{bmatrix} X_w \\ Y_w \\ Z_w \end{bmatrix} = \begin{bmatrix} X_{CG} \\ Y_{CG} \\ Z_{CG} \end{bmatrix} + A \cdot \begin{bmatrix} x_w \\ y_w \\ z_w \end{bmatrix} \quad \text{(Eq. 57)}$$

where

Term	Definition
$[X_w\ Y_w\ Z_w]^T$	Earth-fixed wheel center coordinates
$[X_{CG}\ Y_{CG}\ Z_{CG}]^T$	Earth-fixed vehicle CG coordinates
$[x_w\ y_w\ z_w]^T$	Vehicle-fixed wheel center coordinates

The calculation of vehicle-fixed wheel center coordinates is dependent upon the suspension type.

Solid Axle Suspension

For solid axle suspension types, the vehicle-fixed wheel center coordinates are given by

$$x_{Wheel} = x_{Initial} + \Delta x_{Displ} \quad \text{(Eq. 58)}$$

$$y_{Wheel} = y_{Initial} - \phi_{Axle}\,\rho_{Axle} + \Delta y_{Displ} \quad \text{(Eq. 59)}$$

$$z_{Wheel} = z_{Initial} + \delta_{Axle} + \phi_{Axle}\,y_{Initial} + \Delta z_{Displ} \quad \text{(Eq. 60)}$$

where

Symbol	Definition
$[x, y, z]_{Initial}$	Vehicle-fixed wheel x, y, z coordinates at the start of the simulation (may include displacements due to occupants and payload)
$[\Delta x, \Delta y, \Delta z]_{Displ}$	Wheel x, y, z displacements from user-entered wheel displacement table
δ_{Axle}	Current roll center displacement from equilibrium
ϕ_{Axle}	Current vehicle-fixed axle roll angle

Independent Suspension

For independent suspension types, the vehicle-fixed wheel center coordinates are given by

$$x_{Wheel} = x_{Initial} + \Delta x_{Displ} \qquad \text{(Same as Eq. 58)}$$

$$y_{Wheel} = y_{Initial} + \Delta y_{Displ} + \Delta y_{Track} \qquad \text{(Eq. 61)}$$

$$z_{Wheel} = z_{Initial} + \Delta z_{Displ} + \delta_{Wheel} \qquad \text{(Eq. 62)}$$

where

Symbol	Definition
Δy_{Track}	Half-track change at the current suspension deflection, interpolated from the vehicle's Half-track Change vs. Jounce/Rebound table and applied outboard on both sides
δ_{Wheel}	Current wheel z-displacement from equilibrium position

(updated: in the current version, when DyMESH wheel contact is enabled, wheel x-y displacement can also result dynamically from collision forces on the wheel mesh — see the Sprung Mass Impact Model section below. Earlier editions wrote the half-track term in Eq. 61 as a rate of change; it is the interpolated half-track change itself. The axle roll term in Eq. 59 moves the wheel toward the inside of the roll.)

Steer Axis Geometry

At a steerable axle — of either suspension type — the wheel does not turn about a vertical axis through the wheel center. It turns about the steer axis, which is inclined by the caster and kingpin angles and offset from the wheel center by the stub axle length. SIMON accounts for this by rotating the wheel about its actual steer axis, which changes three things at once as the wheel steers:

- the **wheel center position** moves, because the wheel center is offset from the steer axis;
- the **camber angle** changes, which is the camber gain a steered wheel develops from kingpin inclination and caster;
- the **steer angle at the road** differs slightly from the rotation applied about the steer axis.

The wheel center coordinates of Eqs. 58 through 62, the camber angle and the steer angle are all replaced by the values that result from this rotation before they are used elsewhere. The effect grows with steer angle and with the size of the caster and kingpin angles, and vanishes for an axle whose steer axis is vertical and passes through the wheel center.

(updated: the steer axis geometry treatment was not described in earlier editions, which placed the wheel center from the suspension deflection alone.)

Vehicle-fixed Wheel Orientation

Wheel orientation with respect to the vehicle-fixed axis system is defined by three angles (see Figure 4-8):

- **Wheel Steer,** δ_M — Wheel rotation about an axis parallel to the vehicle-fixed z (yaw) axis
- **Wheel Camber,** γ_M — Wheel rotation about an axis parallel to the vehicle-fixed x (roll) axis
- **Wheel Spin,** ω_M — Wheel rotation about an axis parallel to the vehicle-fixed y (pitch) axis

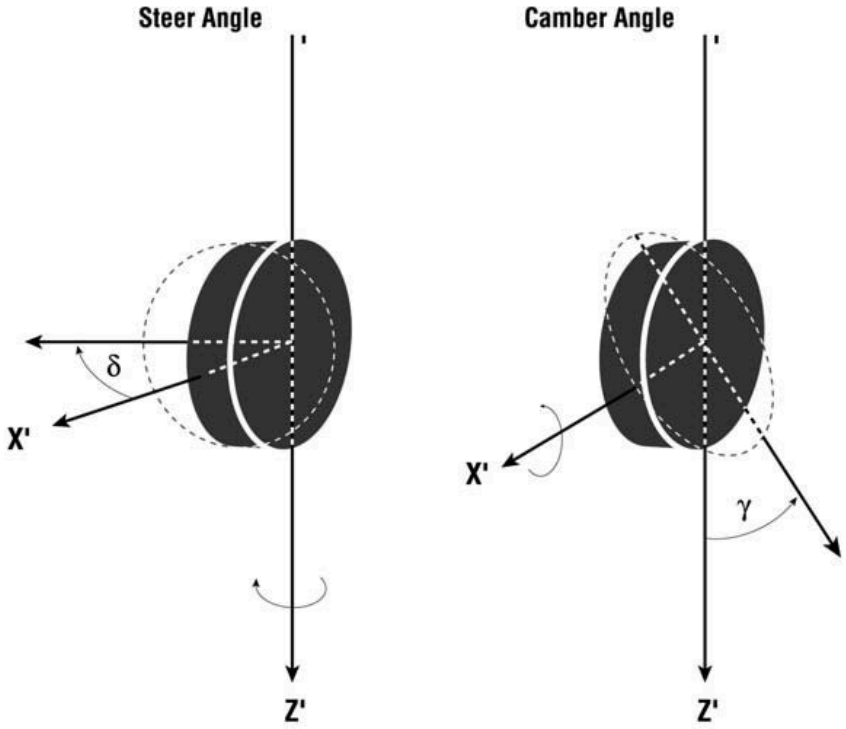


Figure 4-8: Vehicle-fixed wheel orientation.

Wheel spin is calculated by the equations of motion for each wheel (see Eqs. 27 through 32). If the steer degree of freedom option is selected, wheel steer is calculated by the equations of motion for the steering system (see Eqs. 33 through 39). Otherwise, wheel steer and wheel camber are determined from user-defined suspension properties, look-up tables and driver control tables. The manner in which the wheel orientations are calculated is dependent upon suspension type.

Independent Suspension

Wheel camber angle, γ_w, for independent suspensions is determined using the vehicle suspension camber change table, by linear interpolation of the tabular inputs of camber vs. wheel jounce/rebound.

Wheel steer angle, δ_w, for an independent suspension is determined first from the open-loop driver input table, using linear interpolation of the tabular inputs of wheel steer angle vs. time. If the At Axle steering option is used, the right-side and left-side steer angles may differ; if the At Steering Wheel option is used, the right-side and left-side steer angles are the same, and are equal to the table value divided by the steering gear ratio.

In addition to the current steering input from the driver steer table, steering may also occur as a result of rotation of the sprung mass about its roll axis (e.g., while cornering). The associated vertical wheel travel (jounce for the outside wheel; rebound for the inside wheel) can induce additional steering because of the kinematics of the suspension and steering arms. Thus, for an independent suspension, the total steer angle at a wheel is

$$\delta_w = \delta_T + \kappa_0 + \kappa_1 \, dz_w + \kappa_2 \, dz_w^2 + \kappa_3 \, dz_w^3 \qquad \text{(Eq. 63)}$$

where

Symbol	Definition
δ_T	Steer angle interpolated from driver control table
dz	Current wheel displacement from equilibrium
$\kappa_0, \kappa_1, \kappa_2, \kappa_3$	Constant, first-order, second-order and third-order coefficients of roll steer vs. wheel jounce/rebound defined for each wheel

Note that the right-side and left-side wheels may have different roll steer values. This is, in fact, what happens when a single wheel drops into a pothole.

> **NOTE:** Roll steer is applied only when the **At Steering Wheel** steering option is used. With the At Axle option, each wheel takes its steer angle from the driver table alone and no roll steer contribution is added. *(updated: this restriction was not stated in earlier editions.)*
>
> **NOTE:** This is called "Bump Steer".

Solid Axle Suspension

Wheel camber angle, γ_w, for solid axle suspensions is determined by the equation of motion: because the camber angle is fixed with respect to the axle, the camber angle is equal to the vehicle-fixed axle roll angle, plus any axle camber set. Thus,

$$\gamma_w = \phi_{Axle} + d\gamma_w \qquad \text{(Eq. 64)}$$

where

Symbol	Definition
ϕ_{Axle}	Vehicle-fixed axle roll angle (see Eq. 26)
$d\gamma_w$	User-entered axle camber set, applied outboard on both sides so that a positive camber set leans the tops of both wheels the same way relative to the axle

Wheel steer angle, δ_w, for a solid axle suspension is determined in the same manner as for independent suspension, described earlier, except that roll steer is not a function of individual wheel jounce/rebound; rather it is a function of the axle roll, ϕ_{Axle}, with respect to the sprung mass. Thus,

$$\delta_w = \delta_T + \kappa_\phi \, \phi_{Axle} \qquad \text{(Eq. 65)}$$

where κ_ϕ = solid axle roll steer coefficient. As for an independent suspension, this term is applied only when the At Steering Wheel steering option is used.

Tire Contact Patch Velocity

The calculation of tire forces requires definition of the tire slip angle, which, in turn, requires calculation of velocity components at the tire contact patch. These velocity components are first calculated in the vehicle-fixed reference frame and then projected onto the ground plane beneath the tire. The calculation of these velocity components is a function of the vehicle's current CG velocity, vehicle-fixed wheel position and suspension type.

Solid Axle

Contact patch velocity for a solid axle suspension is the result of the superimposed components of vehicle CG velocity and individual wheel center velocities. In addition, the contact patch is at a distance, $\dot{h}_w$, from the wheel center; this distance affects the individual contact patch velocity components. For a solid axle suspension, the resulting contact patch velocities are

$$u_{Tire} = u - r(y_w + h_y) + q(z_w + h_z) \qquad \text{(Eq. 66)}$$

$$v_{Tire} = v - p(z_w + h_z) + r(x_w + h_x) \qquad \text{(Eq. 67)}$$

$$w_{Tire} = w - q(x_w + h_x) + (p + \dot{\phi}_{Axle})(y_w + h_y) + \dot{z}_{Axle} \qquad \text{(Eq. 68)}$$

where

Symbol	Definition
u, v, w	Vehicle-fixed forward, lateral and vertical velocity components
p, q, r	Roll, pitch and yaw angular velocities about the vehicle-fixed axes
x_w, y_w, z_w	Current vehicle-fixed wheel coordinates
$\dot{z}_{Axle}$	Current axle vertical velocity
$\dot{\phi}_{Axle}$	Current axle roll velocity
h_x, h_y, h_z	Current vehicle-fixed components of distance from wheel center to tire contact patch

Independent Suspension

Contact patch velocity for an independent suspension is calculated using the same approach as that used for a solid axle suspension, except that the wheel positions and velocities are handled differently (e.g., see Eqs. 15 through 24). For an independent suspension, the contact patch velocities are

$$u_{Tire} = u - r(y_w + h_y) + q(z_w + h_z) \qquad \text{(Same as Eq. 66)}$$

$$v_{Tire} = v - pz_w - h_z(p + \frac{d\gamma}{dz}\dot{z}_w) + r(x_w + h_x) + \frac{dy_w}{dz}\dot{z}_w \qquad \text{(Eq. 69)}$$

$$w_{Tire} = w - q(x_w + h_x) + py_w + (p + \frac{d\gamma}{dz}\dot{z}_w)h_y + \dot{z}_w \qquad \text{(Eq. 70)}$$

where

Symbol	Definition
$\frac{d\gamma}{dz}$	Wheel camber change with respect to jounce/rebound interpolated from camber change table
$\dot{z}_w$	Current vertical wheel velocity
$\frac{dy_w}{dz}$	Wheel half-track change with respect to jounce/rebound interpolated from half-track table

Velocity in the Ground Plane

Finally, the velocity in the ground plane is determined. The forward and lateral components of this velocity are required for the tire slip angle calculation.

First define

$$\begin{bmatrix} \cos\alpha_x \\ \cos\beta_x \\ \cos\gamma_x \end{bmatrix} = A \cdot \begin{bmatrix} 1 \\ 0 \\ 0 \end{bmatrix} = \text{the direction cosines for the vehicle-fixed x-axis}$$

and

$$\begin{bmatrix} \cos\alpha_y \\ \cos\beta_y \\ \cos\gamma_y \end{bmatrix} = A \cdot \begin{bmatrix} 0 \\ 1 \\ 0 \end{bmatrix} = \text{the direction cosines for the vehicle-fixed y-axis}$$

The desired velocity is the projection onto the ground plane of the previously calculated contact patch velocity. The components of the projection of the vehicle-fixed x-axis onto the ground plane are

$$\vec{A}_X = \begin{bmatrix} a_{x_i} \\ a_{x_j} \\ a_{x_k} \end{bmatrix} = \begin{bmatrix} u_{Terrain_Z}\cos\beta_y - u_{Terrain_Y}\cos\gamma_y \\ u_{Terrain_X}\cos\gamma_y - u_{Terrain_Z}\cos\alpha_y \\ u_{Terrain_Y}\cos\alpha_y - u_{Terrain_X}\cos\beta_y \end{bmatrix} \qquad \text{(Eq. 71)}$$

The angle between the vehicle-fixed x-axis and its projection onto the ground plane is their dot product

$$\theta_X = \arccos\left(\frac{\cos\alpha_x\, a_{x_i} + \cos\beta_x\, a_{x_j} + \cos\gamma_x\, a_{x_k}}{\sqrt{a_{x_i}^2 + a_{x_j}^2 + a_{x_k}^2}}\right) \qquad \text{(Eq. 72)}$$

and the sign of θ_x is given by

$$\text{sgn}(\theta_x) = \text{sgn}\left(\cos\gamma_x - \frac{a_{x_k}}{\sqrt{a_{x_i}^2 + a_{x_j}^2 + a_{x_k}^2}}\right) \qquad \text{(Eq. 73)}$$

The forward velocity component of the tire contact point in the ground plane is

$$u_{GP} = u_{Tire}\cos\theta_x - w_{Tire}\sin\theta_x \qquad \text{(Eq. 74)}$$

The same procedure is used to calculate the components of the projection of the vehicle-fixed y-axis onto the ground plane,

$$\vec{A}_y = \begin{bmatrix} a_{y_i} \\ a_{y_j} \\ a_{y_k} \end{bmatrix} = \begin{bmatrix} u_{Terrain_Y}\cos\gamma_x - u_{Terrain_Z}\cos\beta_x \\ u_{Terrain_Z}\cos\alpha_x - u_{Terrain_X}\cos\gamma_x \\ u_{Terrain_X}\cos\beta_x - u_{Terrain_Y}\cos\alpha_x \end{bmatrix} \qquad \text{(Eq. 75)}$$

The angle between the vehicle-fixed y-axis and its projection onto the ground plane is their dot product

$$\phi_y = \arccos\left(\frac{\cos\alpha_y\, a_{y_i} + \cos\beta_y\, a_{y_j} + \cos\gamma_y\, a_{y_k}}{\sqrt{a_{y_i}^2 + a_{y_j}^2 + a_{y_k}^2}}\right) \qquad \text{(Eq. 76)}$$

and the sign of ϕ_y is given by

$$\text{sgn}(\phi_y) = \text{sgn}\left(\cos\gamma_y - \frac{a_{y_k}}{\sqrt{a_{y_i}^2 + a_{y_j}^2 + a_{y_k}^2}}\right) \qquad \text{(Eq. 77)}$$

The lateral velocity component of the tire contact point in the ground plane is

$$v_{GP} = v_{Tire}\cos\phi_y - w_{Tire}\sin\phi_y \qquad \text{(Eq. 78)}$$

Tire-Road Contact Patch

Tire force calculations require that the tire-road interface be defined in terms of normal and shear forces, and resulting tire deflection. To perform these calculations requires a careful analysis of the wheel's orientation with respect to an arbitrary surface (terrain), as shown in Figure 4-9.

Terrain Definition

The first step is to define the terrain (contact patch coordinates, friction multiplier and surface normal) beneath the tire. This step is performed using HVE's surface-information search, controlled by the **Get Surface Information Options**. Given the current earth-fixed coordinates, X_t, Y_t, of the tire's center of rotation, the search examines the environment polygon database until it finds the polygon beneath the tire (see Figure 4-10). For this polygon, the search returns the earth-fixed elevation, Z_t, beneath the tire center, as well as the friction multiplier, f, and unit vector, $\vec{U}_{Terrain}$, representing the surface normal for the polygon directly beneath each tire.

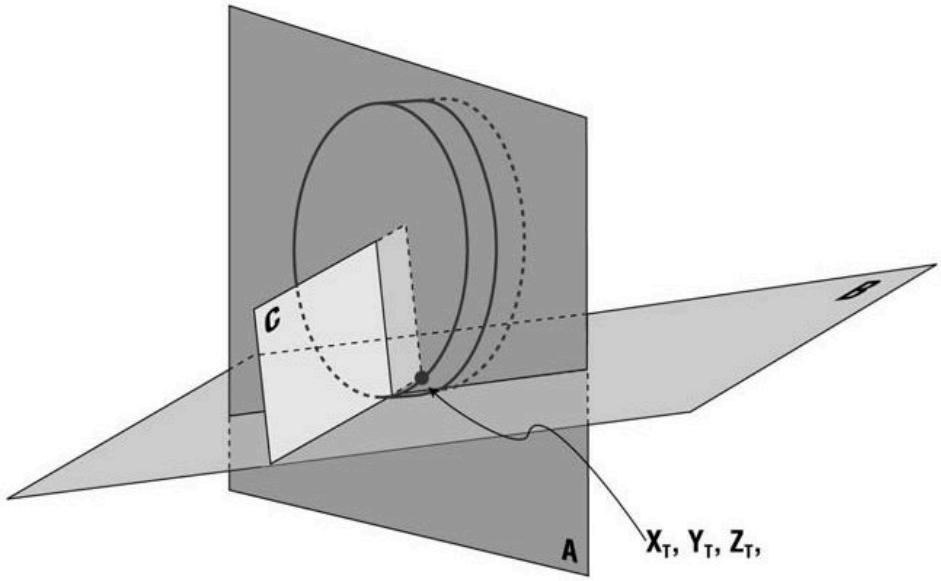


Figure 4-9: Wheel-terrain geometry.

Tire-Ground Orientation

The tire-ground orientation is defined by the tire inclination angle, γ_w, and steer angle, δ_w, relative to the surface beneath the wheel, as shown in Figure 4-9. A line in the earth-fixed coordinate system normal to the wheel plane is defined by a unit vector, $\vec{U}_w$:

$$\vec{U}_w = \begin{bmatrix} u_{w_X} \\ u_{w_Y} \\ u_{w_Z} \end{bmatrix} = A \cdot \begin{bmatrix} -\sin\delta_w \\ \cos\gamma_w \cos\delta_w \\ \sin\gamma_w \cos\delta_w \end{bmatrix} \qquad \text{(Eq. 79)}$$

where

Symbol	Definition
A	Transformation matrix (see Eq. 53)
γ_w	Vehicle-fixed camber angle
δ_w	Vehicle-fixed steer angle

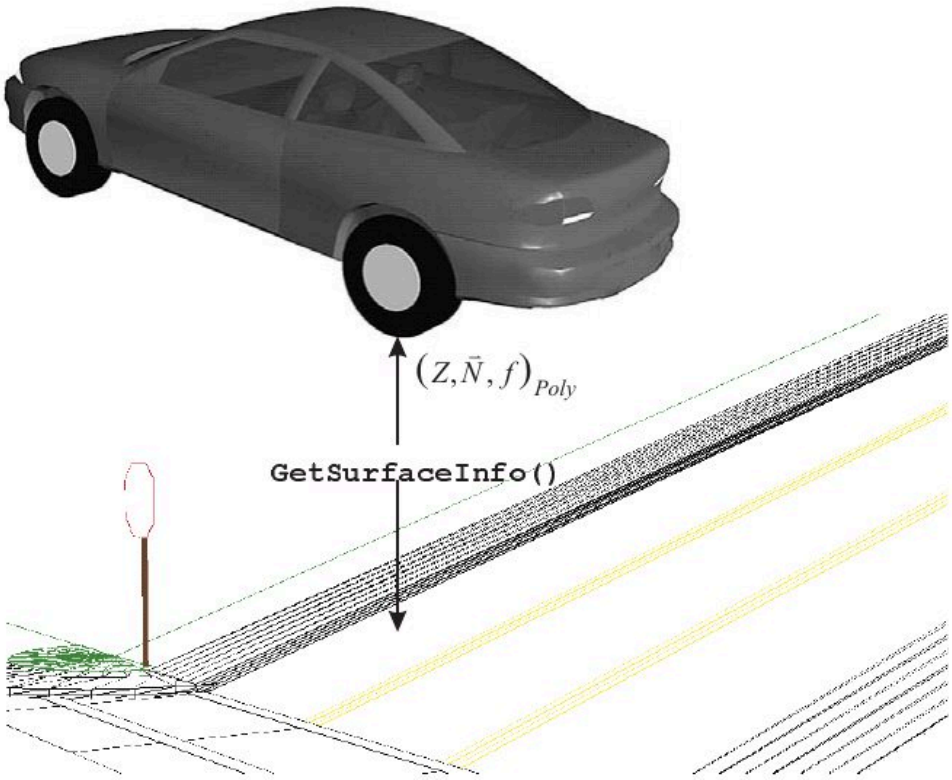


Figure 4-10: The surface-information search returns the terrain surface elevation, Z, friction multiplier, f, and surface normal, U, for the earth-fixed X,Y coordinates beneath the tire.

Defining the surface normal returned by the surface-information search as

$$\vec{U}_{Terrain} = \begin{bmatrix} U_{Terrain_X} \\ U_{Terrain_Y} \\ U_{Terrain_Z} \end{bmatrix}, \qquad \text{(Eq. 80)}$$

the angle between the terrain surface normal, $\vec{U}_{Terrain}$, and the wheel normal, $\vec{U}_w$, is

$$\alpha = \arccos(U_{Terrain_X} U_{w_X} + U_{Terrain_Y} U_{w_Y} + U_{Terrain_Z} U_{w_Z}) \qquad \text{(Eq. 81)}$$

and the inclination angle relative to the ground, Γ_w, is

$$\Gamma_W = \frac{\pi}{2} - \alpha \qquad \text{(Eq. 82)}$$

The steer angle in the ground plane is the product of the vehicle-fixed steer angle, δ_w, and the difference between the earth-fixed steer axis, $\vec{U}_\delta$, and surface normal, $\vec{U}_T$. The unit vector defining the earth-fixed angle of the steer axis is given by

$$\vec{U}_\delta = \begin{bmatrix} U_{\delta_X} \\ U_{\delta_Y} \\ U_{\delta_Z} \end{bmatrix} = A \cdot \begin{bmatrix} 0 \\ -\sin\gamma_w \\ \cos\gamma_w \end{bmatrix} \qquad \text{(Eq. 83)}$$

Then the steer angle relative to the ground, Ψ_w, is

$$\Psi_w = \delta_w (U_{Terrain_X} U_{\delta_X} + U_{Terrain_Y} U_{\delta_Y} + U_{Terrain_Z} U_{\delta_Z}) \qquad \text{(Eq. 84)}$$

Tire-Ground Contact Point

The tire-ground contact point is defined by the intersection of three planes: the wheel plane, the plane of the terrain beneath the wheel center, and a plane through the wheel center perpendicular to the first two planes. This point is defined by three equations:

$$\lambda_1 = X_w U_{w_X} + Y_w U_{w_Y} + Z_w U_{w_Z} \qquad \text{(Eq. 85)}$$

$$\lambda_2 = X_w U_{Terrain_X} + Y_w U_{Terrain_Y} + Z_w U_{Terrain_Z} \qquad \text{(Eq. 86)}$$

$$\lambda_3 = X_w D_1 + Y_w D_2 + Z_w D_3 \qquad \text{(Eq. 87)}$$

where

Symbol	Definition
λ_1	The equation defining the wheel plane
λ_2	The equation defining the plane for the terrain beneath the wheel
λ_3	The equation defining a plane perpendicular to λ_1 and λ_2 passing through the wheel center X_w, Y_w, Z_w
$\vec{U}_w$	Unit vector for the line in the earth-fixed coordinate system normal to the wheel plane (see Eq. 79)
$\vec{U}_{Terrain}$	Unit vector for the terrain surface normal returned by the surface-information search
$\vec{D}$	Unit vector defining a line normal to both the wheel plane and the plane for the terrain beneath the wheel

$$\vec{D} = \begin{bmatrix} D_1 \\ D_2 \\ D_3 \end{bmatrix} = \begin{bmatrix} U_{w_Y} U_{Terrain_Z} - U_{Terrain_Y} U_{w_Z} \\ U_{w_Z} U_{Terrain_X} - U_{Terrain_Z} U_{w_X} \\ U_{w_X} U_{Terrain_Y} - U_{Terrain_X} U_{w_Y} \end{bmatrix} \quad \text{(Eq. 88)}$$

The earth-fixed coordinates, $\vec{T} = [X_T \; Y_T \; Z_T]^T$, for the tire-ground contact point may now be solved directly using Eqs. 85 through 87. In matrix form,

$$\begin{bmatrix} U_{w_X} & U_{w_Y} & U_{w_Z} \\ U_{Terrain_X} & U_{Terrain_Y} & U_{Terrain_Z} \\ D_1 & D_2 & D_3 \end{bmatrix} \cdot \begin{bmatrix} X_T \\ Y_T \\ Z_T \end{bmatrix} = \begin{bmatrix} \lambda_1 \\ \lambda_2 \\ \lambda_3 \end{bmatrix} \quad \text{(Eq. 89)}$$

Tire Radial Deflection

Once the contact point is determined, the tire radial deflection can be calculated directly from the earth-fixed distance, D, between the wheel center, X_w, Y_w, Z_w, and the tire-ground contact point, X_T, Y_T, Z_T:

$$D = \sqrt{(X_w - X_T)^2 + (Y_w - Y_T)^2 + (Z_w - Z_T)^2} \quad \text{(Eq. 90)}$$

Based on the distance from the wheel center to the terrain contact point, the current tire rolling radius, r_T, is

$$r_T = \begin{cases} D, & \text{for } D < r_u \\ r_u, & \text{for } D \geq r_u \end{cases} \quad \text{(Eq. 91)}$$

It is now a simple matter to calculate the tire radial deflection in the wheel plane by comparing the tire unloaded radius, r_u, with the current tire radius, r_T:

$$\delta_T = r_u - r_T \quad \text{(Eq. 92)}$$

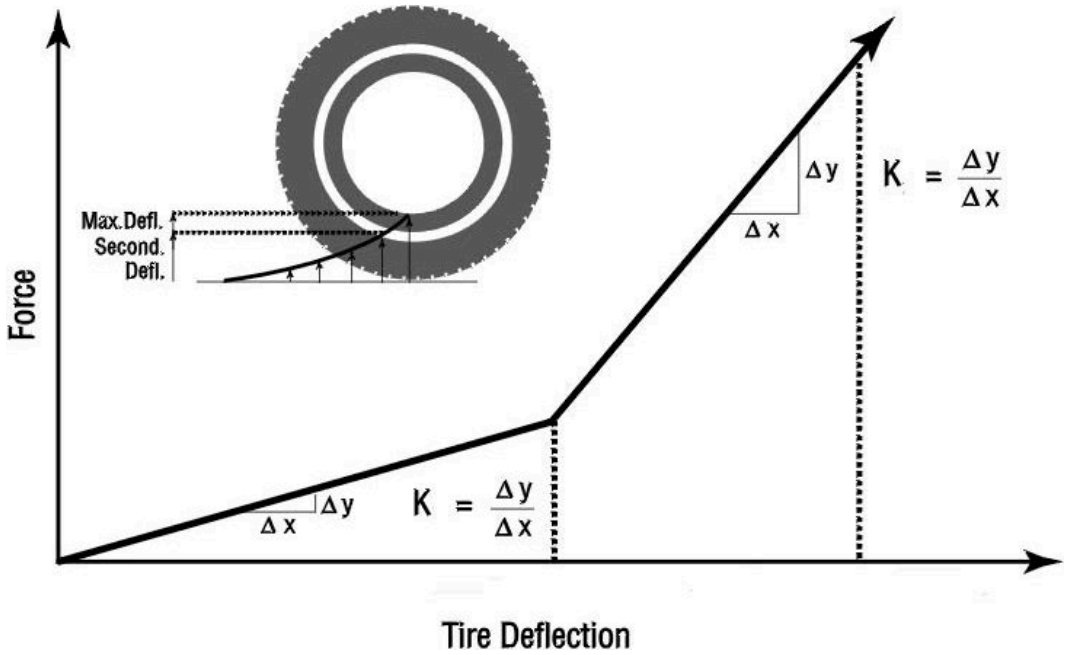


Figure 4-11: 2-step radial tire stiffness used for calculating the current tire radial force.

Tire Radial Force

Tire radial force is calculated in the direction of the tire deflection according to the tire force-deflection properties (see Figure 4-11) and current deflection,

$$F_R = \begin{cases} K_T\,\delta_r, & \text{for } \delta_r < \delta_2 \\ [4pt] K_T(\bar{K}_T(\delta_r - \delta_2) + \delta_2), & \text{for } \delta_r \geq \delta_2 \end{cases} \qquad \text{(Eq. 93)}$$

where

Symbol	Definition
F_R	Radial tire force (force in the tire plane in the direction of the tire deflection)
K_T	Tire initial radial stiffness
$\bar{K}_T$	Tire secondary stiffness multiplier
δ_2	Tire radial deflection at which $\bar{K}_T$ begins

(updated: earlier editions included a restitution characteristic that reduced the radial force while the tire was rebounding. The radial force is loaded and unloaded along the same curve; the tire's rebound restitution value is not applied.)

Radial stiffness during a tire blow-out

While a tire is blowing out, its initial radial stiffness is ramped linearly from its normal value to its blown value over the blow-out duration:

$$K_T(t) = \begin{cases} K_{T_0} & t \leq t_{BO} \\ [4pt] K_{T_0}\left(1 + (m_K - 1)\dfrac{t - t_{BO}}{\Delta t_{BO}}\right) & t_{BO} < t < t_{BO} + \Delta t_{BO} \\ [10pt] m_K\,K_{T_0} & t \geq t_{BO} + \Delta t_{BO} \end{cases} \qquad \text{(Eq. 93a)}$$

where m_K is the blow-out *Stiffness Factor*, t_{BO} the blow-out start time and Δt_{BO} the blow-out duration. The secondary stiffness multiplier is divided by the same factor, so that the second segment of the force-deflection curve keeps the slope it had before the blow-out. The blow-out treatment of cornering stiffness, camber stiffness and rolling resistance is described under *Tire Model — Tire blow-out*.

(updated: the effect of a tire blow-out on radial stiffness was not described in earlier editions.)

Deflection limits

If the current tire deflection, δ_r, exceeds the maximum allowable tire deflection, δ_{max}, by more than 20 percent, SIMON terminates and returns a message to HVE telling the user that the current tire deflection is excessive. The check is not applied to a wheel carrying its own collision mesh, because deflections during a wheel impact legitimately exceed it.

> **NOTE:** δ_{max} is user-editable (see Tire, Physical Properties). By default, δ_{max} is set equal to the section height of the tire. Tire deflection beyond δ_{max} indicates wheel rim deformation — a condition beyond the scope of the tire model. (To properly model such behavior would require non-linear finite element analysis.)

(updated: earlier editions gave the termination threshold as δ_{max} itself. A 20 percent margin is allowed before the run is stopped.)

Separately, while a tire is inclined more than about 80 degrees to the ground — during a rollover, for example — the deflection used in Eq. 93 is clamped to δ_2. Without this limit the geometry of a nearly horizontal wheel would produce a very large apparent deflection and an unrealistic radial force.

(updated: the rollover deflection limit was not described in earlier editions.)

Tire radial force, F_R, is an important fundamental property as one of the primary inputs to the tire model which is used to calculate braking and cornering force.

Tire Model

SIMON uses the EDC semi-empirical tire model developed for EDVDS [2]. The basis for the EDC semi-empirical tire model is the HSRI tire model, developed at the University of Michigan Transportation Research Institute [3]. The model was extended to allow large tire slip angles, drive torque (i.e., tire forces that accelerate the vehicle) and drive and/or brake torque when the vehicle is rolling backwards. The SIMON implementation for load- and speed-dependent tire properties has been extended by replacing the method of partial derivatives with a table look-up method. An overview of the extended model is provided below for purposes of comparison with the HSRI and EDVDS tire models.

Tire model versions

(updated: this subsection describes behavior added since the Fifth Edition manual, which documented only a single semi-empirical tire model.)

The Calculation Options dialog offers three versions of the semi-empirical tire model. They share the structure described in the rest of this section and differ only in the specific respects listed below. **Vers. 3 is the default for a new event**, and is the recommended choice for new work; Vers. 1 and Vers. 2 are retained so that results from earlier releases can be reproduced exactly.

	Vers. 1	Vers. 2	Vers. 3
Slip-angle function	sinα	tanα	tanα
Lateral force reverses when the tire travels rearward	No	Yes	Yes
Effective friction limited to the peak friction value	No	No	Yes
Low-speed cornering and camber stiffness reduction	No	No	Yes

The three differences are described where they arise in the development below: the slip-angle function in Eq. 103a, the rearward-travel sign in Eqs. 106 and 112, the friction limit in Eq. 102b and the low-speed reduction in Eq. 104a.

Selecting a version affects only the tire shear-force calculation. It does not change the radial force model, the rolling resistance model or the hydroplaning model, all of which are common to all three versions.

Model assumptions and inputs

The semi-empirical tire model describes empirically what is occurring at the tire-road shear interface, according to the current tire-road conditions. It employs a simplified theory assuming an adhesion region and a sliding region. The major assumptions made by the tire model are:

- The contact patch can be divided into two regions: an adhesion region and a sliding region.
- The shear force generated in the adhesion region depends on the elastic properties of the tire, and the shear force generated in the sliding region depends on the frictional properties at the tire-road interface.

The inputs required by the tire model are

Input	Definition
F_o	Vertical load for up to three test loads
V_o	Longitudinal velocity for up to three test speeds
μ_p, μ_s, S_{μ_p}	Peak and slide tire-road friction and longitudinal slip at peak friction for each load and speed (these data generate a μ-slip curve for each test load and speed; see Figure 4-12)
C_s	Longitudinal stiffness for each test load and speed
C_α	Cornering stiffness for up to three test loads and speeds (see Figure 4-13)
C_γ	Camber stiffness for up to three test loads and speeds (see Figure 4-14)

During execution, the current tire radial load, F_R, rolling radius, r_T, and forward velocity component in the ground plane, u_{GP}, are calculated (see Eqs. 93, 91 and 74, respectively). In addition, the current longitudinal slip, S, is calculated:

$$S = 1 - \Omega(\frac{r_T}{u_{GP}}) \qquad \text{(Eq. 94)}$$

S is limited to the range $-1 \le S \le 1$. When the tire is travelling too slowly for Eq. 94 to be meaningful (a forward ground-plane speed below approximately 1 in/sec), longitudinal slip is instead set to zero if the wheel is also not spinning, or to ±1 if the wheel is spinning while the vehicle slides sideways.

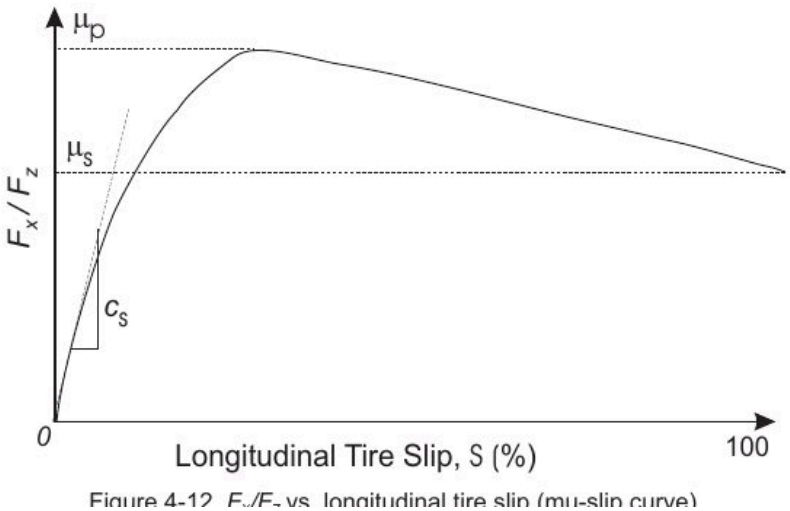


Figure 4-12 Fx/Fz vs. longitudinal tire slip (mu-slip curve)

Figure 4-12: Fx/Fz vs. longitudinal tire slip (mu-slip curve).

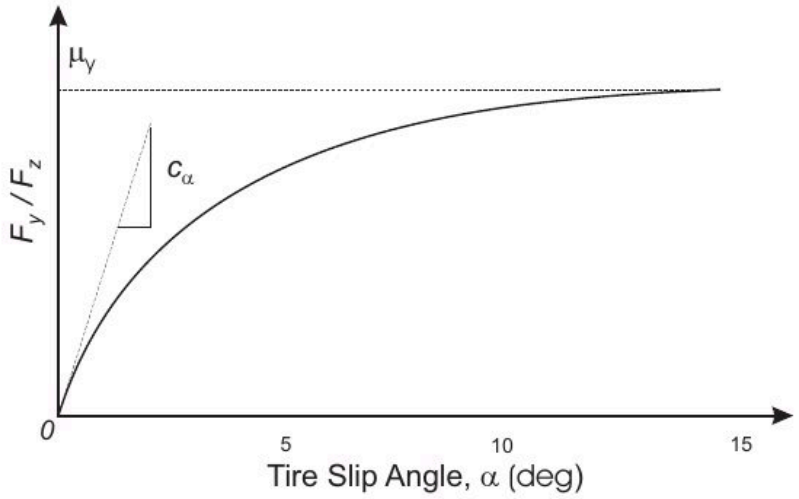


Figure 4-13: Fy/Fz vs. tire slip angle, α.

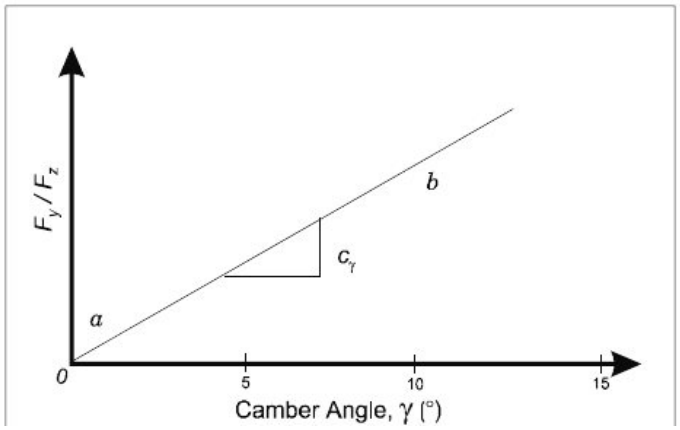


Figure 4-14: Fy/Fz vs. tire inclination angle, γ.

Based on the above test tire parameters and the current tire load and velocity, the effective values are calculated using linear interpolation,

$$\mu_p' = f(F_R, u_{GP}), \quad \mu_s' = f(F_R, u_{GP}), \quad S_{\mu_p}' = f(F_R, u_{GP}), \quad C_s' = f(F_R, u_{GP}), \quad C_\alpha' = f(F_R, u_{GP}), \quad C_\gamma' = f(F_R, u_{GP})$$

(updated: the longitudinal stiffness, C_s', is now interpolated from the tire's Longitudinal Stiffness data in the same way as the other load- and speed-dependent properties. Earlier editions of this manual gave a formula that derived C_s from the μ-slip curve parameters; that formula is no longer used.)

The interpolated peak and slide friction values are then scaled by the friction multiplier of the terrain beneath the tire:

$$\mu_p' \leftarrow \mu_p' \cdot f_{terrain}, \qquad \mu_s' \leftarrow \mu_s' \cdot f_{terrain} \qquad \text{(Eq. 94a)}$$

where $f_{terrain}$ is the surface friction multiplier returned for the current tire position, further reduced by the hydroplaning model when one is selected (see *Hydroplaning Model*, below). Note that the terrain friction multiplier scales friction only; it does not scale longitudinal, cornering or camber stiffness.

Effective friction

From these data, the following intermediate parameters are computed:

$$a = (1.0 + S_{\mu_p}')(1.0 - S_{\mu_p}')^2 \qquad \text{(Eq. 95)}$$

$$b = (1.0 - S_{\mu_p}')(\mu_s'(S_{\mu_p}' + 2.0) - \mu_p'(2.0 S_{\mu_p}' + 1.0)) \qquad \text{(Eq. 96)}$$

$$c = (\mu_s' - \mu_p')\mu_s' \qquad \text{(Eq. 97)}$$

$$B = \frac{-b + \sqrt{b^2 - 4ac}}{2a} \qquad \text{(Eq. 98)}$$

$$A = \mu_s' + B \qquad \text{(Eq. 99)}$$

$$C = \mu_s' + B(1.0 - S_{\mu_p}') \qquad \text{(Eq. 100)}$$

The effective friction, μ, is then

$$\mu = A - B|S| \qquad \text{(Eq. 102)}$$

μ is subject to two limits. First, it is never allowed to fall below the value it would take at fully locked or fully spinning slip:

$$\mu \geq A - B \qquad \text{(Eq. 102a)}$$

Second, in **Vers. 3** only, the effective friction is not allowed to exceed the interpolated peak friction:

$$\mu \leq \mu_p' \qquad \text{(Vers. 3 only)} \qquad \text{(Eq. 102b)}$$

(updated: Eqs. 102a and 102b describe limits not documented in earlier editions. Eq. 102b is one of the differences between Vers. 3 and the earlier versions; in Vers. 1 and Vers. 2 the effective friction may momentarily exceed the peak value returned by the friction table.)

Cornering stiffness at large slip angles

The interpolated cornering stiffness, C_α', describes the tire's response to small slip angles. Because the lateral force per degree of slip angle falls off as the slip angle grows, SIMON reduces the cornering stiffness used in the force calculation as a function of the magnitude of the current slip angle:

$$C_\alpha = \begin{cases} C_\alpha' & |\alpha| \leq 1^\circ \\ [4pt] C_\alpha'\left(1.0 - \frac{1}{3}\cdot\frac{|\alpha| - 1^\circ}{4^\circ}\right) & 1^\circ < |\alpha| < 5^\circ \\ [8pt] \frac{2}{3}C_\alpha' & |\alpha| \geq 5^\circ \end{cases} \qquad \text{(Eq. 100a)}$$

The reduction is symmetric about 180 degrees, so that a tire travelling rearward sees the same treatment as one travelling forward.

(updated: Eq. 100a describes behavior not documented in earlier editions of this manual.)

Adhesion and sliding regions

Based on the above parameters, the fraction of the adhesion region of the total contact patch, X_s / L (where L is the total length of the contact patch, and X_s is the distance from the front of the contact patch to the point where sliding starts), is calculated as follows:

$$D_t = \sqrt{(C_s' S)^2 + (C_\alpha T(\alpha))^2} \qquad \text{(Eq. 103)}$$

$$\frac{X_s}{L} = \frac{\mu F_z (1.0 - |S|)}{2.0 D_t} \qquad \text{(Eq. 104)}$$

where $T(\alpha)$ is the slip-angle function, which is the single largest difference between Vers. 1 and the later versions:

$$T(\alpha) = \begin{cases} \sin\alpha & \text{Vers. 1} \\ \tan\alpha & \text{Vers. 2 and Vers. 3} \end{cases} \qquad \text{(Eq. 103a)}$$

In Vers. 2 and Vers. 3 the tangent is guarded against overflow: within a quarter of a degree of $\pm 90^\circ$, $T(\alpha)$ is clamped to a large finite value carrying the sign of the tire's forward ground-plane velocity, so that the lateral force saturates instead of diverging.

(updated: earlier editions of this manual stated that the EDC model replaces the HSRI tangent function with the sine function. That is true of Vers. 1 only. Vers. 2 and Vers. 3 restore the tangent function — which is the correct form for the underlying brush model — and handle the singularity at $\pm 90^\circ$ by clamping rather than by substituting the sine.)

X_s / L is limited to 1.0 (note that $X_s / L = 1.0$ means there is no sliding region).

Low-speed refinement

*(updated: this subsection describes behavior added since the Fifth Edition manual. It applies to **Vers. 3** only.)*

At very low wheel speeds, the cornering and camber stiffnesses of the undisturbed model produce lateral forces that are unrealistically large for the amount of relative motion actually present at the contact patch, which can make a nearly stopped vehicle drift or jitter. Vers. 3 therefore fades the stiffnesses toward zero as the wheel comes to rest.

When the tire's circumferential speed, $r_T \Omega$, falls below approximately 2 ft/sec, the cornering and camber stiffnesses used in the force calculation are scaled by a blending factor:

$$\Phi(v, v_{ref}) = \frac{2.0}{\frac{v_{ref}}{|v|} + \frac{|v|}{v_{ref}}} \qquad \text{(Eq. 104a)}$$

$$C_\alpha \leftarrow C_\alpha\, \Phi(r_T \Omega,\, v_{ref}), \qquad C_\gamma \leftarrow C_\gamma\, \Phi(r_T \Omega,\, v_{ref}) \qquad \text{(Eq. 104b)}$$

where $v_{ref} \approx 2$ ft/sec. Φ equals 1.0 when $|v| = v_{ref}$ and approaches zero as $|v| \to 0$, so the reduction phases in smoothly rather than switching on at a threshold. If the wheel is exactly stopped, both stiffnesses are set to zero.

While the wheel is below that speed, if the lateral component of the tire's ground-plane velocity also falls below approximately 1 in/sec, the slip-angle function is faded by the same blending factor:

$$T(\alpha) \leftarrow T(\alpha)\,\Phi(v_{GP},\ 1\ \text{in/sec}) \qquad \text{(Eq. 104c)}$$

again with $T(\alpha)$ set to zero if the lateral velocity is exactly zero.

Tire forces

If X_s / L = 1.0, there is no sliding region and the longitudinal and lateral tire forces, $F_{x'}$ and $F_{y'}$, respectively, are

$$F_{x'} = s_u \frac{C'_s S}{1.0 - |S|} \qquad \text{(Eq. 105)}$$

and

$$F_{y'} = \frac{-C_\alpha T(\alpha)}{1.0 - |S|} \cdot k_u \qquad \text{(Eq. 106)}$$

where s_u and k_u account for the direction the tire is travelling:

$$s_u = -\,\text{sgn}(u_{GP}), \qquad k_u = \begin{cases} 1 & \text{Vers. 1} \\ -s_u & \text{Vers. 2 and Vers. 3} \end{cases} \qquad \text{(Eq. 106a)}$$

s_u orients the longitudinal force so that it always opposes the direction of travel for a braking slip and drives the vehicle for a drive slip, whether the tire is rolling forward or backward. k_u reverses the lateral force when the tire is travelling rearward, so that a slip angle measured relative to the wheel plane produces a lateral force in the correct direction in reverse. Vers. 1 omits this correction, and consequently returns a lateral force of the wrong sign for a tire travelling rearward.

(updated: the direction-of-travel factors in Eq. 106a were not documented in earlier editions. k_u is one of the differences between Vers. 1 and the later versions.)

$F_{y'}$ is set to zero when both ground-plane velocity components of the tire are below approximately 1 in/sec, so that a stopped tire generates no lateral force.

If X_s / L < 1.0, there is some sliding at the tire-road interface. For this condition, the tire forces in the adhesion region and sliding region are computed separately. $F_{x'}$ and $F_{y'}$ tire forces in the adhesion region are

$$F_{x'_{Adhesion}} = C'_s S \left(\frac{\mu F_z}{2.0 D_t}\right)^2 (1.0 - |S|) \qquad \text{(Eq. 107)}$$

and

$$F_{y'_{Adhesion}} = -\,C_\alpha T(\alpha) \left(\frac{\mu F_z}{2.0 D_t}\right)^2 (1.0 - |S|) \qquad \text{(Eq. 108)}$$

The tire force components in the sliding region are:

$$F_{x'_{Sliding}} = \mu F_z \left(1.0 - \frac{X_s}{L}\right)\left(\frac{S}{\sqrt{S^2 + T(\alpha)^2}}\right) \qquad \text{(Eq. 109)}$$

and

$$F_{y'_{Sliding}} = -\,\mu F_z \left(1.0 - \frac{X_s}{L}\right)\left(\frac{T(\alpha)}{\sqrt{S^2 + T(\alpha)^2}}\right) \qquad \text{(Eq. 110)}$$

The total tire force is the sum of the force in the adhesion and sliding regions, with the same direction-of-travel factors applied as in Eqs. 105 and 106,

$$F_{x'} = s_u (F_{x'_{Adhesion}} + F_{x'_{Sliding}}) \qquad \text{(Eq. 111)}$$

and

$$F_{y'} = (F_{y'_{Adhesion}} + F_{y'_{Sliding}}) k_u \qquad \text{(Eq. 112)}$$

Camber force

The lateral force produced by the tire's inclination relative to the ground is added to the lateral force from Eq. 106 or Eq. 112:

$$F_{y'} \leftarrow F_{y'} + C_\gamma \sin\gamma_G \qquad \text{(Eq. 112a)}$$

where γ_G is the tire inclination angle relative to the ground plane (see *Tire-Ground Orientation*) and C_γ is the camber stiffness — the value interpolated from the Camber Stiffness table, modified by the tire blow-out model if one is active and by the low-speed refinement of Eq. 104b.

(updated: the camber contribution to the lateral tire force was listed as a model input in earlier editions but the equation using it was not given.)

Tire blow-out

(updated: this subsection describes how the HVE Tire Blow-out Model, which is set up on the Wheels dialog, enters the tire force calculation. Earlier editions of this manual documented the blow-out inputs but not their effect on the tire model.)

When a tire is flagged as blowing out and the simulation time has reached the blow-out start time, the cornering stiffness, camber stiffness and both rolling resistance coefficients are ramped linearly from their normal values to their blown values over the specified blow-out duration:

$$X(t) = \begin{cases} X_0 & t \le t_{BO} \\ X_0 + (m_X X_0 - X_0)\dfrac{t - t_{BO}}{\Delta t_{BO}} & t_{BO} < t < t_{BO} + \Delta t_{BO} \\ m_X X_0 & t \ge t_{BO} + \Delta t_{BO} \end{cases} \qquad \text{(Eq. 112b)}$$

where X is any of the affected parameters, X_0 is its unblown value, t_{BO} is the blow-out start time, Δt_{BO} is the blow-out duration, and m_X is the *Stiffness Factor* (for cornering and camber stiffness) or the *Rolling Resistance Factor* (for the two rolling resistance coefficients). The reduced values are used for the remainder of the event.

The blow-out model also changes the tire's radial stiffness; that effect is described under

Tire Radial Force

, above.

Steeply inclined tires

(updated: this subsection describes behavior not documented in earlier editions.)

The tire force calculation is valid only while the tire's contact plane is reasonably close to horizontal relative to the wheel. When the tire inclination angle relative to the ground exceeds approximately 80 degrees, the model can no longer resolve a meaningful contact patch. In that case:

- If the wheel is being contacted by the 3-D mesh collision model, all tire forces, ploughing forces, slip values and the rolling resistance moment are faded smoothly toward zero using the blending factor of Eq. 104a, applied to the cosine of the inclination angle. This avoids a discontinuity in the forces during a wheel impact.
- Otherwise, all of these quantities are set to zero for that tire.

Skidding

The tire model uses the current vertical tire load, F_z, the current longitudinal slip and the fraction of the sliding region at the tire-road interface to set the skid flag, SF, which controls tiremark generation:

$$\text{if } F_Z > F_{z,min} \text{ and } \left\{ \begin{array}{c} |S| > S'_{\mu_p} \\ \text{or} \\ \frac{X_s}{L} < \tau \end{array} \right\}, \quad SF = \text{TRUE} \qquad \text{(Eq. 113)}$$

where $F_{z,min}$ is the minimum tire load for tiremark generation and $\tau = 0.25$. In words, a tire is marked as skidding when it is carrying enough load to leave a mark and either its longitudinal slip has passed the slip at peak friction, or less than a quarter of its contact patch remains in adhesion.

(updated: earlier editions of this manual gave only the adhesion-region condition. The longitudinal slip condition is also present and either one is sufficient.)

The tire model additionally reports a normalized lateral saturation value, used in tiremark determination, which compares the lateral force the tire would develop from its cornering stiffness alone against the friction available:

$$\bar{A} = \frac{1}{2} \min\left(\left|\frac{C_\alpha T(\alpha)}{\mu F_z}\right|,\ 2.0\right) \qquad \text{(Eq. 113a)}$$

$\bar{A}$ ranges from 0.0 (no lateral demand) to 1.0 (lateral demand at or beyond twice the available friction) and is reported as the tire's lateral slip output variable.

Summary of changes from the HSRI model

In the above development, the original HSRI model has been modified in the following ways:

1. Load- and speed-dependent tire parameters are calculated from data tables, using linear interpolation, rather than using partial derivatives.
2. Longitudinal slip, S, has been replaced by $|S|$ wherever it appears as a magnitude, to allow for drive torque at the tire-road interface and for a vehicle rolling backwards.
3. The singularity in the HSRI lateral force at $\alpha = \pm 90^\circ$ is removed. Vers. 1 does this by substituting the sine function for the tangent, which allows the model to handle slip angles throughout the continuous range $-\pi \le \alpha \le \pi$ but changes the small-angle behavior. Vers. 2 and Vers. 3 retain the tangent function and instead clamp it near $\pm 90^\circ$, preserving the correct small-angle response.
4. Direction-of-travel factors are applied to the tire forces (Eq. 106a) so that both longitudinal and lateral forces act in the correct direction when the vehicle is travelling rearward.
5. Cornering stiffness is reduced at large slip angles (Eq. 100a), and, in Vers. 3, cornering and camber stiffness are faded at very low wheel speeds (Eq. 104b).

Rolling Resistance

The original EDC semi-empirical tire model has also been extended to include a circumferential tire moment from rolling resistance. The rolling resistance moment, $M_{Rolling}$, is given by

$$M_{Rolling} = r(\sigma_0 F_R + \sigma_v |u_{GP}|) \cdot (-\text{sgn}(\Omega)) \qquad \text{(Eq. 114)}$$

where

Symbol	Definition
r	Tire rolling radius
σ_0	Load-dependent rolling resistance coefficient
σ_v	Velocity-dependent rolling resistance coefficient
F_R	Radial tire force
u_{GP}	Forward component of tire velocity in ground plane
Ω	Wheel spin angular velocity

The $-\text{sgn}(\Omega)$ factor makes the moment resist wheel rotation. For $|\Omega|$ below a minimum spin threshold the moment is linearly ramped toward zero to avoid chatter.

The rolling resistance moment contributes to the equations of motion for the wheel spin degree of freedom (see Eqs. 27 through 32).

Hydroplaning Model

(updated: this section documents a capability added since the Fifth Edition manual.)

When a tire rolls through standing water, a wedge of water builds ahead of the contact patch. Above a critical speed the wedge lifts the tire off the pavement and the available tire-road friction collapses. SIMON models this effect by testing each tire, at each timestep, against a hydroplaning

threshold speed

computed from the tire and surface conditions, and switching that tire's friction between its dry-surface and water-surface values accordingly.

Selecting a model

The hydroplaning model is chosen in the SIMON Calculation Options dialog (**Hydroplaning Model**):

Option	Notes
Off	Default. Friction is the dry-surface value.
NASA	NASA (Horne/Borne) spin-down model.
NASA-TTI	Modified NASA model including contact-patch aspect ratio.
Gallaway	Gallaway/TTI model including water depth and pavement macrotexture.

A fifth model, *Blythe-Day*, is named in the program's message text but is **not implemented**: it is not offered as a radio button in the dialog, and any attempt to request it is rejected with a fatal error, so the event will not run.

> **NOTE:** Hydroplaning cannot be combined with the Soft Soil tire-terrain model. Selecting both produces a fatal error and the event will not run.

Required input data

Hydroplaning is evaluated only where a tire is in contact with an environment polygon whose object type is **Water**. Each water polygon carries its own water depth (static or dynamic), friction and material properties, assigned in the 3-D Editor's Object Attributes dialog. If the environment contains no water polygons, selecting a hydroplaning model has no effect on the simulation.

The water polygon's **material friction** is the value the tire's friction multiplier switches to while hydroplaning, so it — not the tread or pressure data — determines *how much* friction remains once the threshold speed is exceeded. The tire and surface data below determine only *when* that switch occurs.

> **NOTE:** The friction used while hydroplaning comes from the water polygon's material friction, assigned in the 3-D Editor's Object Attributes dialog.

The models draw on the following data:

Symbol	Quantity	Source
P	Tire inflation pressure (lb/in²)	Tire Physical Data — *Nominal Pressure*
w_t	Tread width (in)	Tire Physical Data — *Tread Width*
d_t	Tread depth (in)	Tire Physical Data — *Tread Depth*
l_t	Contact patch (tread) length (in)	Computed each timestep from the deflected tire radius
WD	Water depth (in)	Water polygon (static or dynamic)
TXD	Pavement macrotexture depth (in)	Environment material properties
u_{Tire}	Forward tire velocity in the ground plane (in/sec)	Simulation state

The contact patch length is obtained from the undeflected and deflected tire radii,

$$l_t = 2\sqrt{r_0^2 - r^2}$$

so a more heavily loaded (more deflected) tire presents a longer contact patch, which raises the NASA-TTI threshold speed.

NASA (Horne/Borne) model

The NASA model predicts the spin-down (total hydroplaning) speed purely from inflation pressure. A tire hydroplanes when

$$|u_{Tire}| > 182.16\sqrt{P}$$

with u_{Tire} in in/sec and P in lb/in². The coefficient is the familiar $v_p = 10.35\sqrt{P}$ mph expressed in in/sec ($10.35 \times 17.6 = 182.16$).

NASA-TTI model

The NASA-TTI model modifies the NASA prediction using the aspect ratio of the contact patch, so that tire deflection and tread geometry influence the threshold. A tire hydroplanes when

$$|u_{Tire}| > \frac{83.35 - 27.59\,\frac{w_t}{l_t} + 1.158319\,P}{0.09144}$$

The bracketed expression evaluates the threshold in km/h; dividing by 0.09144 converts it to in/sec. The implementation follows the form given by Ong and Fwa citing the 1986 Horne/TTI/NASA work. (An alternative expression attributed to Badger, $139.92\sqrt{P\,w_t / l_t}$, is not used, because no supporting reference was available for it.)

If the computed contact patch length is less than 0.5 in — for example, a tire carrying almost no load — the NASA-TTI model reports no hydroplaning rather than dividing by a vanishing patch length.

Gallaway (TTI) model

The Gallaway model is the most detailed of the three, adding water depth and pavement macrotexture. The predicted hydroplaning speed V (mph) is

$$V = SD^{0.04}\,P^{0.3}\,(TD + 1)^{0.06}\,A$$

where TD is the tread depth expressed in 32nds of an inch ($TD = 32\,d_t$), SD is the tire spin-down percentage, which is 10 percent, and A is the larger of two water-depth/texture terms:

$$A = \max(A_1,\ A_2)$$

$$A_1 = \frac{10.409}{WD^{0.06}} + 3.507 \qquad A_2 = \left(\frac{28.952}{WD^{0.06}} - 7.817\right)TXD^{0.14}$$

For water depths below 0.005 in the water-depth terms are dropped, giving $A_1 = 3.507$ and $A_2 = -7.817\,TXD^{0.14}$, which avoids the singularity as $WD \to 0$. The tire hydroplanes when

$$|u_{Tire}| > 17.6\,V$$

(the factor 17.6 converts V from mph to in/sec).

Effect on the tire forces

Each model returns the friction multiplier, F_{Mult}, that scales the tire's friction coefficients. Two multipliers are available for every tire: one for the Road, Friction Zone or Curb polygon under the tire, and a second taken from the material friction of the **Water** polygon under the tire. Below the threshold speed the hydroplaning model returns the first (dry-surface) value; at or above the threshold it returns the water value. The returned multiplier is applied to *both* the peak and sliding friction coefficients interpolated from the tire's friction tables:

$$\mu_{peak} = F_{Mult}\,\mu_p(F_z, u) \qquad \mu_{slide} = F_{Mult}\,\mu_s(F_z, u)$$

Because μ_{peak} and μ_{slide} set the ceiling on the shear force the contact patch can develop, substituting the (much lower) water friction for the pavement friction reduces the tire force available in *every* direction. Longitudinal force is lost, so braking and drive traction fall away; lateral force is lost, so the tire can no longer generate the side force needed to hold a curve or resist yaw. This loss of available friction is the entire mechanism by which hydroplaning affects the simulated vehicle; the extra drag of displacing water is not modeled.

> **NOTE:** If no water polygon lies under the tire, the water multiplier is the same as the dry-surface multiplier, so selecting a hydroplaning model has no effect unless the environment actually contains water polygons beneath the tire path.

Because the transition is a threshold test evaluated per tire per timestep, an individual tire may enter and leave hydroplaning repeatedly during an event — for example as it crosses a puddle, as load transfer changes its contact patch, or as the vehicle decelerates back below the threshold speed.

Hydroplaning state and reporting

SIMON tracks a hydroplaning state for each tire and records when hydroplaning starts and stops:

State	Meaning
Not yet hydroplaned	Tire has not yet hydroplaned.
Hydroplaning	Tire is currently hydroplaning; start time and speed recorded.
Contact regained	Tire has regained contact; end time and speed recorded.

The simulation time and forward tire speed are stored at both the start and the end of each hydroplaning episode. A tire that regains and then again loses contact returns to the hydroplaning state.

When a hydroplaning model is active, SIMON also fills the water-related tire output tracks — *Water Depth*, *Macrotexture* and the water friction multiplier — which can be plotted or tabulated in a Variable Output report (see Chapter 3). The *Rolling Resist Mult*, *Fx' (water)* and *Fy' (water)* tracks are not used.

Suspension Force

Independent and solid axle suspensions are supported in SIMON. The equations of motion for each suspension type were discussed earlier (see Eqs. 7 through 10 and 18 through 26). The force is calculated using a spring-damper model with additional coulomb friction, as shown in the model in Figure 4-15. The spring is free to move only in the vehicle-fixed z-direction. The total suspension force is the sum of the spring force, the force from any engaged suspension stop, the damping (shock absorber) force, the coulomb friction force, the anti-sway bar force, the jacking force and the static suspension force. Mathematically, the force, F_S, at each suspension location is

$$F_S = K(\delta - \delta_{Payload}) + F_{Stop} + C\dot{\delta} + F_\mu + F_{Sway} + F_{Jack} + F_{S_0} \qquad \text{(Eq. 115)}$$

where

Symbol	Definition
K	Linear spring rate of suspension spring
δ	Spring deflection from equilibrium (+ down), Eq. 115a
$\delta_{Payload}$	Static deflection produced by the payload, Eq. 115c
F_{Stop}	Force from an engaged jounce or rebound stop, Eq. 116
C	Velocity-dependent damping rate
$\dot{\delta}$	Spring deflection rate, Eq. 115b
F_μ	Coulomb friction, Eq. 115d
F_{Sway}	Anti-sway bar force, Eq. 115e
F_{Jack}	Jacking force (independent suspensions only), Eq. 115f
F_{S_0}	Static suspension force at this suspension location

(updated: earlier editions of this manual gave F_S as the sum of the spring, damping, coulomb friction and anti-sway bar forces only. The static suspension force, the suspension stop force and the jacking force are also part of the total. Earlier editions also stated that the suspension force calculation is identical for the two suspension types; the spring deflection, deflection rate, axle roll angle, anti-sway bar force and jacking force are each calculated differently for independent and solid axle suspensions, as described below.)

Spring deflection and deflection rate

For an **independent suspension**, each side has its own vertical suspension degree of freedom, and the spring deflection and deflection rate are simply that degree of freedom and its derivative:

$$\delta = z_{Susp} + \delta_{Payload}, \qquad \dot{\delta} = \dot{z}_{Susp} \qquad \text{(Eq. 115a, independent)}$$

For a **solid axle suspension**, the axle has a single vertical degree of freedom and a roll degree of freedom, and the deflection at each spring depends on the spring's lateral position on the axle:

$$\delta = z_{Axle} + \phi_{Axle}\, y_s + \delta_{Payload}, \qquad \dot{\delta} = \dot{z}_{Axle} + \dot{\phi}_{Axle}\, y_s \qquad \text{(Eq. 115b, solid axle)}$$

where y_s is the vehicle-fixed y-coordinate of the spring and ϕ_{Axle} is the vehicle-fixed axle roll angle.

(updated: Eqs. 115a and 115b were not given in earlier editions.)

Payload static deflection

Adding a payload changes the static load carried by each suspension, and therefore the position each suspension sits at when the event begins. That change is applied as a fixed offset:

$$\delta_{Payload} = \frac{F_{S_0} - F_{S_0,Empty}}{K} \qquad \text{(Eq. 115c)}$$

where F_{S_0} is the static suspension force with the payload in place and $F_{S_0,Empty}$ is the static suspension force for the vehicle without it. $\delta_{Payload}$ is limited so that it cannot place the suspension beyond its jounce travel limit. The offset is also applied to the initial wheel center and steer axis positions, so that the vehicle begins the event at the ride height corresponding to its load.

Note in Eq. 115 that the offset is removed again before the spring force is computed. The result is that the spring force is measured from the vehicle's loaded equilibrium position, while the suspension stops engage at the loaded position — that is, a heavily loaded vehicle begins the event closer to its jounce stop, as it should, but does not carry a spring preload from the payload in addition to its static suspension force.

(updated: this subsection describes behavior not documented in earlier editions.)

Coulomb friction

The coulomb friction force opposes suspension motion at a constant magnitude, except at very low deflection rates, where it is blended linearly through zero so that the force does not chatter as the suspension changes direction:

$$F_\mu = \begin{cases} F_f \dfrac{\dot{\delta}}{\dot{\delta}_h}, & \dot{\delta}_h > 0 \text{ and } |\dot{\delta}| < \dot{\delta}_h \\ [10pt] F_f \operatorname{sgn}(\dot{\delta}), & |\dot{\delta}| > 0 \\ [4pt] 0, & \dot{\delta} = 0 \end{cases} \qquad \text{(Eq. 115d)}$$

where F_f is the suspension friction force and $\dot{\delta}_h$ is the hysteresis deflection rate. Setting the hysteresis rate to zero disables the blend and applies the full friction force at any non-zero deflection rate.

(updated: earlier editions listed coulomb friction as a term in Eq. 115 but did not give its definition.)

Anti-sway bar force

The anti-sway bar produces equal and opposite vertical forces at the two sides of an axle, proportional to the axle roll angle. The lever arm over which the auxiliary roll stiffness acts differs by suspension type:

$$F_{Sway} = \pm \frac{K_{rs}}{s} \phi_{Axle} \quad \text{(solid axle)}, \qquad F_{Sway} = \pm \frac{K_{rs}}{t} \phi_{Axle} \quad \text{(independent)} \qquad \text{(Eq. 115e)}$$

where

Symbol	Definition
K_{rs}	Auxiliary roll stiffness of the anti-sway bar
s	Lateral spring spacing on the axle (solid axle suspensions)
t	Current track width, the lateral distance between the two wheel centers (independent suspensions)
ϕ_{Axle}	Vehicle-fixed axle roll angle

The sign is positive on the right side of the axle and negative on the left.

For a solid axle suspension, ϕ_{Axle} is the axle roll degree of freedom. For an independent suspension there is no axle roll degree of freedom, so the axle roll angle is estimated from the current vertical positions of the two wheel centers:

$$\phi_{Axle} = \frac{z_{Wheel,Right} - z_{Wheel,Left}}{t} \qquad \text{(Eq. 115e-1)}$$

This is a small-angle approximation. If the track width of an independent suspension is less than one inch, the vehicle data are rejected as invalid and the event is terminated.

(updated: earlier editions gave a single anti-sway bar term, $K_{rs}\phi_{Axle}$ divided by the y-coordinate of the spring location, applied without a sign change between sides. The divisor is the lateral spring spacing for a solid axle and the track width for an independent suspension, and the force acts in opposite directions on the two sides.)

SIMON also reports an auxiliary roll moment for each axle,

$$M_{Roll} = K_{rs}\,\phi_{Axle} \qquad \text{(Eq. 115e-2)}$$

which appears in the variable output as the axle sway bar roll moment. This is a reported quantity only; the roll stiffness enters the equations of motion through the anti-sway bar forces of Eq. 115e.

Jacking force

On an independent suspension, the tire's lateral and vertical forces act at the tire-ground contact point, which is offset from the wheel center. As the suspension deflects, the linkage changes the wheel's camber angle and its lateral position. Those two geometric rates convert the tire forces into a vertical force at the spring — the jacking force:

$$F_{Jack} = \sum_{Tires} \left[\frac{d\gamma}{dz}\left(F_y h_{c_z} - F_z h_{c_y}\right) - F_y \frac{d(t/2)}{dz}\right] \qquad \text{(Eq. 115f)}$$

where

Symbol	Definition
$d\gamma/dz$	Rate of change of vehicle-fixed wheel camber angle with suspension jounce/rebound
$d(t/2)/dz$	Rate of change of the wheel center's vehicle-fixed y-coordinate with suspension jounce/rebound (half-track change)
F_y, F_z	Vehicle-fixed lateral and vertical tire forces at this wheel
h_{c_y}, h_{c_z}	Vehicle-fixed y and z components of the distance from the wheel center to the tire-ground contact point

Both geometric rates come from the vehicle's Camber vs. Jounce/Rebound and Half-track Change vs. Jounce/Rebound data. The jacking force is calculated for independent suspensions only; for a solid axle the equivalent effect is included in the axle equations of motion.

(updated: the jacking force was not documented in earlier editions.)

Suspension stops

The model also includes suspension stops for both jounce (−) and rebound (+) spring deflections (see Figure 4-16). The effect of a suspension stop is to limit suspension travel by increasing significantly the suspension stiffness. A stop engages when the spring deflection passes the corresponding travel limit,

$$\delta_{Stop} = \begin{cases} \delta - \delta_{Rebound}, & \delta > \delta_{Rebound} \\ \delta - \delta_{Jounce}, & \delta < \delta_{Jounce} \\ 0, & \text{otherwise} \end{cases} \qquad \text{(Eq. 115g)}$$

and the suspension force generated at a suspension stop is expressed mathematically as

$$F_{Stop} = \begin{cases} K_1\delta_{Stop} + K_3\delta_{Stop}^3, & \text{for } \delta\cdot\dot{\delta} \geq 0 \\ \eta\left(K_1\delta_{Stop} + K_3\delta_{Stop}^3\right), & \text{for } \delta\cdot\dot{\delta} < 0 \end{cases} \qquad \text{(Eq. 116)}$$

where

Symbol	Definition
K_1	Stop linear rate
K_3	Stop cubic rate
δ_{Stop}	Deformation of suspension stop
$\delta_{Jounce}, \delta_{Rebound}$	Jounce and rebound travel limits
η	Stop energy ratio
$\delta\cdot\dot{\delta}$	(−) if stop deformation is decreasing (used for reducing F_{Stop}; see Eq. 116)

The stop energy ratio makes the stop dissipate energy: while the stop is unloading, only the fraction η of the elastic force is returned. On a **rebound** stop, the remaining fraction, $(1-\eta)F_{Stop}$, is added to the damping force, so the energy removed from the spring appears in the shock absorber. On a **jounce** stop the remaining fraction is simply not applied.

(updated: the treatment of the energy-ratio remainder was not described in earlier editions, and it is not the same for the two stops.)

Reported suspension forces

The spring force reported in the variable output is the sum of the spring and stop forces, $K(\delta - \delta_{Payload}) + F_{Stop}$, rather than the spring force alone. The reported damping force includes the rebound-stop contribution described above.

Suspension limits

SIMON terminates the event and reports a message if a suspension exceeds any of the following limits:

Condition	Limit	Message
Spring deflection	50 in	*Event Termination: Excessive Suspension Deflection!*
Deflection rate	50,000 in/sec	*Event Termination: Excessive Suspension Velocity!*
Spring plus stop force	20 times the vehicle's sprung plus payload weight	*Event Termination: Excessive Suspension Force!*

These limits are not applied to a wheel while it is being contacted by the 3-D mesh collision model, because the forces and deflections during a wheel impact legitimately exceed them. Wheels being contacted by the collision model are instead checked against the wheel damage force and moment limits, which produce the messages *Event Termination: Excessive suspension force from DyMESH wheel impact* and *Event Termination: Excessive suspension moment from DyMESH wheel impact*.

Reaching any of these limits normally means the vehicle or event data are producing an unrealistic response — for example a spring rate, stop rate or terrain feature that is far outside the range the model was built for.

(updated: this subsection describes behavior not documented in earlier editions.)

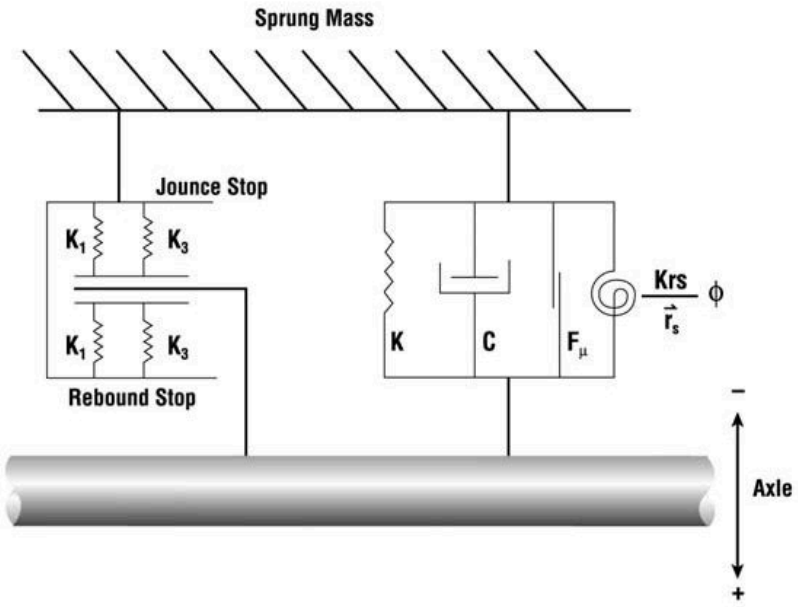


Figure 4-15: Suspension force model.

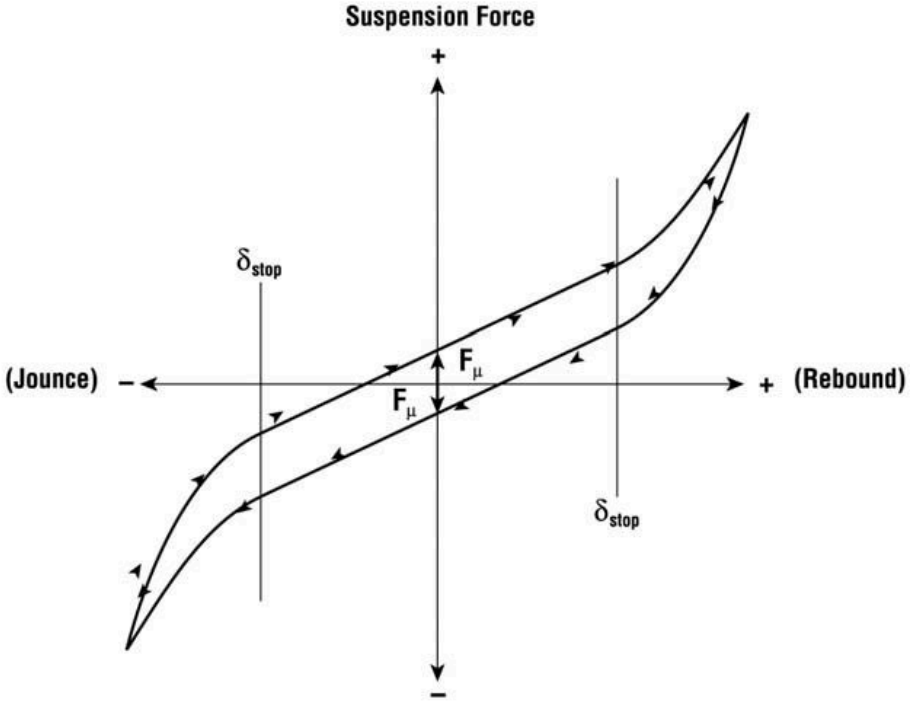


Figure 4-16: Force produced by suspension stops.

Inter-Vehicle Connection Model

In a multi-vehicle train (e.g., a tractor-semitrailer), each sprung mass is allowed six degrees of freedom. The sprung masses are connected by constraint forces and moments acting at the inter-vehicle connection points, as shown in Figure 4-17.

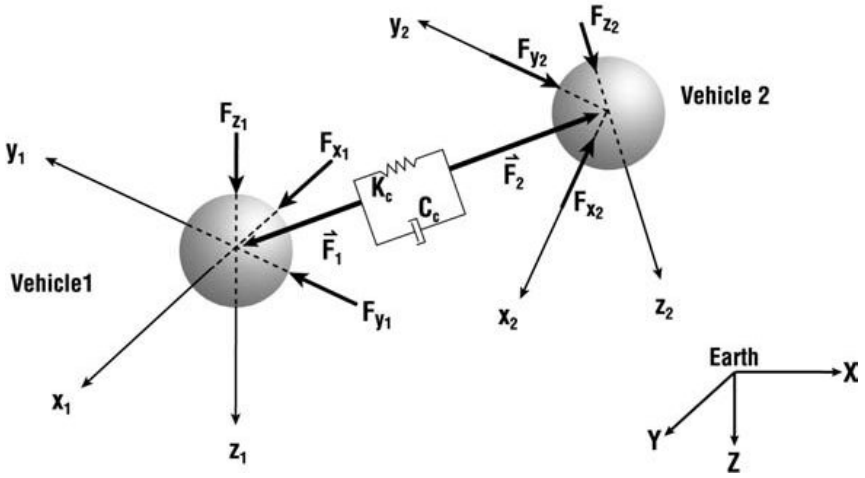


Figure 4-17: Inter-vehicle connection force.

Connection Force

The constraint force is first calculated in the inertial (earth-fixed) reference system, then resolved according to each vehicle-fixed system. In the earth-fixed system,

$$\vec{F}_C = \vec{F}_{C_0} + (\vec{R}_n - \vec{R}_{n+1}) \cdot K_C + (\vec{V}_n - \vec{V}_{n+1}) \cdot C_C \qquad \text{(Eq. 117)}$$

where

Symbol	Definition
$\vec{F}_{C_0}$	Static connection force, the load carried at the connection with the vehicles at rest in their initial positions
$\vec{R}_n - \vec{R}_{n+1}$	Earth-fixed distance between connection points on vehicles n and $n+1$
K_C	Constraint spring rate for vehicle pair. For the default Connection Model (Use Heavier Vehicle) $K_C = \max(m_{n_{Sprung}}, m_{n+1_{Sprung}}) \cdot g$
$\vec{V}_n - \vec{V}_{n+1}$	Earth-fixed relative velocity between connection points on vehicles n and $n+1$
C_C	Constraint damping rate for vehicle pair $= 0.5\sqrt{K_{C_n} \max(m_{n_{Sprung}}, m_{n+1_{Sprung}})}$

(updated: the current Calculation Options dialog adds a Connection Model option that selects which sprung mass governs the constraint rates, and a Connection Failure Start Time that sets the earliest time at which a connection is permitted to fail; see Chapter 2. The three Connection Model settings are: - Use Heavier Vehicle (default) — the constraint weight is the heavier sprung mass, $\max(m_{n_{Sprung}}, m_{n+1_{Sprung}}) \cdot g$*, as shown above. - Use Both Vehicles — a weighted blend of the two sprung masses is used,* $0.75\, m_{Sprung,min} + 0.25\, m_{Sprung,max}$ *(times* g*), for both* K_C *and* C_C*. - Use Tow Veh Properties — the tow vehicle's user-entered rear-connection stiffness and damping values are used directly instead of a mass-derived rate.)*

Calculation of connection earth-fixed position, $\vec{R}$, employs the A matrix defined in Eq. 53,

$$\vec{R} = \begin{bmatrix} R_X \\ R_Y \\ R_Z \end{bmatrix} = \begin{bmatrix} CG_X \\ CG_Y \\ CG_Z \end{bmatrix} + A \cdot \begin{bmatrix} r_x \\ r_y \\ r_z \end{bmatrix} \qquad \text{(Eq. 118)}$$

where

Term	Definition
$[R_X \; R_Y \; R_Z]^T$	Earth-fixed X, Y, Z coordinates of the connection point
$[r_x \; r_y \; r_z]^T$	Vehicle-fixed x, y, z coordinates of connection point

The equations of motion require the earth-fixed connection force to be resolved in the vehicle-fixed reference frame for each vehicle. The vehicle-fixed components of the connection force are

$$\vec{f}_{C_n} = \begin{bmatrix} f_{x_{C_n}} \\ f_{y_{C_n}} \\ f_{z_{C_n}} \end{bmatrix} = -A_n^T \cdot \begin{bmatrix} F_{X_{C_n}} \\ F_{Y_{C_n}} \\ F_{Z_{C_n}} \end{bmatrix} \qquad \text{(Eq. 119)}$$

for the towing vehicle and

$$\vec{f}_{C_{n+1}} = \begin{bmatrix} f_{x_{C_{n+1}}} \\ f_{y_{C_{n+1}}} \\ f_{z_{C_{n+1}}} \end{bmatrix} = A_{n+1}^T \cdot \begin{bmatrix} F_{X_{C_{n+1}}} \\ F_{Y_{C_{n+1}}} \\ F_{Z_{C_{n+1}}} \end{bmatrix} \qquad \text{(Eq. 120)}$$

for the trailing vehicle.

A **pintle eye, hinged** connection cannot carry vertical load: for that connection type the vehicle-fixed z-component of the connection force is set to zero on both vehicles, leaving only the longitudinal and lateral constraint.

(updated: the vertical force exception for hinged pintle connections was not described in earlier editions.)

The total force sums from each connection (front and rear) are

$$\sum \vec{F}_{Conn} = \vec{f}_{C_{Front}} + \vec{f}_{C_{Rear}} \qquad \text{(Eq. 121)}$$

To calculate the velocities, a similar procedure is used. First, the vehicle-fixed velocity components for the connection are required:

$$u_{Conn} = u - r y_{Conn} + q z_{Conn} \qquad \text{(Eq. 122)}$$
$$v_{Conn} = v - p z_{Conn} + r x_{Conn} \qquad \text{(Eq. 123)}$$
$$w_{Conn} = w - q x_{Conn} + p y_{Conn} \qquad \text{(Eq. 124)}$$

These velocities are next transformed to the earth-fixed reference system,

$$\vec{V}_C = \begin{bmatrix} V_{X_C} \\ V_{Y_C} \\ V_{Z_C} \end{bmatrix} = A \cdot \begin{bmatrix} u_{Conn} \\ v_{Conn} \\ w_{Conn} \end{bmatrix} \qquad \text{(Eq. 125)}$$

These earth-fixed velocity components are used in Eq. 117.

Connection Moments

SIMON models roll, pitch and yaw moment transfers between connected vehicles. The roll and pitch moments are the result of a difference in roll or pitch angle between connected vehicles, and the yaw moment results from friction in the connection.

Which transfers occur depends on the connection type:

Transfer	Applies to
Roll moment (Eq. 127)	Fifth wheel connections only
Pitch moment (Eq. 127a)	Dollies with a hinged drawbar only
Yaw friction moment (Eq. 128)	All connection types

A pintle eye or other hinged connection transfers no roll moment, because the joint is free in roll.

(updated: earlier editions gave the roll moment transfer without stating that it applies only to fifth wheel connections, and did not describe the pitch moment transfer at all.)

The roll moment is calculated as follows:

$$\gamma = \psi_n - \psi_{n+1} \qquad \text{(Eq. 126)}$$

$$M_{x_n} = (\phi_{n+1}\cos\gamma - \phi_n)K_{Frame_n} + (\dot{\phi}_{n+1}\cos\gamma - \dot{\phi}_n)C_{Frame_n} \qquad \text{(Eq. 127)}$$

where

Symbol	Definition
γ	Yaw articulation angle between vehicles n and $n+1$
K_{Frame_n}	Frame torsional stiffness for vehicle n (user-entered)
C_{Frame_n}	Frame torsional damping for vehicle n

For a dolly with a hinged drawbar the same treatment is applied about the pitch axis:

$$M_{y_n} = (\theta_{n+1}\cos\gamma - \theta_n)K_{Frame_n} + (\dot{\theta}_{n+1}\cos\gamma - \dot{\theta}_n)C_{Frame_n} \qquad \text{(Eq. 127a)}$$

where θ is the vehicle pitch angle. The resulting moment is distributed to the trailing vehicle's front connection as a roll component $-M_{y_n}\sin\gamma$ and a pitch component $-M_{y_n}\cos\gamma$, mirroring the distribution of the roll transfer in Eqs. 132 and 133.

(updated: the pitch moment transfer was not described in earlier editions.)

The friction moment is

$$M_{Friction_n} = \begin{cases} -\text{sgn}(\dot{\gamma})\mu_{Conn}F_{z_{Conn}}r_{Conn}, & \text{for } |\dot{\gamma}| > \omega \\ [8pt] -\left(\frac{\dot{\gamma}}{\omega}\right)\mu_{Conn}F_{z_{Conn}}r_{Conn}, & \text{for } |\dot{\gamma}| \le \omega \end{cases} \qquad \text{(Eq. 128)}$$

The friction moment opposes the articulation rate, and below the null band it is blended linearly through zero so that it does not chatter as the articulation reverses.

(updated: earlier editions omitted the moment radius from the low-rate branch and gave the moment without the sign that makes it oppose the articulation rate.)

where

Symbol	Definition
μ_{Conn}	Friction coefficient at connection
$F_{z_{Conn}}$	Vehicle-fixed z-component of connection force (see Eqs. 119 and 120)
r_{Conn}	Moment radius for connection
ω	Null band for friction torque at connection

The total moments from the rear connection of vehicle n are

$$M_{x_{Conn_n}} = -f_{y_{C_n}}r_z + f_{z_{C_n}}r_y + M_{Roll_x} \qquad \text{(Eq. 129)}$$

$$M_{y_{Conn_n}} = -f_{z_{C_n}}r_x + f_{x_{C_n}}r_z \qquad \text{(Eq. 130)}$$

$$M_{z_{Conn_n}} = -f_{x_{C_n}}r_y + f_{y_{C_n}}r_x + M_{Friction_n} \qquad \text{(Eq. 131)}$$

And the total moments from the front connection of vehicle $n+1$ are

$$M_{x_{Conn_{n+1}}} = -f_{y_{C_{n+1}}} r_z + f_{z_{Conn_{n+1}}} r_y - M_{Roll_n} \cos\gamma \qquad \text{(Eq. 132)}$$

$$M_{y_{Conn_{n+1}}} = -f_{z_{C_{n+1}}} r_x + f_{x_{C_{n+1}}} r_z - M_{Roll_n} \sin\gamma \qquad \text{(Eq. 133)}$$

$$M_{z_{C_{n+1}}} = -f_{x_{C_{n+1}}} r_y + f_{y_{C_{n+1}}} r_x - M_{Friction_n} \qquad \text{(Eq. 134)}$$

The total connection moments on vehicle n are

$$\sum \vec{M}_C = \vec{M}_{n_{Rear}} + \vec{M}_{n_{Front}} \qquad \text{(Eq. 135)}$$

Articulation Angle Limit

Each rear connection carries a

Maximum Articulation Angle

. If the yaw articulation angle between the two vehicles exceeds it, the event terminates with a message. This limit is a data check rather than a mechanical stop: the connection does not resist reaching it, and the run ends rather than the articulation being restrained.

(updated: the articulation angle limit was not described in earlier editions.)

Connection Failure

A connection may fail during the event if it is overloaded. Failure is tested against both the connection force and the connection moments:

$$|\vec{f}_C| > F_{C,Max} \quad \text{on either vehicle} \qquad \text{(Eq. 135a)}$$

$$|M_x| > M_{C,Max} \ \text{ or } \ |M_y| > M_{C,Max} \ \text{ or } \ |M_z| > M_{C,Max} \quad \text{on either vehicle} \qquad \text{(Eq. 135b)}$$

Two qualifications apply:

- Neither test is made before the **Connection Failure Start Time** set in the Calculation Options dialog. This prevents a connection from failing on the transient of the first few timesteps, before the vehicles have settled.
- A **king pin** connection cannot fail in compression. When the vertical connection force on a king pin is compressive it is excluded from the force magnitude of Eq. 135a, so only tension and shear can pull the connection apart.

When a connection fails, its forces and moments are set to zero and remain zero for the rest of the event; the connection does not re-engage. The moments the connection had already contributed during the failing timestep are also backed out, so the failure does not leave a spurious moment on either sprung mass.

(updated: the connection failure model was not described in earlier editions.)

Dollys

Two types of dollys are supported by SIMON:

- **Converter Dolly** — The fifth wheel articulates about the pitch axis, and the drawbar is rigidly attached to the dolly. The result is that brake torque is resisted at the pintle hook of the tow vehicle.

> **NOTE:** Thus, trailer braking results in a vertical load transfer to the rear of the tow vehicle.

- **Fixed Dolly** — The fifth wheel is fixed to the trailer and is not free to articulate about its pitch axis. The drawbar is hinged.

> **NOTE:** Thus, there is no load transfer to the tow vehicle.

The two dolly types are shown in Figure 4-18.

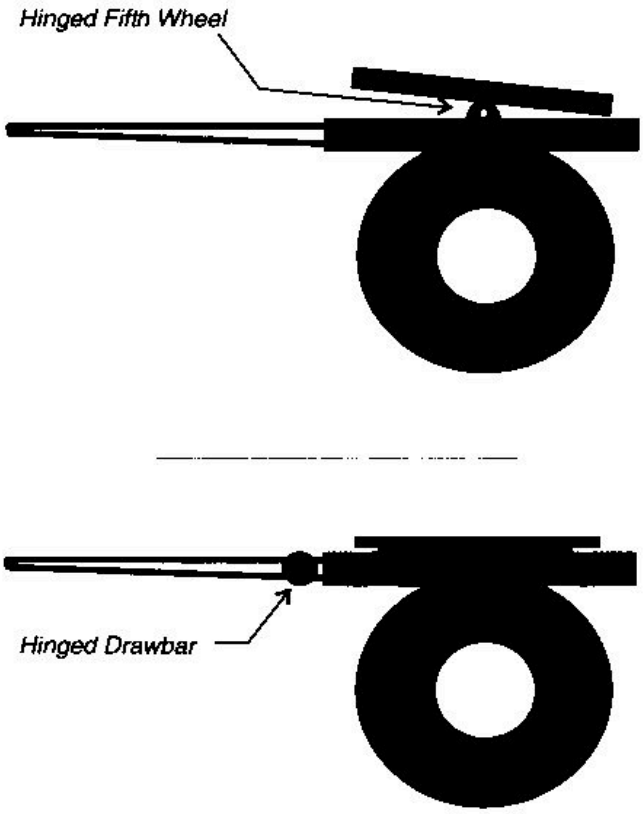


Figure 4-18: Converter (above) and Fixed (below) dollys.

Aerodynamics Model

SIMON includes a lumped parameter aerodynamics model that includes drag on all six vehicle surfaces (front, back, right, left, top and bottom), as well as two additional, user-defined devices, such as a front or rear wing (see Figure 4-19). Air properties (temperature and pressure) are supplied by the Environment Editor, allowing SIMON to study various effects, such as cross-wind, on vehicle handling performance.

Aerodynamic forces and moments produced by each surface act on the sprung mass. To compute these forces, the earth-fixed wind velocity must first be calculated,

$$\vec{V}_{Wind} = \begin{bmatrix} V_{X_{Wind}} \\ V_{Y_{Wind}} \\ V_{Z_{Wind}} \end{bmatrix} = \begin{bmatrix} S_{Wind}\cos(\psi_{Wind} - \pi) \\ S_{Wind}\sin(\psi_{Wind} - \pi) \\ 0 \end{bmatrix} \quad \text{(Eq. 136)}$$

where

Symbol	Definition
S_{Wind}	User-entered wind speed
ψ_{Wind}	User-entered wind direction relative to earth-fixed X-axis

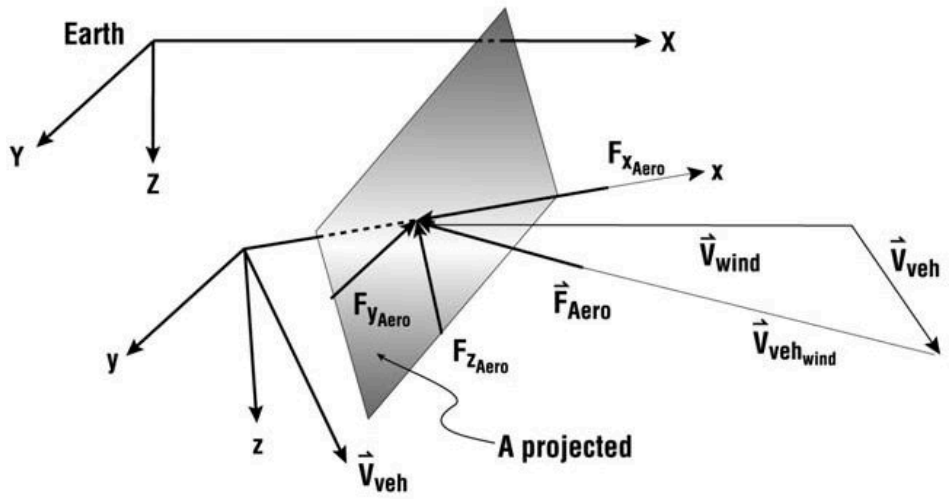


Figure 4-19: Aerodynamic forces and moments.

The earth-fixed components of the vehicle velocity vector are

$$\vec{V}_{Veh} = \begin{bmatrix} V_{X_{Veh}} \\ V_{Y_{Veh}} \\ V_{Z_{Veh}} \end{bmatrix} = A \cdot \begin{bmatrix} u \\ v \\ w \end{bmatrix} \qquad \text{(Eq. 137)}$$

Next, calculate the earth-fixed components of the total relative wind velocity acting on the vehicle,

$$\vec{V}' = \begin{bmatrix} V'_{X_{Wind}} \\ V'_{Y_{Wind}} \\ V'_{Z_{Wind}} \end{bmatrix} = \vec{V}_{Wind} - \vec{V}_{Veh} = \begin{bmatrix} V_{X_{Wind}} \\ V_{Y_{Wind}} \\ V_{Z_{Wind}} \end{bmatrix} - \begin{bmatrix} V_{X_{Veh}} \\ V_{Y_{Veh}} \\ V_{Z_{Veh}} \end{bmatrix} \qquad \text{(Eq. 138)}$$

Finally, the vehicle-fixed components of the total relative wind velocity are computed:

$$\vec{V}_{VehWind} = \begin{bmatrix} V_{x_{Wind}} \\ V_{y_{Wind}} \\ V_{z_{Wind}} \end{bmatrix} = A^T \begin{bmatrix} V'_{X_{Wind}} \\ V'_{Y_{Wind}} \\ V'_{Z_{Wind}} \end{bmatrix} \qquad \text{(Eq. 139)}$$

SIMON uses HVE's aerodynamics model, which allows up to eight individual surfaces to produce aerodynamic forces and moments on the sprung mass: the six vehicle surfaces (front, right, back, left, top and bottom) plus two user-defined devices.

*(updated: the aerodynamics model has been extended since the Fifth Edition manual. The printed manual described a single drag coefficient per surface, producing one force component along each vehicle axis. Each surface now carries three independent coefficients — **Drag**, **Lift** and **Side** — entered in the Aerodynamic Drag dialog, and each surface develops force in all three vehicle-fixed directions. In addition, a surface now generates force only while the relative wind is actually striking it.)*

Each surface in use has three aerodynamic constants, formed from its coefficients and projected area:

$$C_D = \frac{C_d A \rho}{2g} \qquad C_L = \frac{C_l A \rho}{2g} \qquad C_S = \frac{C_s A \rho}{2g} \qquad \text{(Eq. 140)}$$

where

Symbol	Definition
C_d, C_l, C_s	Drag, Lift and Side coefficients for the surface
A	Projected surface area
ρ	Air density $= \frac{P_{Barometric}}{GasConst \cdot T_{Absolute}}$
g	Gravitational constant

Directional loading

A surface is loaded only when the relative wind strikes its outward face. For each surface, the driving term is the signed square of the relevant relative-wind component, which is zero whenever the wind is blowing on the opposite face:

$$q^{-} = \begin{cases} -V^2 & V < 0 \\ 0 & V \geq 0 \end{cases} \qquad q^{+} = \begin{cases} +V^2 & V > 0 \\ 0 & V \leq 0 \end{cases} \qquad \text{(Eq. 141)}$$

The front, right, bottom and front-device surfaces use q^{-}; the back, left, top and back-device surfaces use q^{+}. Each uses the relative-wind component along its own normal axis — $V_{x_{Wind}}$ for the front, back and both devices, $V_{y_{Wind}}$ for the right and left, and $V_{z_{Wind}}$ for the top and bottom.

This means, for example, that a vehicle travelling forward loads its front surface but not its back surface, and a cross-wind from the right loads the right surface only.

Force on each surface

The drag constant always acts along the surface's own normal axis; the lift and side constants produce the cross-axis components:

Surface	f_x	f_y	f_z	Driving term
Front	$C_D\, q$	$-C_S\, q$	$-C_L\, q$	q^{-} on $V_{x_{Wind}}$
Right	$-C_S\, q$	$C_D\, q$	$-C_L\, q$	q^{-} on $V_{y_{Wind}}$
Back	$C_D\, q$	$C_S\, q$	$C_L\, q$	q^{+} on $V_{x_{Wind}}$
Left	$C_S\, q$	$C_D\, q$	$C_L\, q$	q^{+} on $V_{y_{Wind}}$
Top	$C_L\, q$	$C_S\, q$	$C_D\, q$	q^{+} on $V_{z_{Wind}}$
Bottom	$-C_L\, q$	$-C_S\, q$	$C_D\, q$	q^{-} on $V_{z_{Wind}}$
Front device	$C_D\, q$	$-C_S\, q$	$-C_L\, q$	q^{-} on $V_{x_{Wind}}$
Back device	$C_D\, q$	$C_S\, q$	$C_L\, q$	q^{+} on $V_{x_{Wind}}$

Because the force varies with the square of the relative wind speed, aerodynamic effects grow rapidly with speed and become significant in high-speed handling, cross-wind and downforce studies.

> **NOTE:** By default, the front surface is normally assigned aerodynamic properties. Assignment of aerodynamic properties for other surfaces is left up to the user. A surface takes part in the calculation only if it has been given aerodynamic data.

Finally, the sum of the vehicle-fixed aerodynamic forces and moments is computed,

$$\vec{F}_{Aero} = \begin{bmatrix} F_{x_{Aero}} \\ F_{y_{Aero}} \\ F_{z_{Aero}} \end{bmatrix} = \begin{bmatrix} \sum_{i=1}^{N} f_{x_{Aero}} \\ \sum_{i=1}^{N} f_{y_{Aero}} \\ \sum_{i=1}^{N} f_{z_{Aero}} \end{bmatrix} \qquad \text{(Eq. 142)}$$

and

$$\vec{M}_{Aero} = \begin{bmatrix} M_{x_{Aero}} \\ M_{y_{Aero}} \\ M_{z_{Aero}} \end{bmatrix} = \begin{bmatrix} \sum_{i=1}^{N} (f_{z_{Aero}}\, r_{y_{CP}} - f_{y_{Aero}}\, r_{z_{CP}}) \\ \sum_{i=1}^{N} (f_{x_{Aero}}\, r_{z_{CP}} - f_{z_{Aero}}\, r_{x_{CP}}) \\ \sum_{i=1}^{N} (f_{y_{Aero}}\, r_{x_{CP}} - f_{x_{Aero}}\, r_{y_{CP}}) \end{bmatrix} \qquad \text{(Eq. 143)}$$

(updated: the printed manual's third row of this moment equation contained a typographical error, using the z-coordinate of the center of pressure in place of the x-coordinate. The yaw moment is the standard cross product of the center-of-pressure position with the surface force, as shown above.)

where

Symbol	Definition
N	Number of aerodynamic surfaces or devices in use
r_{CP}	Center of pressure for surface, i

These forces and moments are included in the dynamics engine (see Eqs. 3 and 7).

Collision Pulse Table

SIMON allows a vehicle's motion during part of an event to be prescribed from a user-entered table rather than computed from the physics. This is used to study the effect of forces the model does not itself produce, or to drive a vehicle along a measured trajectory. The table is entered from Set-up, Collision Pulse for the selected vehicle, and one of three table types may be chosen.

Force and moment table

The table gives the vehicle-fixed force and moment components as a function of time. They are interpolated at the current time and added to the sprung mass force and moment sums of Eqs. 3 and 7. The forces act at a user-specified vehicle-fixed location, so they also produce a moment about the CG:

$$\vec{M}_{Pulse} = \vec{M}_{Table} + \vec{r}_{Pulse} \times \vec{F}_{Table} \qquad \text{(Eq. 11a)}$$

where $\vec{r}_{Pulse}$ is the vehicle-fixed location at which the pulse force is applied. This is the table type described in earlier editions of this manual, and it is the only one that leaves the vehicle's equations of motion intact — the prescribed force is simply one more external force.

Velocity table

The table gives the vehicle's six velocity components as a function of time. Rather than being added as a force, the interpolated values are written directly into the vehicle's velocity states and their accelerations are taken from the slope of the table. The vehicle therefore follows the tabulated velocities exactly, whatever forces are acting on it.

A value is only applied if it is within range: a linear velocity component larger than 3500 in/sec, or an angular velocity component larger than 12.56 rad/sec, is ignored and the corresponding state is left to the physics.

Position table

The table gives the vehicle's earth-fixed position and orientation as a function of time. The interpolated values are written directly into the vehicle's position and Euler angle states, and its velocity states are taken from the slope of the table. The vehicle follows the tabulated path exactly.

> **NOTE:** The velocity and position table types override the vehicle's equations of motion for as long as the table runs. The forces the vehicle would have needed to follow the prescribed motion are not calculated, so quantities that depend on them — tire forces, suspension deflections and the Accident History delta-V — should not be interpreted as physical results while a prescribed table is in effect.

(updated: earlier editions described only the force and moment table type. The velocity and position table types were added since.)

Sprung Mass Impact Model

SIMON uses HVE's DyMESH collision model to compute 3-dimensional forces and moments resulting from inter-vehicle collision. Simultaneous collision forces between any number of vehicles are allowed. The DyMESH collision model is described in references 7 and 8. In general, vehicle-fixed force components are calculated for each vertex. These forces are summed and the resulting summed forces and moments acting on the sprung mass are supplied to the dynamics engine (see Eqs. 3 and 7).

(updated: the current DyMESH implementation includes several capabilities beyond those described in the Fifth Edition manual:)

- *Two selectable algorithm versions (Version 3 and Version 4) and two mesh smoothing models, selected in the DyMESH Options dialog (see Chapter 2).*
- *Vehicle-environment collision detection (Include Environment) with independent vehicle and environment start times.*
- *Tow vehicle/trailer mesh contact for articulated vehicles.*
- *DyMESH wheel contact and wheel damage. Each wheel can carry its own collision mesh. Collision forces acting on a wheel mesh are added to the sprung mass, and forces exceeding the wheel's user-entered damage thresholds (Max No-Damage Force/Moment, Deformation Rate and Max Force/Moment — see the Event Editor's Vehicle Wheels dialog) produce permanent wheel displacement and reorientation, which in turn affect the wheel kinematics of Eqs. 58–62.*
- *Wheel displacement capping. The collision-induced x-y displacement of a wheel during a timestep is tracked against the maximum contact movement of the wheel mesh vertices for that timestep; the computed force-based displacement is capped so it can never exceed the actual contact motion, and no displacement is produced when there is no mesh contact movement. This prevents overprediction of wheel set-back or track-width change in severe impacts.*

Primary and secondary vehicles

DyMESH evaluates a collision between a pair of objects, treating one as the primary and the other as the secondary. The two roles are not symmetric, so SIMON alternates them from timestep to timestep, and averages the resulting forces and moments over successive timesteps so that both treatments contribute to every step:

$$\vec{F}_{Collision} = \frac{1}{2}(\vec{F}_{Current} + \vec{F}_{Previous}), \qquad \vec{M}_{Collision} = \frac{1}{2}(\vec{M}_{Current} + \vec{M}_{Previous}) \qquad \text{(Eq. 144)}$$

The averaging history is kept separately for each pair of colliding objects, so a vehicle involved in two simultaneous collisions does not mix the two. Wheel collision forces and moments are averaged in the same way.

Alternation is suspended where the roles are not interchangeable: a fixed barrier or the environment is always the primary, because neither deforms.

(updated: primary/secondary alternation and collision force smoothing were not described in earlier editions. The smoothing model is selectable in the DyMESH Options dialog; see Chapter 2.)

Collision integration timestep

Collision forces rise and fall far more quickly than the forces governing ordinary trajectory motion, so SIMON integrates the collision phase with a separate, shorter timestep. At each step the model tests whether any pair of objects is in contact; if so, the *Collision Integration Timestep* is used for that step, and otherwise the *Vehicle Trajectory Integration Timestep* is used. Both are set in the Simulation Controls dialog.

The switch is made for the whole event, not per vehicle: if any pair anywhere in the event is in contact, every vehicle is integrated at the collision timestep for that step.

(updated: the separate collision integration timestep was not described in earlier editions.)

Impact and separation

The Accident History report records the position and velocity of each vehicle at impact and at separation. Impact is detected when the collision model first reports contact between the pair. Separation is detected when contact between the pair is lost.

If the **Accident History Basis** calculation option is set to *Use Impact Acceleration*, separation is additionally detected when the total acceleration of both vehicles falls below the *Collision Acceleration Threshold*, whichever occurs first:

$$\sqrt{a_x^2 + a_y^2 + a_z^2} < a_{Threshold} \quad \text{for both vehicles} \qquad \text{(Eq. 145)}$$

(updated: earlier descriptions of this option stated that the choice governs both impact and separation. Impact is detected from collision contact regardless of the setting; the acceleration threshold applies only to separation, and only as an additional condition alongside loss of contact.)

Velocity change

Two different velocity changes are reported, computed in different ways.

The **Accident History delta-V** is the magnitude of the vehicle's change in earth-fixed velocity between the impact and separation states:

$$\Delta V = \sqrt{(V_{X,Sep} - V_{X,Imp})^2 + (V_{Y,Sep} - V_{Y,Imp})^2} \qquad \text{(Eq. 146)}$$

where each earth-fixed velocity component is formed from the vehicle's forward and lateral velocity and its heading at that instant. This is a planar quantity: the vertical velocity change is not included.

The **collision pulse delta-V**, reported for each impulse in the Damage Data report, is instead obtained by integrating the vehicle's acceleration over the pulse using the trapezoidal rule:

$$\Delta\vec{V}_{Pulse} \mathrel{+}= \frac{1}{2}(\vec{a} + \vec{a}_{Prev})\Delta t \qquad \text{(Eq. 147)}$$

with the reported value being the largest magnitude reached before separation. This one is a full 3-D quantity.

(updated: neither velocity change calculation was described in earlier editions. The two are not interchangeable, and they will not agree exactly.)

Collision pulses

A vehicle may experience several distinct impulses during an event — a sideswipe followed by a secondary contact, for example. SIMON identifies each as a separate collision pulse, up to a maximum of ten per vehicle. Exceeding that limit terminates the event.

A pulse begins at the first timestep at which the collision model reports force on the vehicle. It is closed only once contact has been absent continuously for 0.025 seconds: on every timestep that contact persists, a provisional closing deadline is set that far ahead, so the deadline rolls forward and the pulse survives any interruption shorter than that. Contact that resumes before the deadline rejoins the pulse already in progress against that partner rather than opening a new one. The intent is that the intermittent contact typical of two vehicles disengaging is reported as one collision rather than a string of small ones.

> **NOTE:** A distinct secondary contact separated from the previous one by less than 0.025 seconds is absorbed into the earlier pulse — its impulse contributes to that pulse's PDOF, its acceleration to the pulse's delta-V, and the vertices it touches to the pulse's CDC — with nothing in the report to indicate that a separate contact occurred. Where a secondary contact is suspected, examine the collision force in the Variable Output, Kinetics group rather than relying on the number of pulses reported.

While a pulse is in progress SIMON records:

- the peak total collision force and the time it occurred;
- the peak total acceleration;
- the principal direction of force, computed from the direction of the accumulated collision impulse as an azimuth and a zenith angle, together with the equivalent clock direction;
- the impulse center, the force-weighted average position of every mesh vertex carrying force during the pulse;
- the collision deformation classification surface, taken from the surface in contact at the start of the pulse.

The principal direction of force and the delta-V are both frozen once separation is declared, so neither is affected by residual contact afterwards.

*(updated: the collision pulse data were not described in earlier editions. Note in particular that the PDOF reported for a pulse is the direction of the impulse accumulated over the whole pulse, **not** the direction of the peak collision force; an earlier edition of this section stated the latter.)*

Software Implementation

The above physics model and component modules are implemented in a single HVE-compatible physics model, named SIMON.

SIMON is programmed using the C programming language. It is a modular program and includes several C functions to perform specific tasks. A general flow chart for SIMON is shown in Figure 4-20.

The equations of motion are advanced with a fixed-step Runge-Kutta integrator. The step size is the Vehicle Trajectory Integration Timestep, or the Collision Integration Timestep while any pair of objects is in contact (see

Sprung Mass Impact Model — Collision integration timestep

). The wheel spin and steer degrees of freedom are advanced separately at the tire force timestep.

(updated: the integration method was not stated in earlier editions.)

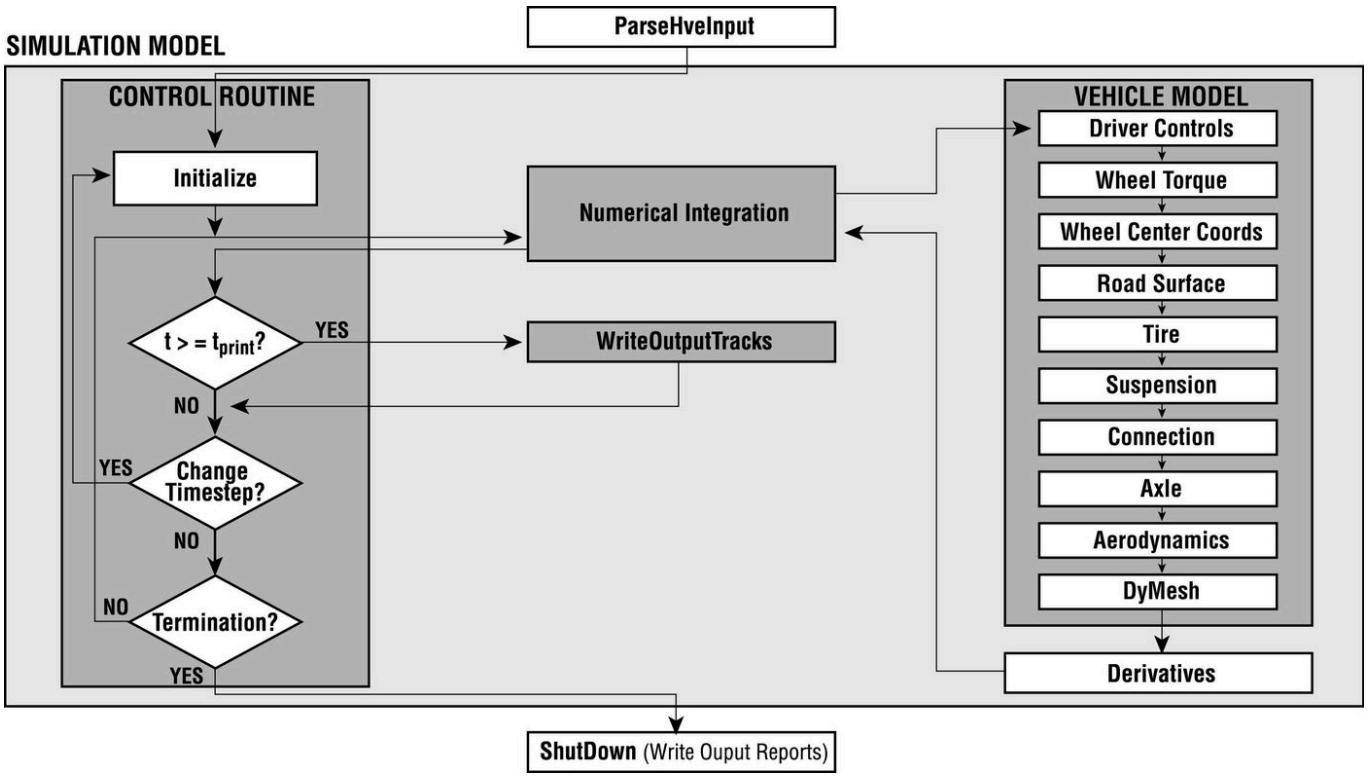


Figure 4-20: Flowchart for SIMON main calculation procedures.

Chapter 5 — SIMON Tutorial

Description

The SIMON Tutorial investigates the effects of an ABS braking system on a combined braking and steering maneuver. The example uses an ISO 3888 lane-change maneuver to illustrate how ABS brake systems improve vehicle controllability during severe (limit) maneuvers.

Like all SIMON events, the procedures used in the tutorial involve the following basic steps:

- Creating the vehicle(s)
- Creating the environment
- Executing the SIMON event(s)
- Reviewing the SIMON output reports

This basic procedure is described in detail in this tutorial.

> **NOTE:** It is assumed that HVE is up and running, and that the user is familiar with HVE's basic features, such as using HVE's dialogs, viewers and editors. The purpose of this tutorial is to illustrate those features while setting up and executing a SIMON event.
>
> **NOTE:** If you are new to HVE and are not familiar with HVE's basic features, you should perform the HVE Tutorial (see HVE User's Manual, Chapter 32) before proceeding with the SIMON Tutorial.

Getting Started

As in other tutorials, before we get started, let's set the user options so we're all starting on the same page.

> **NOTE:** Most options simply affect the appearance in a viewer during Event or Playback mode. However, some options affect the data used in the analysis. For example, if AutoPosition is On, the vehicle position conforms to the local surface; otherwise, the position is set by the Position/Velocity dialog. Obviously, the resulting difference in initial conditions could substantially change the event.
>
> **NOTE:** Some of the following options are "toggles" that switch between two different modes. Make sure these options are set correctly.

To set the initial user options, choose the following from the Options Menu:

- Show Key Results
- Hide Axes
- Hide Contacts
- Hide Velocity Vectors
- Show Skidmarks
- Show Targets (this is required for our HVE Driver Model)
- AutoPosition On
- Units equals US
- Render Options:
- Show Humans as Simple
- Show Vehicles as Actual
- Phong Shading Mode
- Complexity equals Object
- Render Quality equals 5
- Texture Quality equals 1
- Anti-aliasing equals 1
- Playback Options:
- Playback Interval equals 0.01 seconds

The remaining options will automatically initialize to their default conditions. We're now ready to proceed with the tutorial.

Our goal is to use SIMON to simulate an ISO 3888 lane-change maneuver with and without anti-lock braking (ABS). This tutorial describes how to perform these simulations.

Creating the Vehicle

First let's add the vehicle to our case. The vehicle is a 2001 Cadillac Catera 4-Dr Sedan. It is fitted with a 4-wheel disc brake system that includes ABS at each wheel. Let's load the vehicle:

1. If the Vehicle Editor is not the current editor, choose Vehicle Mode. The Vehicle Editor is displayed.
2. Click Add New Object. The Vehicle Information dialog is displayed. The Vehicle Information dialog allows the user to select the basic vehicle attributes according to Type, Make, Model, Year and Body Style.

> **NOTE:** The Vehicle Information dialog also allows you to edit the Driver Location, Engine Location, Number of Axles and Drive Axle(s).

1. Using the option buttons, choose the following vehicle from the Vehicle Database:
2. Type = Passenger Car
3. Make = Cadillac
4. Model = Catera
5. Year = 1997–2001
6. Body Style = 4-Door
7. Source Database = Tutorial.db
8. Drive Axle = Axle 2
9. Click OK to add the Cadillac Catera to the Active Vehicles list.

The Cadillac Catera is displayed in the viewer, as shown in Figure 5-1.

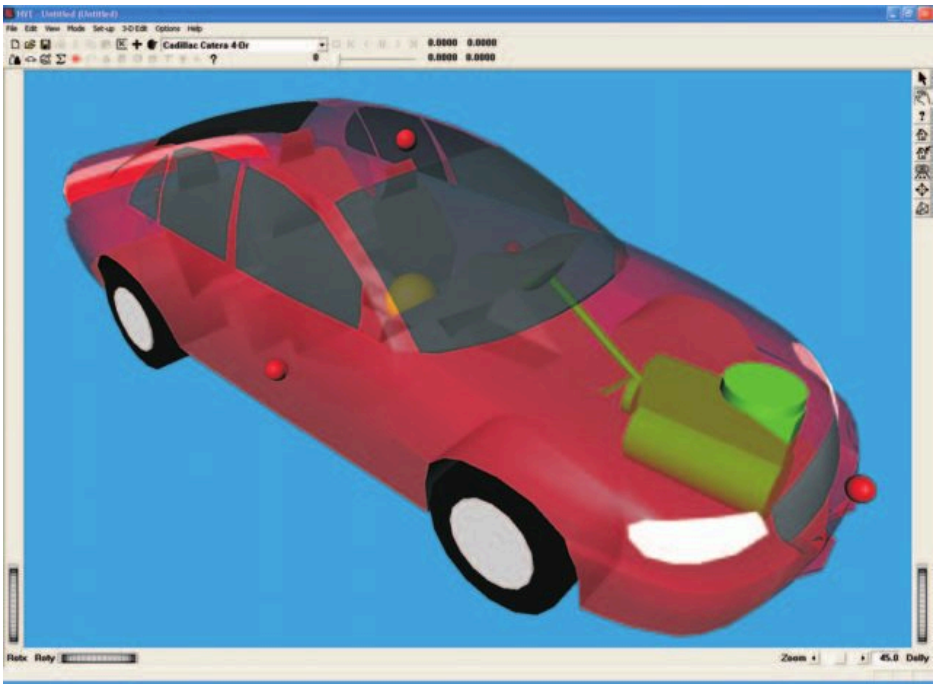


Figure 5-1: 2001 Cadillac Catera used in the ISO 3888 Lane-Change Maneuver.

Editing the Vehicle

Like all vehicles in the EDC Vehicle Database, the Cadillac Catera comes out of the database just as it comes off the showroom floor. However, there is one feature we need to check: the ABS system.

> **NOTE:** In the current version of the EDC Vehicle Database, if a vehicle had the factory option for ABS, ABS is enabled by default.

To check to see if the ABS system is enabled, perform the following steps:

1. Click on the vehicle's brake system icon (i.e., the brake pedal — you may need to look at the vehicle's undercarriage to locate the brake pedal). The Brake System dialog is displayed (see Figure 5-2).
2. Ensure the ABS Installed checkbox is marked to activate the vehicle's ABS system.
3. Press OK to remove the Brake System Data dialog.

The Cadillac Catera is now ready for our ISO lane-change maneuver.

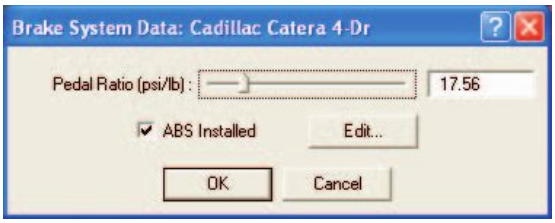


Figure 5-2: Brake System Data dialog, with the ABS Installed checkbox marked. Note that the Edit button becomes enabled after clicking the checkbox.

Creating the Environment

Now, let's add the environment:

1. Choose Environment Mode. The Environment Editor is displayed.
2. Click on Add New Object. The Environment Information dialog is displayed.
3. Using the Location Database combo box, choose Detroit, Michigan, USA. The latitude (42.23.00N), longitude (85.05.00W) and GMT hours from the prime meridian (−5.00) are displayed for the selected location.
4. Edit the date and time of the experimental study: 10/02/2001 and 1500, respectively.
5. Edit the angle from true north to the earth-fixed X axis in our environment: 165 degrees.

> **NOTE:** The Latitude, Longitude, GMT, Date/Time and angle from true north are used to position the sun in the scene. This is, of course, important because the sun is the primary light source for the scene.

1. Edit the default environment name; enter *ISO 3888 Lane-Change Course*.
2. To add the environment geometry file to our case, click on Open. The Environment Geometry File browser is displayed.
3. Click on the Files of Type option list and choose files of type HVE Geometry Files (*.h3d). A list of environment geometry files using the HVE file format is displayed in the file browser. Double-click on **IsoLaneChangeb.h3d** to choose the environment file and remove the dialog.
4. Press OK.

The selected environment is added to our case and displayed in the Environment Viewer (see Figure 5-3). Manipulate the viewer to view the scene.

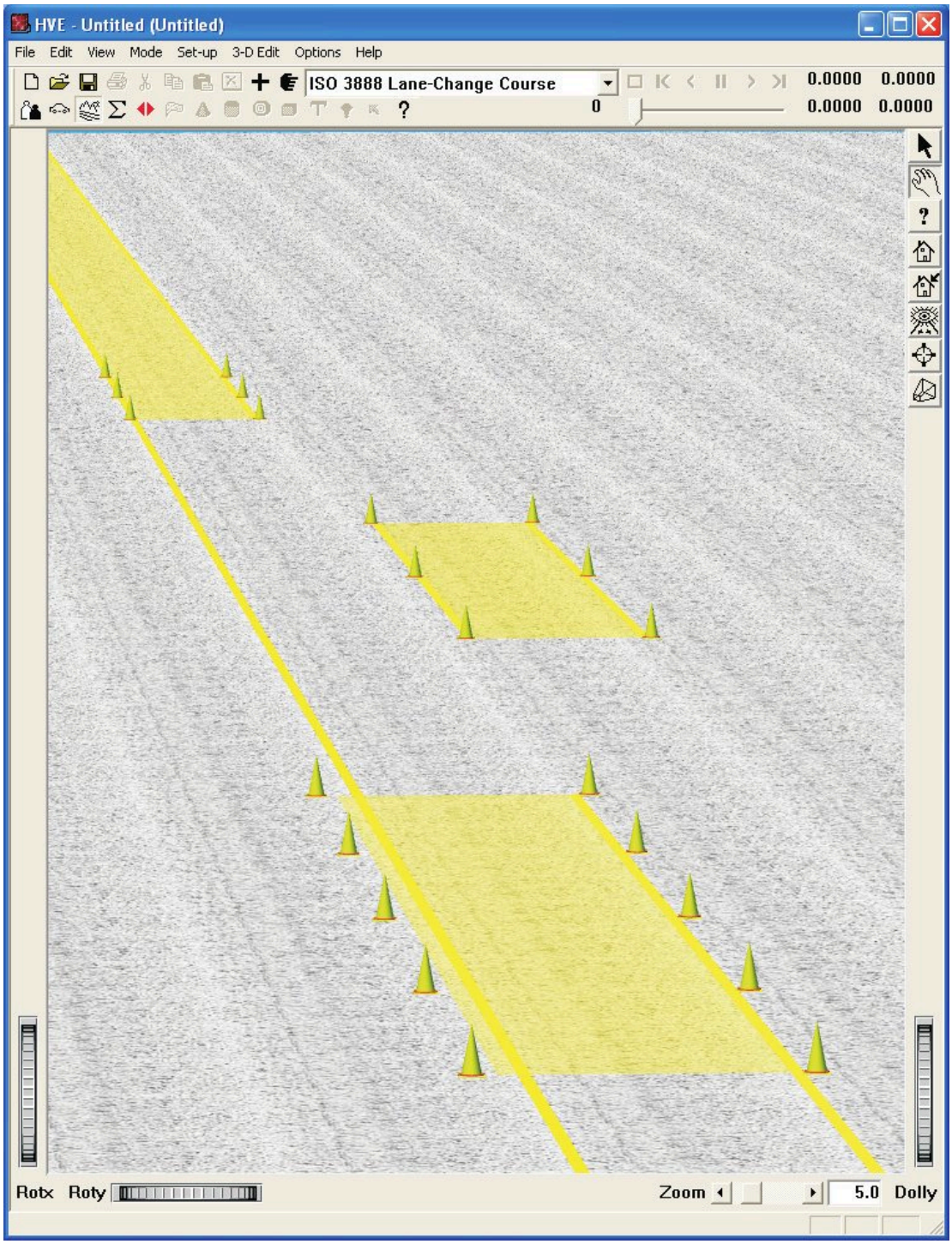


Figure 5-3: Environment for ISO 3888 Lane-Change Maneuver.

Saving the Case

Now that we've created all the objects (vehicle and environment) for our case, let's save the case file.

1. Click on the File menu and choose Save. The Save-as File Selection dialog is displayed.

> **NOTE:** The Save-as dialog is displayed because the case has not been saved previously, so we need to enter a filename.

1. In the Case Title text field, replace *Untitled* with **SIMON Tutorial Case**.

> **NOTE:** The Case Title is displayed as a heading on all printed output reports.

1. Place the mouse cursor in the Filename text field and enter **SimonTutorial**.
2. Click SAVE. The current case data are saved in the `hve/supportFiles/case` subdirectory.

> **NOTE:** Saving the file occasionally is a highly recommended practice.

Creating the Events

Our SIMON Tutorial will include several events. They are both ISO 3888 lane-change maneuvers. The events are identical, with one exception: the first event uses ABS, while the second has ABS disabled. Let's proceed by creating, setting up and executing our first event.

ISO Lane-Change with ABS

To create the first event, perform the following steps:

1. Choose Event Mode. The Event Editor is displayed.
2. Click on Add New Object. The Event Information dialog is displayed.
3. Select Cadillac Catera from the Active Vehicles list.
4. Select SIMON from the Calculation Method options list.
5. Enter a name for the event: **ISO Lane-Change, w/ABS**.

> **NOTE:** HVE will append the name of the calculation method to the event name, thus the complete event name will become "SIMON, ISO Lane-Change, w/ABS."

1. Press OK to display the Event Editor.

Now we're ready to set up the first event:

1. Choose Set-up from the menu bar and select Position/Velocity. The Cadillac Catera is displayed at the earth-fixed origin. The Position/Velocity dialog displays the position and velocity for the Initial path position (see Figure 5-4).
2. If the Cadillac is not visible in the viewer, manipulate the viewer until the Cadillac becomes visible.
3. Using the Position/Velocity dialog, enter the vehicle's initial position: X = −80.0 ft, Y = 0.0 ft. Its initial heading angle is 0.0 degrees and need not be modified.

> **NOTE:** Adjust the viewer by dollying back (using the Dolly thumb wheel) until you can see enough of the entire scene.

1. Click the Velocity Is Assigned checkbox. Enter the initial velocity: 55 mph.

> **NOTE:** Remember to press Apply or \<Enter> after entering a value; otherwise the value is not assigned!

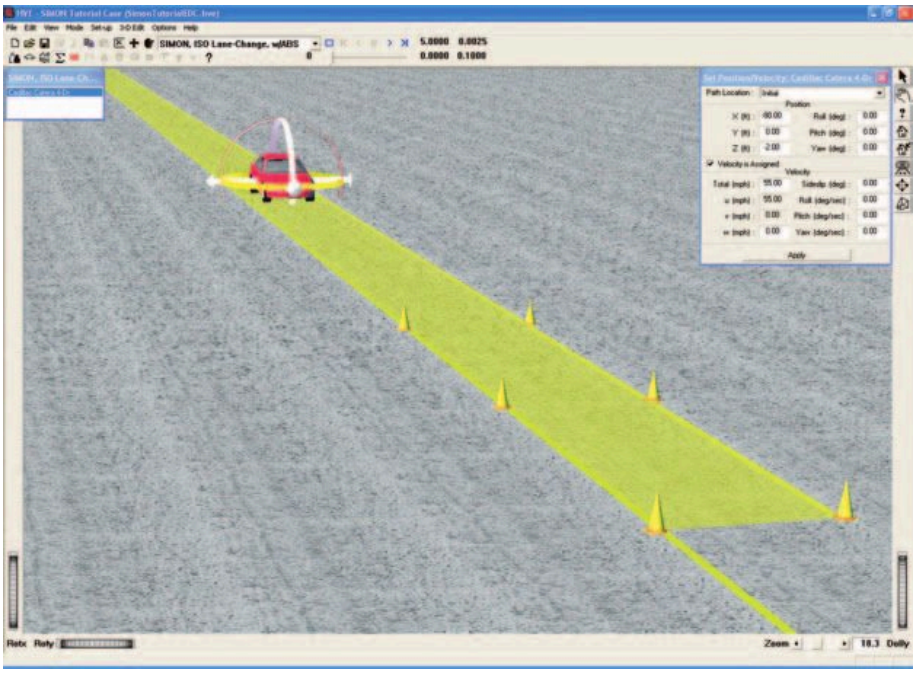


Figure 5-4: Event Editor, setting up our first SIMON event. The Position/Velocity dialog is used for assigning the Cadillac's initial position and velocity.

Steering Input

The vehicle initial conditions are now established. Let's enter the driver controls.

Because the ISO lane-change maneuver has a pre-defined path, our tutorial is a perfect application for the HVE Driver Model [4]. By defining the same driver model and path for both events, our experiment has only one variable (i.e., with and without ABS). To set up the HVE Driver Model, perform the following steps:

1. Click on the Set-up Menu and select Driver Controls. The Driver Controls dialog is displayed with an empty steer table.
2. Click on the HVE Driver tab. The HVE Driver Model (Path Follower) dialog is displayed.
3. Click the Use Path Follower checkbox.
4. Click on the Driver Data tab. The Driver Data page is displayed.
5. Increase Driver Comfort Level to 1.0 g.
6. Press OK to dismiss the Driver Controls dialog.

We now need to assign the target positions required by the HVE Driver Model. Use the Position/Velocity dialog to assign the target positions shown in Table 5-1.

Table 5-1 Position and heading angle for each target position.

Path Position	X-coord (ft)	Y-coord (ft)	Heading (deg)
Initial	−80.0	0.0	0.0
Begin Perception	50.0	0.0	0.0
Begin Braking	148.0	−11.5	0.0
Impact	325.0	−11.5	0.0

To assign these target positions, perform the following steps:

> **NOTE:** The Initial position has already been entered.

1. Click on the Path Location list and choose Begin Perception. Then enter the data shown in Table 5-1 for the Begin Perception position.
2. Click on the Path Location list and choose Begin Braking. Then enter the data shown in Table 5-1 for the Begin Braking position.
3. Click on the Path Location list and choose Impact. Then enter the data shown in Table 5-1 for the Impact position.

> **NOTE:** The actual names of these target positions (e.g., Impact) are irrelevant to the HVE Driver Model.
>
> **NOTE:** Remember to press Apply or \<Enter> after entering each position; otherwise it is not assigned!

After assigning all the target positions, the Event Editor should appear as shown in Figure 5-5.

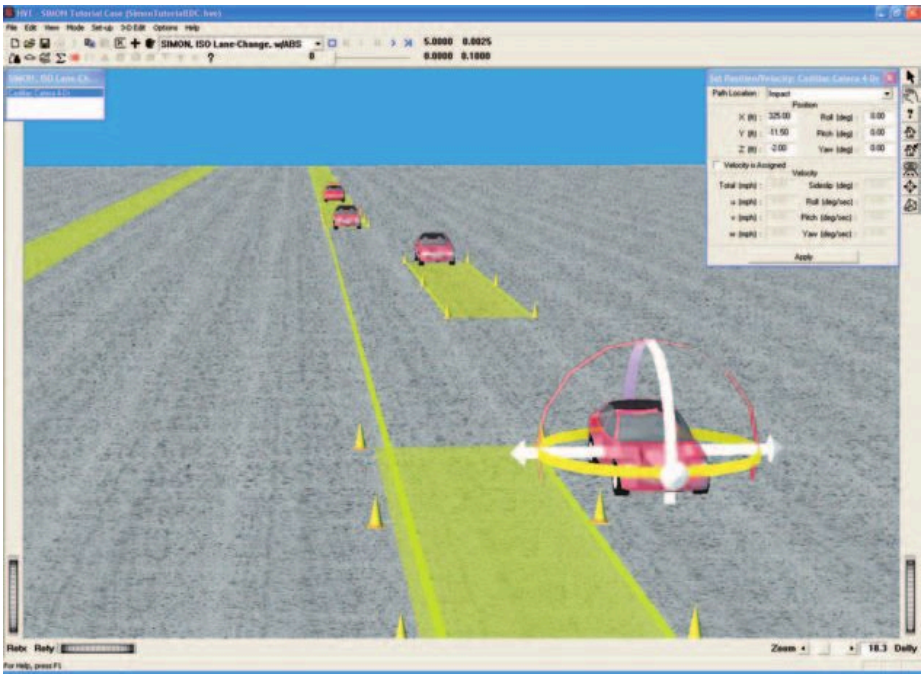


Figure 5-5: Event Editor, after assigning the target positions used by the HVE Driver Model.

Brake Input

The brake input for our maneuver is very simple: we're simply going to apply a very large pedal force (say 120 lb) at the start of the steering maneuver. To enter the brake table, perform the following steps:

1. Click on the Set-up menu and select Driver Controls. The Driver Controls dialog is displayed.
2. Click on the Driver Controls dialog's Brake tab. The Brake Pedal Force table is displayed.
3. Enter the times and pedal forces as shown in Table 5-2.
4. Press OK to accept the brake table.

Table 5-2 Brake Table.

Time (sec)	Pedal Force (lb)
1.5	0.0
1.6	120.0

The brake table is now ready for our event.

Let's make one more (rather subtle) change before we execute the event. Because the ABS system will modulate the brake pressure at a relatively high frequency, we should reduce the simulation output time interval to 0.01 seconds from its default value of 0.10 seconds. This change will ensure that we capture the detail in the wheel brake pressure modulation.

1. Click on the Options menu and select Simulation Controls. The Simulation Controls dialog is displayed.
2. Edit the default Output Time Interval, changing it to 0.01 seconds.
3. Press OK to accept the new value.

> **NOTE:** The smaller output interval will increase the execution time.

Now, we're ready to execute the event.

- Using the Event Controller, click Play to execute the event.

Note that the vehicle successfully negotiates the lane change maneuver, even with a heavy brake application. The vehicle comes to a stop after travelling approximately 120 ft. The average deceleration is 0.80 g. Figure 5-6 shows the vehicle at its rest position. Note also the faint tire marks caused by tire scuffing.

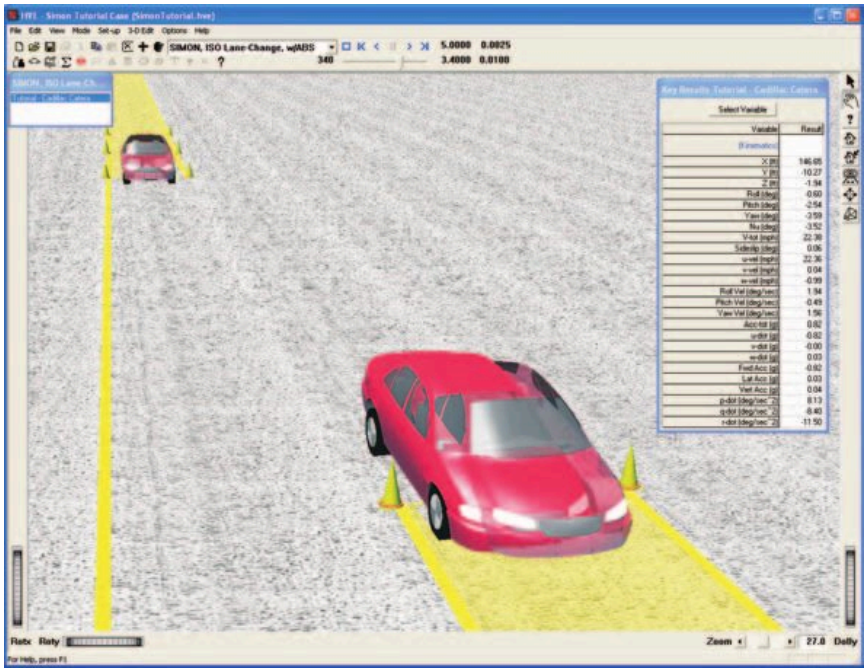


Figure 5-6: ISO Lane-Change Maneuver, w/ABS.

> **NOTE:** If you remove the pavement texture, you can see a faint tire mark!

ISO Lane-Change without ABS

We have now completed the first SIMON event. Let's repeat the test with the ABS disabled. To disable the ABS, we need to go to the Vehicle Editor and work on the brake system:

1. Choose Vehicle Mode. The Vehicle Editor displays the Cadillac Catera.
2. Click on the vehicle's brake system icon (i.e., the brake pedal — you may need to look at the vehicle's undercarriage to locate the brake pedal). The Brake System dialog is displayed (as shown earlier in Figure 5-2).
3. Click on the ABS Installed checkbox to disable the ABS.
4. Press OK to remove the Brake System dialog.

To create the next event, perform the following steps:

1. Choose Event Mode. The Event Editor is displayed.

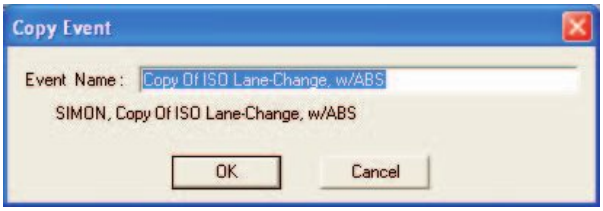


Figure 5-7: Making a copy of the ISO Lane-Change, w/ABS event.

We have a choice now. One way to create our next event is to duplicate the steps used while setting up the first event. But there is a much better way!

1. Click on the Edit menu and choose Copy. The Copy Event dialog is displayed (see Figure 5-7).
2. Enter a name for the new event: **ISO Lane-Change, w/o ABS**.

> **NOTE:** HVE will append the name of the calculation method to the event name, thus the complete event name will become "SIMON, ISO Lane-Change, w/o ABS."

1. Press OK to display the Event Editor.

A duplicate of the first event has been created and set up, and we're ready to execute the lane-change maneuver simulation without ABS enabled.

1. Using the Event Controller, click Reset.
2. Click Play to execute the event with the ABS disabled.

Note that the vehicle's brakes lock and it does not successfully negotiate the lane change maneuver. The vehicle skids straight, running over the cones, and coming to a stop after travelling approximately 140 ft. The average deceleration is 0.66 g. Figure 5-8 shows the vehicle at its rest position. Note also the dark skidmarks caused by locked-wheel braking.

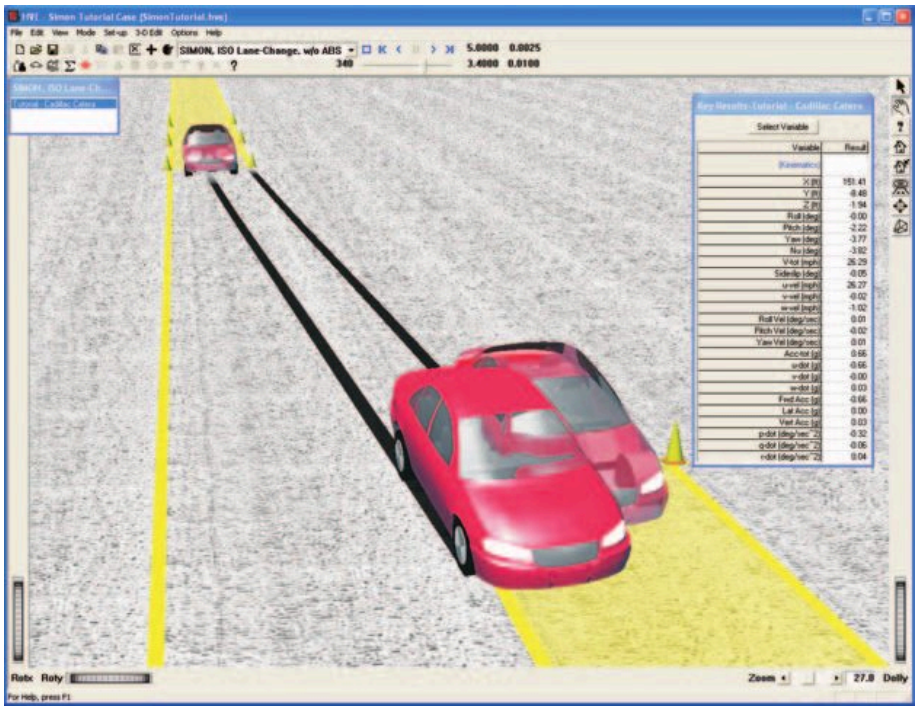


Figure 5-8: ISO Lane-Change, w/o ABS.

Viewing Results

Now that we have produced our SIMON simulations of an ISO 3888 lane-change maneuver with and without ABS, let's take a detailed look at the results. The Playback Editor is used for reviewing and printing reports for each event in the current case, as well as for producing videos.

SIMON produces the following reports:

- **Messages** — A list of messages produced by the current run
- **Accident History** — A table of initial and final positions and velocities
- **Driver Data** — A table of the driver controls used for the current run
- **Environment Data** — A list of the visual and physical environment parameters used by SIMON
- **Event Data** — A table of event-related SIMON options used in the current run
- **Vehicle Data** — A series of tables containing the vehicle data used by SIMON, including tire blow-out information
- **Program Data** — A table containing program control information
- **Variable Output** — A table containing user-selectable, time-dependent simulation results
- **Trajectory Simulation** — A 3-D visualization of the event, displayed at a user-selectable time interval
- **Damage Profiles** — A 3-D visualization of the vehicle damage, displayed at a user-selectable time interval

> **NOTE:** The Damage Profiles report is available only for SIMON events using the DyMESH collision algorithm. The events in this tutorial do not produce a Damage Profile report.

To view the output reports, we need to be in Playback mode:

- Choose Playback Mode. The Playback Editor is displayed.

> **NOTE:** Our tutorial uses the ISO Lane-Change, w/ABS event to illustrate the procedures for viewing output reports. You can use the same procedures for viewing reports for the ISO Lane-Change, w/o ABS event.

Report Windows

The reports listed above are displayed by selecting Report Windows. Each Report Window contains an individual report.

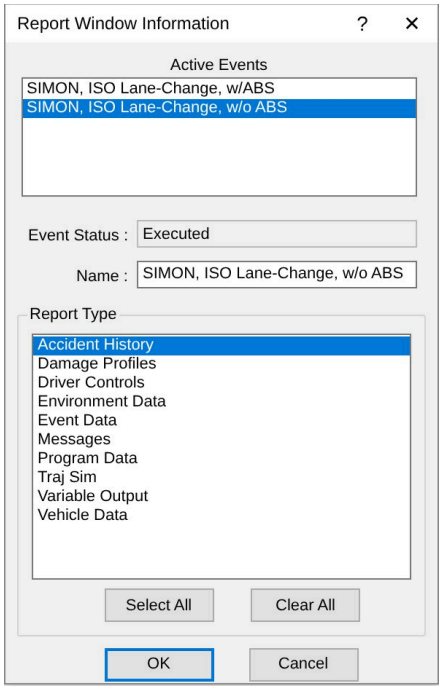


Figure 5-9: Report Window Information dialog, displaying the name of each event in the current case.

To view the reports produced by the SIMON, ISO Lane-Change, w/ABS event, perform the following steps:

1. Click Add New Object. The Report Window Information dialog is displayed, as shown in Figure 5-9, and includes a list of the active events. The Report Window Information dialog also includes the user-editable Report Window Name text field and the Report Type list.
2. Select SIMON, ISO Lane-Change, w/ABS from the Active Events list.
3. In the Report Type list, select one or more of the available reports. *Select All* selects every report; each selected report opens in its own Report Window.
4. Press OK to display the report.

The selected report will be displayed in a resizable window. The following sections illustrate the reports produced for the SIMON, ISO Lane-Change, w/ABS event.

Messages

SIMON produces a number of messages, depending on the outcome of the event. For a complete listing and explanation of the messages, refer to Chapter 6.

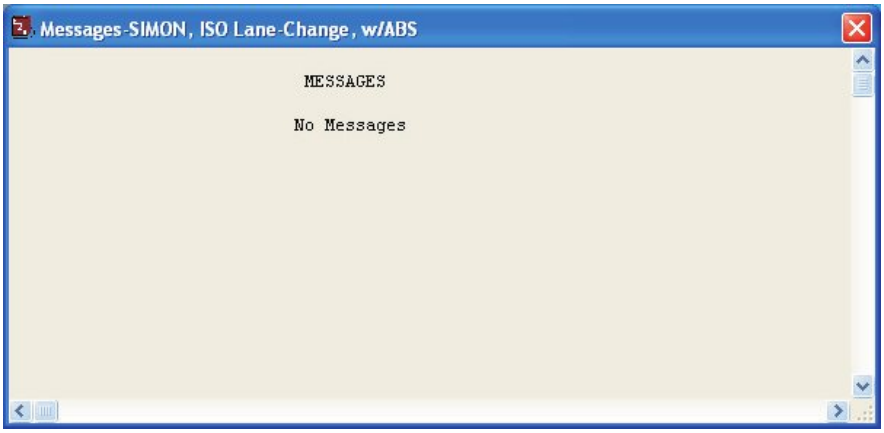


Figure 5-10: Messages Report for SIMON, ISO Lane-Change, w/ABS.

To view the Messages report: Add a new Report Window, select the event, choose **Messages** from the Report Type list and press OK. The Messages report is displayed as shown in Figure 5-10.

Accident History

The Accident History report displays the time and total distance traveled, as well as the position and velocity at the start and end of the run.

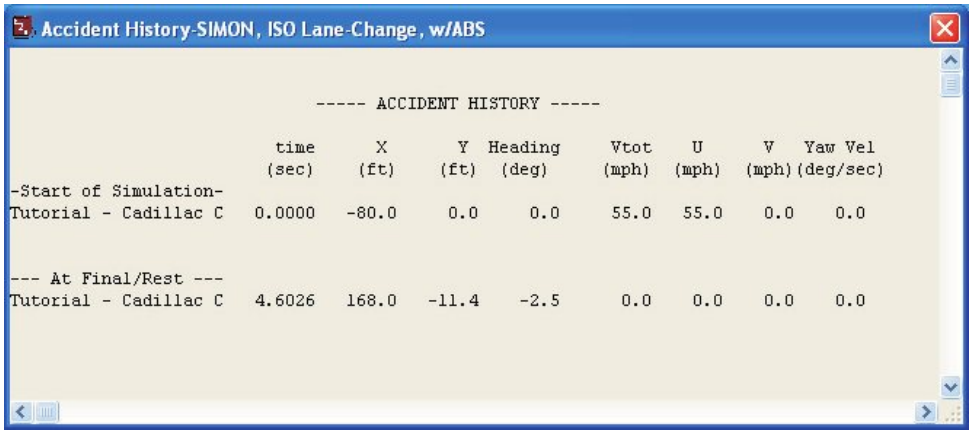


Figure 5-11: Accident History Report for SIMON, ISO Lane-Change, w/ABS.

To view the Accident History report: Add a new Report Window, select the event, choose **Accident History** and press OK. The Accident History report is displayed as shown in Figure 5-11.

Driver Data

The Driver Data report displays the parameters used by the HVE Driver Model and the Braking and Throttle (open-loop) driver tables.

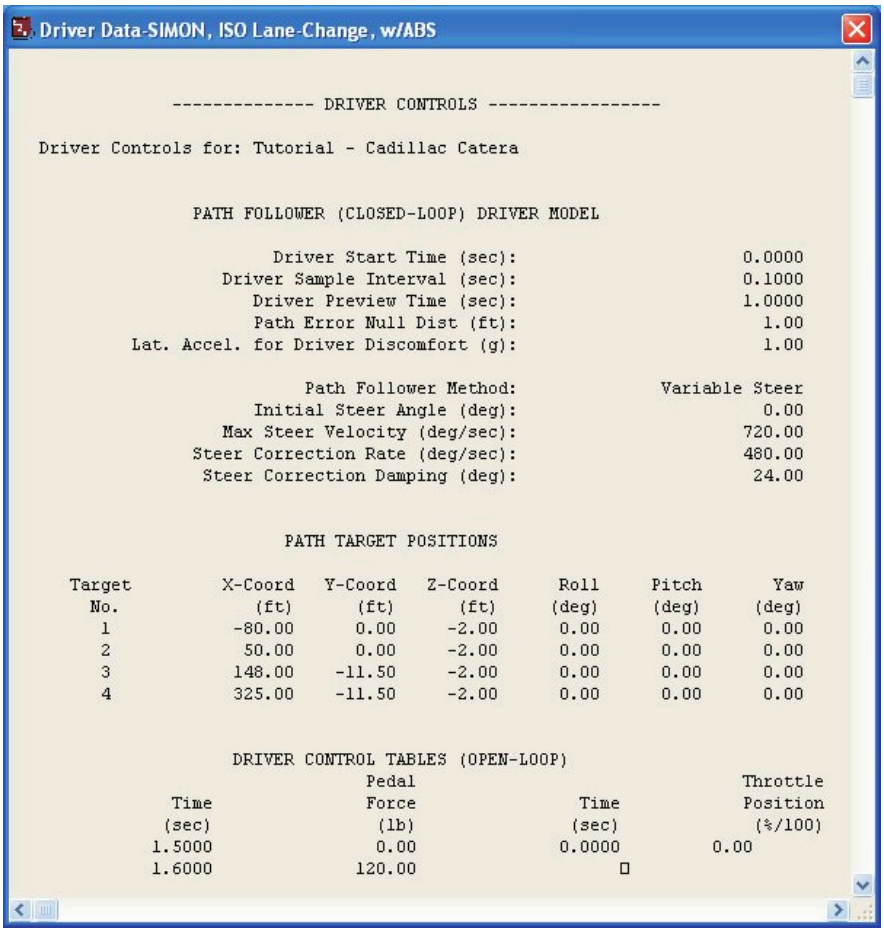


Figure 5-12: Driver Data Report for SIMON, ISO Lane-Change, w/ABS.

To view the Driver Data report: Add a new Report Window, select the event, choose **Driver Data** and press OK. The Driver Data report is displayed as shown in Figure 5-12.

Environment Data

The Environment Data report displays the physical and visual parameters describing the environment.

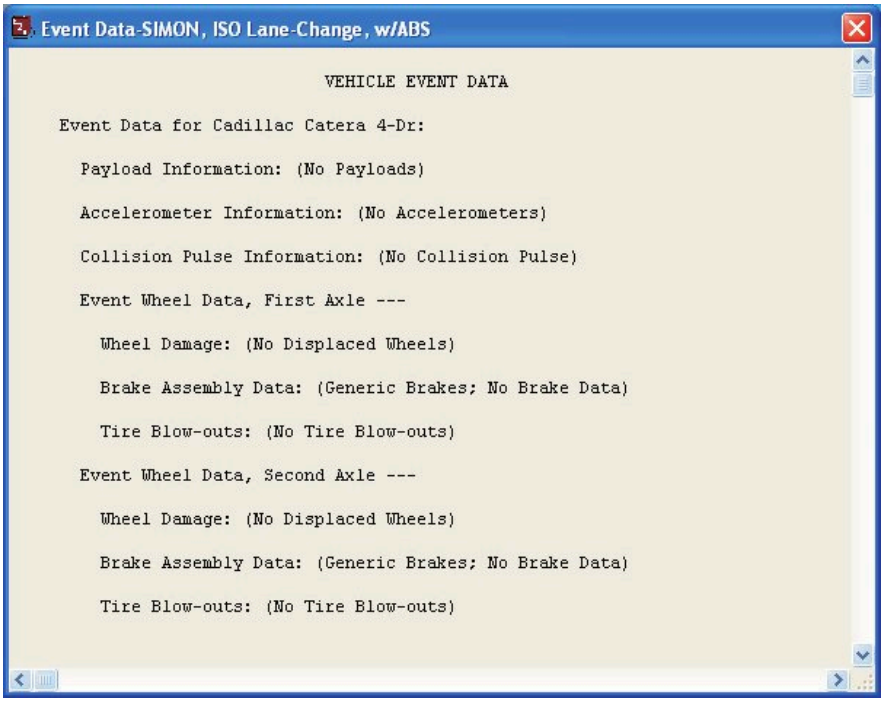


Figure 5-13: Environment Data Report for SIMON, ISO Lane-Change, w/ABS.

To view the Environment Data report: Add a new Report Window, select the event, choose **Environment Data** and press OK. The Environment Data report is displayed as shown in Figure 5-13.

Event Data

The Event Data report displays the event-related parameters (payload, accelerometer, wheel displacement, brake and tire blow-out).



Figure 5-14: Event Data Report for SIMON, ISO Lane-Change, w/ABS.

To view the Event Data report: Add a new Report Window, select the event, choose **Event Data** and press OK. The Event Data report is displayed as shown in Figure 5-14.

Vehicle Data

The Vehicle Data report for SIMON contains all the vehicle data groups (Sprung Mass, Suspensions, Tires, Brakes, etc.).

To view the Vehicle Data report: Add a new Report Window, select the event, choose **Vehicle Data** and press OK. A portion of the Vehicle Data report is shown in Figure 5-15.

> **NOTE:** The SIMON Vehicle Data report is too large to fit in the viewer. Use the scroll bars to view the entire report.

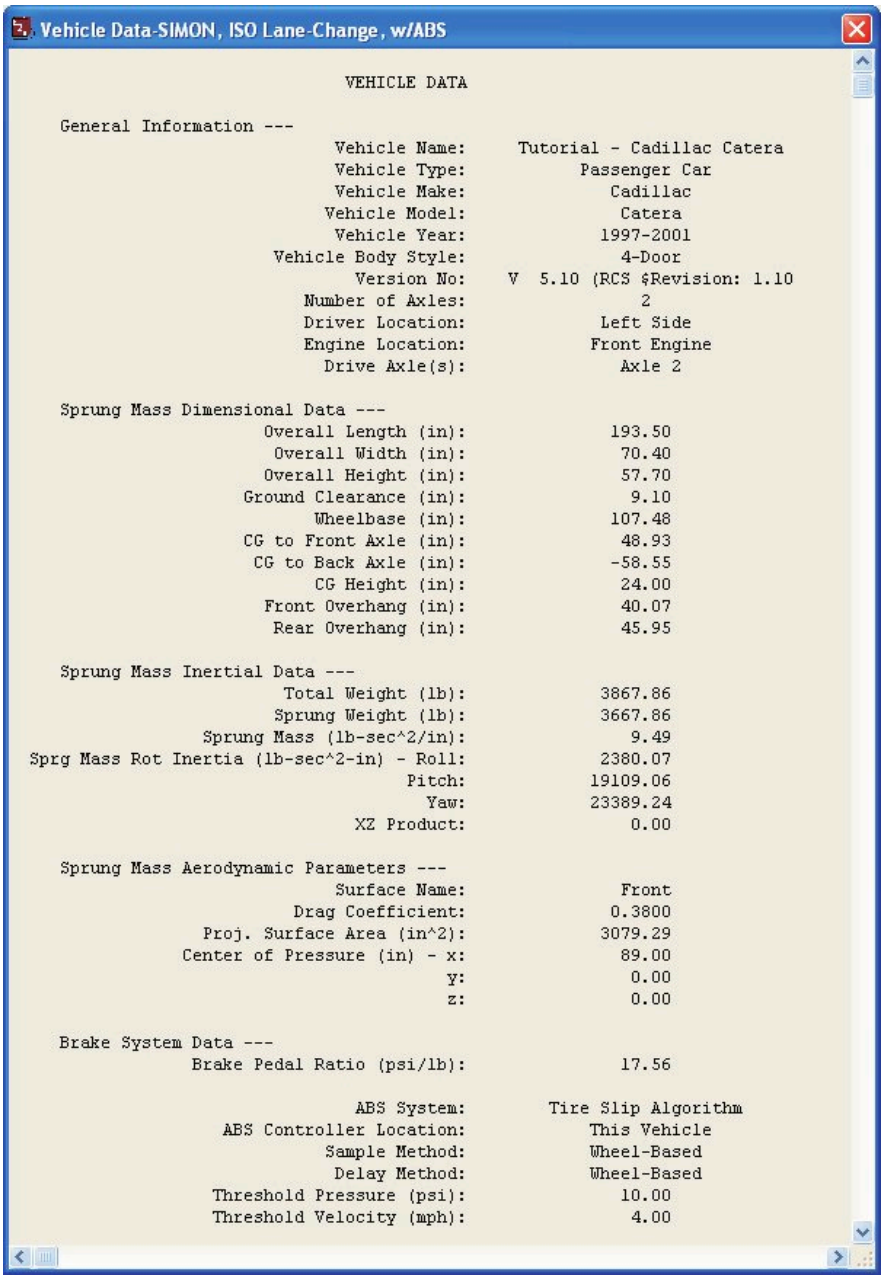


Figure 5-15: Vehicle Data Report for SIMON, ISO Lane-Change, w/ABS (only a portion of the total report is shown).

Program Data

The Program Data report includes the simulation control parameters and other run-time information.

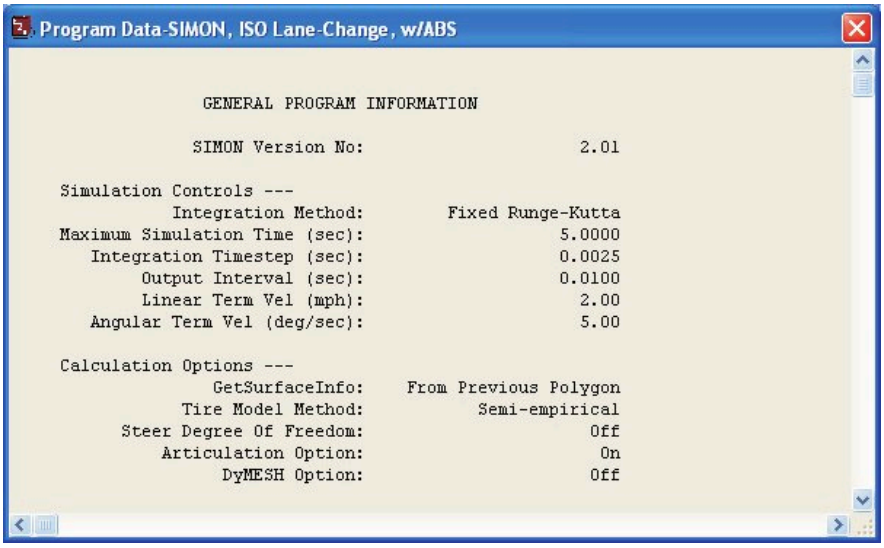


Figure 5-16: Program Data Report for SIMON, ISO Lane-Change Maneuver.

To view the Program Data report: Add a new Report Window, select the event, choose **Program Data** and press OK. The Program Data report is displayed as shown in Figure 5-16.

Variable Output

The Variable Output report is a table of user-selectable, time-dependent simulation results for the current event. To view the Variable Output report: Add a new Report Window, select the event, choose **Variable Output** and press OK.

The Variable Output report is displayed for the SIMON, ISO Lane-Change, w/ABS event. The table is initially empty, so the next step is to select the time-dependent results we wish to display in the table.

Variable Selection

The purpose of our SIMON study is to simulate the effects of the ABS brake system on a lane-change maneuver during heavy braking. To document the resulting path, as well as some other pertinent results, let's select the CG path coordinates, velocity and acceleration from the Variable Selection dialog.

- Click on Select Variables in the SIMON, ISO Lane-Change, w/ABS Variable Output window. The Variable Selection dialog for Cadillac Catera 4-Dr is displayed, as shown in Figure 5-17.

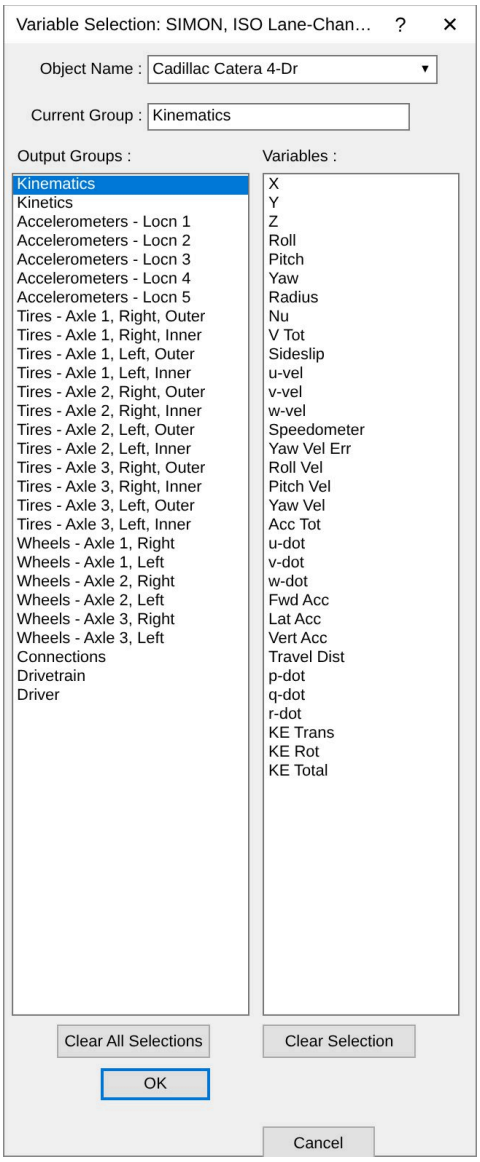


Figure 5-17: Variable Selection dialog for the Cadillac Catera 4-Dr.

The Kinematics Output group is the default selection and the Kinematics variable list is displayed. Let's add the System Brake Pressure and the Right Front and Right Rear Wheel Brake Pressures to the Key Results window:

1. Select the Driver Output Group and then select Brake System from the Variables list.
2. Select the Wheel, Axle 1, Right Output Group and then select Brake Press from the Variables list.
3. Select the Wheel, Axle 2, Right Output Group and then select Brake Press from the Variables list.
4. Press OK to add the selected variables to the Variable Output table.

The Variable Output report for the SIMON, ISO Lane-Change, w/ABS event now includes the selected results (see Figure 5-18).



Time (sec)	Cadillac Cater Brake System (psi)	Cadillac Cater Axle 1, Righ Brake Press (psi)	Cadillac Cater Axle 2, Righ Brake Press (psi)
2.5300	2107.17	1365.89	2107.17
2.5400	2107.17	1365.89	2107.17
2.5500	2107.17	1365.89	2107.17
2.5600	2107.17	1365.89	2057.17
2.5700	2107.17	1365.89	1957.17
2.5800	2107.17	1390.89	1857.17
2.5900	2107.17	1440.89	1757.17
2.6000	2107.17	1490.89	1657.17
2.6100	2107.17	1540.89	1557.16
2.6200	2107.17	1590.89	1457.16
2.6300	2107.17	1640.89	1357.16
2.6400	2107.17	1690.89	1257.16
2.6500	2107.17	1740.89	1157.16
2.6600	2107.17	1790.89	1057.16
2.6700	2107.17	1790.89	1007.16
2.6800	2107.17	1790.89	1007.16
2.6900	2107.17	1790.89	1007.16
2.7000	2107.17	1790.89	1007.16
2.7100	2107.17	1790.89	1007.16
2.7200	2107.17	1690.89	1032.16
2.7300	2107.17	1590.88	1082.16
2.7400	2107.17	1490.88	1132.16
2.7500	2107.17	1440.88	1182.16
2.7600	2107.17	1440.88	1232.16
2.7700	2107.17	1440.88	1282.16
2.7800	2107.17	1440.88	1332.16
2.7900	2107.17	1440.88	1382.16
2.8000	2107.17	1465.88	1432.16

Figure 5-18: Variable Output Report for SIMON, ISO Lane-Change, w/ABS.

Graphing Results

It is easy to graph the results displayed in the Variable Output table: press **Graph**. The selected results are displayed vs. Time, as shown in Figure 5-19.

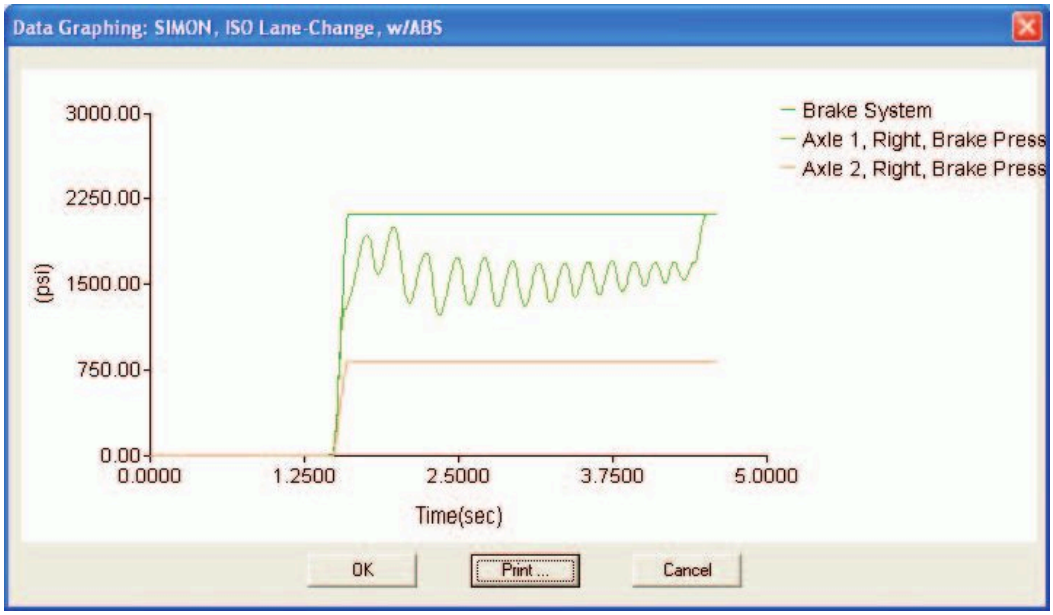


Figure 5-19: Graph of master cylinder pressure and right front and right rear wheel brake pressures.

> **NOTE:** The variables selected for graphing must have the same units.
>
> **NOTE:** Only the first six variables are graphed.
>
> **NOTE:** Results from the Variable Output table can also be exported to other applications (e.g., Excel).

Trajectory Simulation

Let's display a trajectory simulation for this event. To view the Trajectory Simulation: Add a new Report Window, select the event, choose **Trajectory Simulation** and press OK.

The Trajectory Simulation viewer is displayed for the SIMON, ISO Lane-Change, w/ABS event (see Figure 5-20). The viewer shows the vehicle at its initial position.

To visualize the motion, perform the following steps:

1. Click Play (single right-arrow). The simulation begins and is displayed at the current Playback output interval.
2. Click Pause. The simulation stops.
3. Click Reverse (single left-arrow). The simulation plays in reverse.
4. Click Pause. The simulation stops.
5. Click Rewind (left arrow with bar). The simulation returns to the start.
6. Click Advance to End (right arrow with bar). The simulation advances to the end of the run.

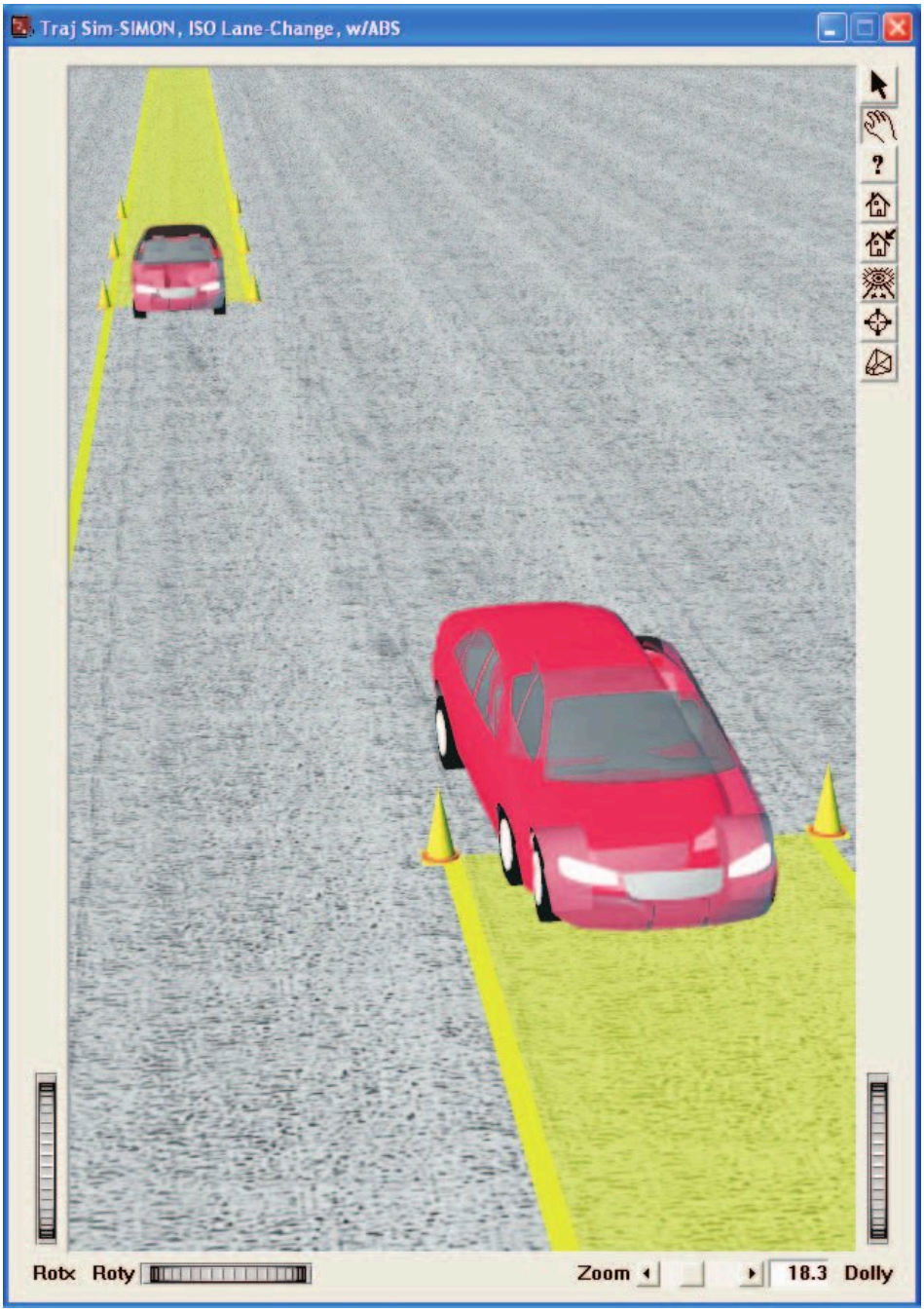


Figure 5-20: Trajectory Simulation Report for SIMON, ISO Lane-Change Maneuver.

Printing

The final step is to print the above reports. Printing reports is simple. All you do is choose a report and print it. For example:

1. Click on the dialog header of the Variable Output — SIMON, ISO Lane-Change, w/ABS report. The dialog header is highlighted and the Variable Output window pops to the top of the display (if it isn't there already), indicating it is the current window.
2. Click on the File menu and choose Print. The Print dialog is displayed, allowing the user to select from several available print options.

> **NOTE:** Alternatively, you can click on the print icon in the upper menu bar.

1. Press OK. The Variable Output report is printed on the system printer.

That's all there is to it! You can print any other report using the same three steps described above.

> **NOTE:** The Print dialog provides several options. Refer to the HVE User's Manual for more information.
>
> **NOTE:** The font size of both the printed reports and screen display may be edited by clicking on the Options menu and choosing Preferences. Use the Font Size option list to change the size.

Chapter 6 — Messages

Messages can be issued by either the operating system, HVE or SIMON. Some of the more common operating system messages are described in the Appendix. A more thorough discussion of operating system messages may be found in the operating system manual.

Messages produced by HVE are described in the HVE User's Manual, Appendix II.

Some conditions detected by SIMON halt the event and are reported by HVE as an *Event Termination* dialog rather than as an entry in the SIMON Messages report. These identify the vehicle, axle, side and wheel involved. The suspension terminations — *Excessive Suspension Force*, *Excessive Suspension Deflection*, *Excessive Suspension Velocity*, and the two DyMESH wheel impact terminations — are described with their limits under *Suspension Force* in Chapter 4 — Calculation Method.

Messages issued by SIMON appear in the SIMON Messages output report and may be one of three types:

- **Fatal Messages (Level 1)** — These messages halt execution and are caused by the program's inability to continue the calculation procedure based on intermediate results that are inconsistent or incompatible with basic program assumptions.
- **Diagnostic Messages (Level 2)** — These messages do not halt execution, but are the result of the internal consistency checks performed by the program and suggest inconsistencies in the results of which the user should be aware. These inconsistencies may be the result of bad or suspicious data.
- **Informative Messages (Level 3)** — These messages do not halt execution, but provide information about execution, not affecting accuracy, but suggestive of something the user may want to review to ensure data were as intended.

All messages issued by SIMON are listed below. Each message includes a description of the cause, the level of the message (Informative, Diagnostic or Fatal), and the recommended action to eliminate the problem. If you receive an error message not listed below, first check to see if it is an HVE or operating system message by referring to the HVE Appendix or your computer's operating system manual. If the message is not listed in either place, please contact EDC with the offending error.

(updated: this catalog covers the complete set of messages issued by the current version of SIMON. Messages present in the Fifth Edition manual are retained below with their original level designations; messages added since then are marked "(added)" and their level is indicated by the message prefix — WARNING/ERROR entries behave as diagnostics or fatal errors, NOTE entries are informative.)

Processing was interrupted by the user. At the time processing was interrupted, the vehicle(s) were still moving. *(Level 3 — Informative)*

This message is issued to indicate that at the time the user halted the simulation, at least one of the vehicles had not reached its rest position. No response by the user is necessary.

The simulation was terminated because a unique solution for the coordinates of the tire-ground contact patch could not be calculated. Please send the offending case file to EDC for evaluation. *(Level 1 — Fatal)*

This message will be issued if a matrix describing the tire-ground contact patch is singular. This condition has never occurred in EDC's testing of SIMON. If you experience this problem, please contact EDC Technical Support.

The simulation was terminated because the current deflection of a tire exceeded the user-specified Maximum Tire Deflection assigned for that tire. *(Level 1 — Fatal)*

A tire hit a tall curb or other significant elevation change (an elevation change greater than the maximum allowed tire deflection; see Vehicle Editor, Tire Physical Properties). This can also occur because of a defect (i.e., a hole) in the terrain beneath the tire. Carefully inspect the terrain beneath the tire. This condition also suggests that a wheel may have been broken during the event; SIMON does not model failed wheel rims.

The first step in calculating tire forces is to calculate the radial tire force from the current tire deflection (the change in the tire rolling radius compared to an unloaded tire). If the deflection of a tire exceeds the user-specified Maximum Tire Deflection assigned for that tire, the radial force could be substantial, leading to dynamic instability in the simulation. To prevent this from occurring, the run terminates.

This problem can sometimes be solved by increasing slightly the Maximum Tire Deflection (see Tire Physical Data parameters) and/or reducing the Vehicle Trajectory Integration Timestep (see Simulation Controls).

(updated: in the current version this message is prefixed "WARNING:".)

The simulation was terminated due to an error in calculating the wheel spin acceleration. Check the tire spin inertia. *(Level 1 — Fatal)*

This message is issued when the wheel spin acceleration is so high that it causes a dynamic instability. If this occurs, try increasing tire spin inertia (see Vehicle Editor, Tire Physical Properties).

The simulation was terminated because the current roll or pitch angle exceeded the allowable limit. *(Fatal; added)*

(updated: this message is issued by the current version but was not listed in the Fifth Edition manual. It indicates the vehicle attitude exceeded the model's allowable roll/pitch range.)

The simulation was terminated because the inertial matrix is singular. This condition usually suggests a defective inertial matrix. Please send the offending case file to EDC for evaluation. *(Level 1 — Fatal)*

The equations of motion are expressed in matrix form and solved by inverting the inertial matrix. A singular matrix cannot be inverted, so the equations of motion could not be solved. If this problem occurs, first check to see that no inertial elements of the vehicle (sprung mass, axles, wheels) have been assigned a zero mass or rotational moment of inertia. If the problem persists, please send your case file to EDC for evaluation.

The simulation was terminated because the current engine speed (RPM) was greater than the maximum RPM specified for the engine (see Vehicle Editor, Drivetrain Data, Engine Power/Torque Table). *(Level 1 — Fatal)*

This problem might occur if the wheels leave the ground during a heavy throttle application; in this case the problem may be eliminated by reducing the throttle input (see Driver Controls, Throttle Table). The problem might also occur while driving in a low gear if the driver has a "lead foot." Engine RPM exceeded the highest value in the RPM vs. Power curve (see Drivetrain dialog). The solution is to shift into a higher gear.

The HVE Driver Model predicted the maneuver required a steering wheel input rate that exceeded the user-defined Maximum Steering Velocity (see Driver Controls, HVE Path Follower dialog). *(Level 2 — Diagnostic)*

This diagnostic message alerts the user to the fact that the HVE Driver Model used a higher Maximum Steering Velocity than was specified in the Driver Controls, HVE Path Follower dialog. Steering Velocity as a function of time can be found in the Variable Output.

The simulation was terminated by the HVE Driver Model because the maneuver resulted in a lateral acceleration that exceeded the user-defined Driver Lateral Acceleration Comfort Level (see Driver Controls, HVE Path Follower dialog). *(Level 1 — Fatal)*

In order to complete the required maneuver, the user should increase the Driver Lateral Acceleration Comfort Level (see Driver Controls, HVE Path Follower dialog). However, this suggests the attempted maneuver is beyond the comfort level of the typical driver.

The simulation was terminated because GetSurfaceInfo() failed. *(Level 1 — Fatal)*

This message occurs if the surface-information search failed to return a valid surface elevation. SIMON should not return this condition; contact EDC if it occurs.

The simulation was terminated because the current articulation angle between a tow vehicle and its trailer exceeded the allowable range (see Vehicle Editor, Connections). This is normally the sign that a "Jackknife" has occurred. *(Level 1 — Fatal)*

This message is self-explanatory. The maximum articulation angle is defined in the Rear connections parameter (see Vehicle Editor). The value is established by the angle required to create pinching between the tow vehicle and trailer. Note that such pinching would cause forces modeled only by DyMESH, so the run terminates.

(updated: the current DyMESH Options dialog provides a Tow Vehicle / Trailer Contact option that enables mesh contact between connected vehicles; see Chapter 2.)

The simulation was terminated because a fatal error occurred while calculating derivatives (accelerations). This occurred due to excessively large forces or a general failure in the calculations that resulted in a non-numeric value. Please send the offending case file to EDC for evaluation. *(Level 1 — Fatal)*

This error should not occur. If it does, however, there is no easy solution for the user. Please send the case to EDC, where we can diagnose the cause of the problem.

NOTE: The inter-vehicle connections for two connected vehicles have different elevations above ground. This is not an error condition, but it is unusual. CG Height + Connection z-coord are normally about equal for both vehicles. *(Level 3 — Informative)*

This message is self-explanatory. To confirm the difference in connection heights, go to the Vehicle Editor and choose Move CG to obtain the static CG height (it is displayed as a positive value; however, it is negative with respect to the earth-fixed coordinate system), then choose Connections to obtain the connection z-coordinate. Do this for both vehicles (note that you'll use the rear connection for the tow vehicle and the front connection for the trailer). The connection elevation is −CG Height + Connection z-coordinate. Adjust the connection z-coordinate for either or both vehicles if you wish to make the connection elevations the same.

Messages Added Since the Fifth Edition

(updated: the following messages are issued by the current version of SIMON but were not documented in the Fifth Edition manual. Most relate to the DyMESH collision model and damage/restitution reporting.)

NOTE: One or more vehicles has (have) a negative crush depth. *(Informative; added)*

Refer to the Damage Data report's Crush Depth Tables to identify the specific vehicle(s). This may be the result of a vehicle geometry with reversed surface normals or missing polygons. It may also be the result of normal folding of the damaged surface. Inspect the vehicle geometry for these conditions.

NOTE: The collision resulted in more than 10 impulses on one or more vehicles. *(Informative; added)*

This does not result in event termination. However, only the first 10 impulses will be included in the CollisionData report and DamageStudio window.

An error occurred while calculating vehicle handling properties. *(Diagnostic; added)*

NOTE: The simulation output interval (see Simulation Controls dialog) is rather large for a collision simulation. *(Informative; added)*

Peak collision forces and/or accelerations may be filtered from Key Results and Variable Output. This issue has no effect on the overall accuracy of the simulated trajectories or damage profiles. To avoid this issue, reduce the output interval to 0.02 seconds (or less).

NOTE: At the time the simulation was terminated, one or more vehicles were still undergoing a collision. *(Informative; added)*

The CollisionData results (see Damage Data output report) are incomplete for the last recorded impulse.

NOTE: One or more vehicles has a bad CDC (see Damage Data output report). *(Informative; added)*

This usually occurs while SIMON attempts to build the damage profile using the damaged vehicle geometry. There may be an insufficient number of damaged vertices to build the damage profile. Review the Damage Data report's Damage Width, Offset and Crush Depth tables.

WARNING: The coefficient of restitution is negative. *(Diagnostic; added)*

This implies the striking vehicle traveled through the struck vehicle during impact. A negative restitution coefficient may also occur as a result of a large angular velocity (> 100 deg/sec) at separation, in which case this message may be ignored.

WARNING: The coefficient of restitution is greater than 1.0. *(Diagnostic; added)*

This implies the crash created energy. The normal range is approximately 0 to 0.25, perhaps slightly greater for low-speed impacts.

NOTE: The coefficient of restitution is larger than normally associated with vehicle collisions. *(Informative; added)*

The normal range is approximately 0 to 0.25, perhaps slightly greater for low-speed impacts.

ERROR: The closing velocity is less than zero, indicating a calculation error has occurred. *(Diagnostic; added)*

Two vehicles cannot collide if their closing velocity is less than zero.

NOTE: The method used to calculate the coefficient of restitution in the Event Data output report (see INTER-VEHICLE COLLISION DATA) assumes an instantaneous impact. *(Informative; added)*

One or more collisions resulted in sustained contact (more than 0.5 seconds). A coefficient of restitution was not calculated for those collisions.

NOTE: One or more vehicles has (have) a manual transmission that is not in gear, but has the throttle applied. *(Informative; added)*

This is like revving the engine in neutral. Is this the intent?

NOTE: The HVE Driver Model was turned off at impact for one or more vehicles because of a collision. *(Informative; added)*

The HVE Driver Model is not valid during or after impact.

Chapter 7 — Technical References

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2. Day, T.D., "Differences Between EDVDS and Phase 4," SAE Paper No. 1999-01-0103, Society of Automotive Engineers, Warrendale, PA, 1999.
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4. Day, T.D., and Metz, L.D., "The Simulation of Driver Inputs Using a Vehicle Driver Model," SAE Paper No. 2000-01-1313, Society of Automotive Engineers, Warrendale, PA, 2000.
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6. Etkin, B., *Dynamics of Flight*, John Wiley and Sons, 1959.
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9. Segal, D.J., *Highway-Vehicle-Object Simulation Model — Volume 3: Engineering Manual — Analysis*, US DOT, Federal Highway Administration Report No. FHWA-RD-76-164, Calspan Corporation, February 1976.
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11. Segal, D.J., *Highway-Vehicle-Object Simulation Model — Volume 2: Validation*, DOT Contract Number DOT-FH-11-8265, FHWA Report Number FHWA-RD-76-165, 1976.
12. Nalecz, A.G., et al., *Sensitivity Analysis of Vehicle Tripped Rollover Model*, NHTSA — U.S. DOT Final Report No. DOT-HS 807 300, Washington D.C., July 1988.
13. Bernard, J.E., Winkler, C.B., and Fancher, P.S., *A Computer-based Mathematical Method for Predicting the Directional Response of Trucks and Tractor-Trailers*, Phase II Technical Report, Motor Truck Braking and Handling Performance Study, NTIS PB-221-630, Highway Safety Research Institute, The University of Michigan, Ann Arbor, June 1973.
14. *Two-Moving Vehicle Crash Test, 1984 Ford F-150 Pickup into a 1992 Ford Explorer*, June 23, 1999, Exponent Failure Analysis Test and Engineering Center, Report No. PH06015/C0F0/0699/R256, Phoenix, 1999.
15. Shoemaker, N.E., *Research Input for Computer Simulation of Automobile Collisions — Volumes II and III — Staged Collision Reconstructions*, Calspan Corporation, DOT HS-805 037 and DOT HS-805 038, Buffalo, 1978.
16. Day, T.D., "Validation of the SIMON Model for Vehicle Handling and Collision Simulation — Comparison of Results with Experiments and Other Models," SAE Paper No. 2004-01-1207, Society of Automotive Engineers, Warrendale, PA, 2004.

(updated: the current SIMON engine also prints a reference list at the end of SIMON reports. That list includes additional works published after the Fifth Edition manual, notably:)

- Day, T.D., Roberts, S.G., "SIMON: A New Vehicle Simulation Model for Vehicle Design and Safety Research," SAE Technical Paper No. 2001-01-0503, Society of Automotive Engineers, Warrendale, PA, 2001.
- Jackson, L., Poland, K., "Downhill Commercial Vehicle Simulations — Part A (Tractor/Semi-trailer Brake Fade)," National Transportation Safety Board, HVE White Paper No. WP-2003-1, Engineering Dynamics Corporation, Beaverton, OR, 2003.
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Chapter 8 — Index

(updated: the printed edition's page-number index has been adapted for this Markdown edition. Entries now point to the chapter files and sections where each topic is discussed. Use your viewer's text search for fine-grained lookup.)

Topic	Where to find it
3-D Surface/Environment Geometry	Chapter 2 — Environment
3-D Vehicle Geometry (Geometry File)	Chapter 2 — Vehicles, Sprung Mass
3-D Visualization (Trajectory Simulation)	Chapter 3 — Trajectory Simulations
ABS (Antilock Braking)	Chapter 1, Chapter 2 — Brake, Chapter 5 — Tutorial
Accelerometers	Chapter 2 — Event Set-up, Chapter 3 — Variable Output
Accident History (report)	Chapter 3 — Alpha-Numeric Reports, Chapter 5
Accident History Basis (Force/Acceleration)	Chapter 2 — Event Calculation Options
Aerodynamics Model	Chapter 4 — Aerodynamics Model
Anti-pitch	Chapter 2 — Unsprung Mass
Brake Designer	Chapter 4 — Driver Controls
Brake System / Brake Table / Brake Torque	Chapter 2, Chapter 4 — Wheel Torque
Camber Stiffness	Chapter 2 — Tire
Collision Acceleration Threshold	Chapter 2 — Event Calculation Options
Collision Pulse	Chapter 2 — Event Set-up
Connections (Inter-Vehicle)	Chapter 2, Chapter 4 — Inter-Vehicle Connection Model
Cornering Stiffness	Chapter 2 — Tire
Damage Data (report), Crush Depth (Excl./Incl. Free Space)	Chapter 3 — Alpha-Numeric Reports, Chapter 2 — Event Calculation Options
Damage Profile	Chapter 3 — Displaying a Damage Profile
Degrees of Freedom	Chapter 4 — Basis For Analysis
Dollys (Converter/Fixed)	Chapter 4 — Dollys
Drive Torque / Drivetrain	Chapter 2 — Drivetrain Data, Chapter 4 — Wheel Torque
Driver Controls (Steering, Braking, Throttle, Gears)	Chapter 2, Chapter 4 — Driver Controls
Driver Data (report)	Chapter 3, Chapter 5
DyMESH (collision model, options, wheel contact)	Chapter 2 — DyMESH Options, Chapter 4 — Sprung Mass Impact Model
Environment (creating, visual/physical data)	Chapter 2 — Environment
Environment Data (report)	Chapter 3, Chapter 5
Equations of Motion	Chapter 4 — Equations of Motion
Event (creating, set-up, executing)	Chapter 2 — Event
Event Data (report)	Chapter 3, Chapter 5
Friction Table	Chapter 2 — Tire
Get Surface Information	Chapter 2 — Get Surface Information Options, Chapter 4 — Terrain Definition
Graphing Results	Chapter 3 — Graphic Reports, Chapter 5 — Graphing Results
Human Occupants	Chapter 2 — Humans, Chapter 4 — Human Occupants
HVE Driver Model (Path Follower)	Chapter 4 — Driver Controls, Chapter 5 — Steering Input
Hydroplaning Model	Chapter 2 — Event Calculation Options; Chapter 4 — Hydroplaning Model
Inertias (sprung/unsprung, payload)	Chapter 2, Chapter 4 — Payloads
Inter-tandem Axle Load Transfer	Chapter 2 — Suspension
Jounce and Rebound Stops	Chapter 2 — Suspension, Chapter 4 — Suspension Force
Messages (catalog)	Chapter 6 — Messages
Payloads	Chapter 2 — Vehicle Payload, Chapter 4 — Payloads
Position/Velocity	Chapter 2 — Event Set-up
Program Data (report)	Chapter 3, Chapter 5
References	Chapter 7 — Technical References
Rolling Resistance	Chapter 4 — Tire Model

Topic	Where to find it
Roll Steer / Bump Steer	Chapter 4 — Vehicle-fixed Wheel Orientation
Simulation Controls	Chapter 2 — Simulation Controls
Springs and Shocks	Chapter 2 — Suspension, Chapter 4 — Suspension Force
Steer Degree of Freedom	Chapter 2 — Event Calculation Options, Chapter 4 — Steering System
Steering System (gear ratio, stops, friction)	Chapter 4 — Steering System
Suspension (independent, solid axle)	Chapter 2, Chapter 4
Tire Blow-out Model	Chapter 2 — Vehicle Wheels
Tire Model (semi-empirical)	Chapter 4 — Tire Model
Tire-Terrain Models (Point Contact, Radial Spring, Sidewall Impact)	Chapter 2, Chapter 4 — Tire-Road Contact Patch
Trajectory Simulation	Chapter 3, Chapter 5
Tutorial (ISO 3888 Lane Change)	Chapter 5 — SIMON Tutorial
Validation	Chapter 1 — Validation, Chapter 7
Variable Output (report, output groups)	Chapter 3 — Variable Output Table, Chapter 5
Vehicle Data (report)	Chapter 3, Chapter 5
Vehicle Mesh (tessellation, friction, restitution)	Chapter 2 — Event Set-up
Wheel Damage / Wheel Displacement	Chapter 2 — Vehicle Wheels, Chapter 4 — Sprung Mass Impact Model
Wheel Position / Orientation	Chapter 4
Wheel Spin	Chapter 4 — Wheel Spin
Wind Speed and Direction	Chapter 2 — Environment Physical Data, Chapter 4 — Aerodynamics Model

Electronic Stability Systems (TCS and YSC)

SIMON models two electronic stability systems as part of each vehicle's drivetrain: **Traction Control (TCS)** and **Yaw Stability Control (YSC)**, collectively referred to as the vehicle's **ESS (Electronic Stability Systems)**. Each system is enabled individually per vehicle and configured with its own activation thresholds and rates in the Vehicle Editor. The Anti-Lock Brake System (ABS) is a separate model, described with the brake system; where both are present, YSC acts on the brake pressures first and ABS then modulates the result.

The ESS model is structured the way a production controller is: a **sensor layer** synthesizes the signals an ECU would measure — a speedometer signal, a yaw-rate signal, and per-wheel speed signals — and two **control laws** act on the resulting errors. TCS intervenes by redistributing drive torque; YSC intervenes by redistributing drive torque *and* modulating individual wheel brake pressures.

The description below reflects the behavior of the current version of HVE.

Sensor models

The sensor signals are recomputed on every pass through the wheel-torque calculation:

- **Speedometer.** The vehicle speed signal V is the average of the wheel-speed sensors on the **drive wheels** (spin velocity × tire static loaded radius).
- **Yaw-velocity error.** The expected yaw velocity follows from the driver's steering input and the vehicle geometry,

$$\dot{\psi}_{expected} = \frac{V \tan\delta}{L}$$

where δ is the front-wheel steer angle (steering-wheel angle ÷ column ratio) and L is the wheelbase computed from the axle positions (for a 3-axle vehicle, the distance to the midpoint of the rear tandem). The yaw velocity error is the sensed yaw velocity minus this expectation:

$$e_{\dot{\psi}} = \dot{\psi}_{sensor} - \dot{\psi}_{expected}$$

A positive error means the vehicle is rotating faster than the driver's steering asks for.

- **Per-wheel spin comparisons.** For each wheel, the expected spin velocity accounts for the wheel's lateral position y during a turn:

$$\omega_{expected} = \frac{V - \dot{\psi}\, y}{R_{SLR}}$$

where R_{SLR} is the tire static loaded radius. From the wheel-speed sensors the model then forms, per wheel: the **spin-velocity error** ($\omega_{sensor} - \omega_{expected}$), the **difference from the other side** of the same axle, the difference from the axle average, the **difference between axle averages** (axle-to-axle), and the difference from the whole-vehicle average.

Traction Control (TCS)

TCS runs during the drive-torque split calculation and is considered only when the total drive torque (engine torque × transmission ratio × final drive ratio) is at or above the **Minimum Drive Torque** threshold. Starting from an even split among the drive axles, it makes two corrections:

1. **Axle-to-axle.** If the average wheel speed of a drive axle differs from another drive axle's average by at least the **Axle Difference** threshold, that axle's torque fraction is reduced in proportion to the difference:

$$S_{axle} - = R_{Tq}\, \Delta\omega_{axle}$$

where R_{Tq} is the user-set **Drive Torque Rate**. The fraction is clamped between 0 and 1, so torque migrates away from a spinning axle toward the axle(s) with traction.

1. **Side-to-side.** If the left and right wheel speeds on a drive axle differ by at least the **Side Difference** threshold, the axle's torque is re-split between the sides. The right-side share is adjusted by $R_{Tq}\, S_{axle}\, \Delta\omega_{side}$ — proportional to both the axle's current torque fraction S_{axle} and the right-minus-left wheel speed difference — and the left side receives the remainder, clamped so neither side goes below zero or above the axle's total. This mimics a brake- or clutch-based limited-slip intervention.

When either correction fires, the **TCS active** flag is set — the equivalent of the TCS indicator lamp — and reported in the Variable Output.

Yaw Stability Control (YSC)

YSC runs per wheel inside the brake-pressure calculation, after the pedal force, proportioning, push-out and rise-time effects and **before ABS**. It is gated twice:

- The vehicle speed must be at or above the **Minimum Velocity** threshold.
- Either the yaw-velocity error magnitude must reach the **Yaw Velocity Error** threshold, or the wheel's spin-velocity error magnitude must reach **twice** the **Spin Velocity Error** threshold.

Once active, YSC makes two interventions:

1. **Drive-torque side shift.** If drive torque is at or above the **Minimum Drive Torque** threshold and the axle's side-to-side wheel speed difference reaches the **Side Difference** threshold, the wheel's share of the axle torque is adjusted by $R_{Tq}\, S_{axle}\, \Delta\omega_{side}$ (proportional to the axle's torque fraction, as in TCS), clamped to the axle total. Each side is adjusted using its own signed side difference, so the two sides' corrections mirror one another: the faster side sheds torque share and the slower side gains it, each clamped independently.
2. **Brake modulation.** If the driver's brake system pressure is at or above the **Minimum Brake Pressure** threshold and the wheel's spin-velocity error magnitude reaches the **Spin Velocity Error** threshold, the wheel's brake pressure is adjusted in proportion to the error:

$$P\,+\,=R_P\,e_\omega$$

where R_P is the **Brake Pressure Rate**. A wheel spinning faster than expected receives more brake pressure; a wheel turning slower than expected (approaching lock) receives less. Pressure is floored at zero.

When either intervention fires, the **YSC active** flag is set — the YSC indicator lamp — and reported in the Variable Output.

User parameters

Both systems are configured in the Vehicle Editor's drivetrain ESS dialog. The thresholds gate when the systems may act; the rates set how aggressively they act per unit of error.

Parameter	Used by	Meaning
Minimum Velocity	YSC	Vehicle speed below which YSC never intervenes
Yaw Velocity Error	YSC	Yaw-rate error that activates YSC
Spin Velocity Error	YSC	Wheel spin error for brake modulation (2× this value also activates YSC)
Side Difference	TCS, YSC	Left/right wheel-speed difference that triggers a side torque shift
Axle Difference	TCS	Axle-to-axle average speed difference that triggers an axle torque shift
Minimum Drive Torque	TCS, YSC	Drive torque below which no torque redistribution occurs
Minimum Brake Pressure	YSC	System brake pressure below which no brake modulation occurs
Drive Torque Rate	TCS, YSC	Change in torque split per unit wheel-speed error
Brake Pressure Rate	YSC	Change in brake pressure per unit wheel-speed error

Outputs

- The **Variable Output** drivetrain group includes the TCS and YSC active flags, so the timing of each intervention can be plotted against the vehicle response — the simulation equivalent of watching the dash lamps.
- The **Program Data** report lists the vehicle's ESS configuration: which systems are fitted, and the threshold and rate values in effect for the run.

Notes and limitations

- The vehicle data structure also carries an **ESP** (Electronic Stability Program) flag, but no combined ESP control law is implemented in the current version of HVE — TCS and YSC are the two active systems. In particular there is no engine-torque cut and no yaw-moment-targeted braking of a specific corner based on the yaw error alone.
- TCS is purely wheel-speed based; the yaw-velocity error influences only YSC, and there only as an activation gate — the magnitude of every correction is driven by wheel-speed errors.
- The sensor models are idealized: the yaw-rate signal is the true simulated yaw velocity, with no noise, bias, or latency.
- The corrections are proportional with clamping. There is no integral or derivative action, and interventions persist only as long as the triggering error persists.