VEHICLE MODELING FOR HIGH-DYNAMIC DRIVING SIMULATOR APPLICATIONS

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ABSTRACT

When introducing commercial driving simulators a few years ago, the main interest was focused either on operating the vehicle or on the reactions of the driver in traffic and in emergency situations. A comprehensive modeling of the vehicle dynamics was not required for these tasks. However, there are driving simulator applications which make much higher demands on the modeling of the vehicle dynamics. Training simulators for drivers of emergency vehicles or cross-country trucks have to provide a realistic vehicle dynamics simulation including a detailed tire-ground interaction.

This paper presents a generic toolbox of vehicle subsystems that can be used to assemble and simulate in real-time almost any kind of wheeled or tracked vehicle. The toolbox includes all the mechanical or electronic components found in on-road or off-road vehicles. The equations of motion are formulated and solved automatically. The vehicle dynamics is calculated for all six degrees of freedom considering the topography of the terrain, the type of the ground and the aerodynamic forces. Interaction of the vehicle with the terrain is based on an elaborated tire model that allows sliding, skidding or completely loosing the contact to the ground.

For illustrating the capabilities of the toolbox, a detailed model of a heavy off-road truck is presented. Several results from the validation procedure and from special driving maneuvers are used to demonstrate the effect of antilock braking system or locking differential, when driving near or beyond the physical limits of adhesion.

INTRODUCTION

Driving simulators for development, research and driver training are available since many years. In automotive engineering, simulators help examining and improving electronic equipment, e.g. antilock braking system, traction control or vehicle dynamics control (1 - 3). Other simulators are used for various kind of research studies concerning human factors (4, 5). When introducing commercial simulators for driver training a few years ago, the main training objective was focused either on operating the vehicle or on the driver behavior and reactions in traffic (6 - 8). For all these tasks, there was no need for an elaborated vehicle dynamics considering all six degrees of freedom of the vehicle body.



FIGURE 1 Off-road vehicles.

After gaining the first experiences with simulator-based driver training, it was obvious that driving simulators are suitable for other training objectives, namely for driving on rough and uneven terrain, for driving on roads covered with ice or snow and for driver training on emergency vehicles (9 - 11). However, such driving tasks make much higher demands on the vehicle dynamics simulation and the interaction of vehicle and terrain (12). Most of the methods are known from automotive engineering and simulation (12 - 19). Nevertheless, the vehicle model for a commercial driving simulator has to be as simple as possible but always detailed enough to ensure the training goals. It has to consider many transient processes, e.g. found when braking, changing the gear or locking the differential. For an effective driver training on ice, snow or soil, the calculation of the tire forces must be close to the reality. And finally, the vehicle model covering all these aspects must be suitable for real-time applications.



FIGURE 2 Emergency vehicles.

For that reason, Oerlikon Contraves AG has developed a sophisticated generic vehicle model that allows to configure and simulate almost any kind of vehicle. This paper is intended to convey the basic concept of multibody simulation and terrain interaction focusing on applications for cross-country and emergency driver training. For illustration, a heavy cross-country truck is modeled and integrated into the real-time driving simulator *ADAMS* (6). Finally, results from the validation procedure and from special driving maneuvers are presented.

VEHICLE MODELING

Kinematics

The vehicle dynamics is based on a set of mechanical components, e.g. rigid link , revolute joint or prismatic joint, and force elements, e.g. masses, actuator forces, springs and dampers, which can be used to assemble and simulate mechanical systems without the need to derive the corresponding kinematric and dynamic equations by hand. The equations of motion are formulated automatically based on the topology of the system and the kinematics of the mechanical transmission elements. Transmission elements, i.e. rigid links and joints, are linked by means of frames holding position, velocity, acceleration and force data for all six degrees of freedom (Figure 3).



FIGURE 3 Transmission elements.

Each element provides a function f mapping the position data q_A of frame A to the position q_B of frame B. For joints, this mapping considers the actual joint angle j. Most of these transmission functions are very simple. They are determined explicitly on a case-by-case basis. For general transmission elements, the function f determines the position of one or several output frames from a number of input frames and independent joint coordinates. We may write

$$\begin{aligned} \boldsymbol{q}_B &= f(\boldsymbol{q}_A) \\ \dot{\boldsymbol{q}}_B &= \boldsymbol{J}_f \cdot \dot{\boldsymbol{q}}_A \\ \ddot{\boldsymbol{q}}_B &= \boldsymbol{J}_f \cdot \ddot{\boldsymbol{q}}_A + \dot{\boldsymbol{J}}_f \cdot \dot{\boldsymbol{q}}_A \end{aligned} \tag{1}$$

where $J_f = \partial f / \partial q_A$ is the Jacobian matrix. Forces are transmitted in the opposite direction, i.e. from the kinematic output to the kinematic input. Using the transposed Jacobian, it is

$$\boldsymbol{Q}_A = \boldsymbol{J}_f^T \cdot \boldsymbol{Q}_B \,. \tag{2}$$

Composite joints, e.g. the spherical joint, the universal joint or the planar joint, can be defined by linking revolute and prismatic joints. Force elements (Figure 4) do not actually transmit any motion data but pass forces and moments to the frames where they are attached.





Mass elements apply inertia forces and moments to a single frame considering its acceleration. Springs, dampers and actuators are mounted between two frames taking action and reaction forces and moments. The force or moment produced by a spring is calculated from the position of the frames while damper forces are based on their relative velocity. Other force elements represent actuator forces, friction forces or general external forces.

For a tree-structured system composed from rigid links and joints, the function *f* determines the position of all local frames from the position of the root frame and the joint coordinates. In practice, the global kinematics of a compound system is determined by sequentially applying the mappings (Equation 1) of position, velocity and acceleration of all rigid links and joints. Forces are transmitted in the opposite direction starting with the force elements which can be considered as the leaves of the tree-structured system. Figure 5 shows a simple model of a rigid axle with two degrees of freedom.



FIGURE 5 Rigid axle model.

Motion information is transmitted top-down from the chassis frame to the wheel frames. The calculation of forces and moments starts with the force elements, i.e. the springs and dampers, and is propagated bottom-up by the rigid links and the joints to the chassis.

Dynamics

The equations of motion describing the dynamics of a multibody system are automatically generated in real-time from the global kinematics of the system. The approach is based on the solution of the inverse dynamics

$$t = t \left(q, \dot{q}, \ddot{q}, Q_{ex} \right) \tag{3}$$

which determines the motor torques t from the actual joint positions, velocities and accelerations and from external forces Q_{ex} . An efficient solution of the inverse dynamics is based on the recursive Newton-Euler method which was originally used for simulation of serial robots (20). After calculating the global kinematics of the system, inertia and external forces are recursively applied from one element to the next. The equations of motion in minimum order can be written as

$$\boldsymbol{M}(\boldsymbol{q}) \cdot \boldsymbol{\ddot{q}} + \boldsymbol{Q}(\boldsymbol{q}, \boldsymbol{\dot{q}}, \boldsymbol{Q}_{ex}) = 0 \tag{4}$$

with the generalized mass matrix M and the vector Q of the generalized applied forces. The number of equations corresponds to the number of independent joint coordinates in the system, i.e. the number of degrees of freedom. The vector Q is obtained when solving the inverse dynamics (Equation 3) with the generalized accelerations $\ddot{q} = 0$:

$$\boldsymbol{Q} = \boldsymbol{t} \left(\boldsymbol{q}, \dot{\boldsymbol{q}}, \boldsymbol{\theta}, \boldsymbol{Q}_{ex} \right) \,. \tag{5}$$

The i^{th} column of the mass matrix M corresponds to the solution of the inverse dynamics (Equation 3) for a single acceleration $\ddot{q}_i = 1$ while setting all other input accelerations to 0 and ignoring the external applied forces Q_{ex} :

$$\boldsymbol{m}_i = \boldsymbol{t} \left(\boldsymbol{q}, \boldsymbol{\theta}, \boldsymbol{e}_i, \boldsymbol{\theta} \right)$$

(6)

The vector e_i denotes a unit vector with element i = 1. The solution of the linear equations (Equation 4) results in a new set of joint accelerations. The corresponding joint velocities and joint positions are then calculated by means of an appropriate integrator. Adams-Bashforth methods (21) are recommendable integration techniques for real-time systems since they require exactly one solution of the system for each simulation step. Most other integration algorithms are varying the number of system evaluations to optimize stability and precision of the solution. Of course, when using Adams-Bashforth methods, stability has to be guaranteed by an adequate simulation frequency. For more details on the dynamics simulation of multibody systems see (22 - 24). In particular, algorithms are presented that are used to handle closed-loop systems. Typically, the only closed loops of a vehicle model without geometrical bindings between the wheels and the ground are in the wheel suspensions. For these multibody loops, the kinematical equations are formulated and solved locally by means of the *characteristic pair of joints* (22). Within the global tree-structured vehicle model, wheel suspensions can then be considered as kinematical transmission elements with one degree of freedom.

Coulomb Friction

Several vehicle subsystems, e.g. the clutch or the brakes, are based on a Coulomb friction model. For handling friction in multibody systems, the integrator was combined with an event propagation mechanism that locates the time t_{event} of velocity zero-crossings of joints with friction by means of linear interpolation (Figure 6). An integration step is performed for the interval [t_i , t_{event}]. The system can then be reconfigured by locking the corresponding joint and completing the simulation step by integrating over the interval [t_{event} , t_{i+1}].



FIGURE 6 Handling of Coulomb friction.

As soon as the external forces in a joint are higher than the friction forces, the joint is unlocked. Since every event leads to an additional evaluation of the dynamics, the maximum number of accepted events within a simulation step has to be strictly limited in real-time systems.

Tire Forces

An important aspect for high-dynamic driving simulator applications is the calculation of the tire forces which are responsible for accelerating, braking and steering the vehicle. For an efficient driver training, these must be very close to reality. Longitudinal force f_x , lateral force f_y and self-aligning torque m_z are determined according to the

HSRI tire model, which was originally designed for on-road driving (25). Nevertheless, by converting soil parameters to appropriate friction and rolling resistance values, off-road tire forces can be calculated with the same method. Another application of tire forces is the prediction of a vehicle rollover based on the torque m_x with respect to the longitudinal axis.

In order to provide a realistic reaction of the vehicle on steering inputs, the lateral response f_y of the HSRI tire model has to be delayed according to the tire relaxation length L (Figure 7). This can be considered to follow a first order differential equation (26). The solution is described by the recursion

$$f_{y_{i+1}} = f_{y_i} + \frac{\Delta s_x}{L} \cdot (f_{y_{static}} - f_{y_i})$$

$$\tag{7}$$

where $f_{y_{static}}$ is the static solution of the HSRI model and Δs_x is the distance of the tire contact point (TCP) between simulation steps *i* and *i*+1. A similar delay occurs for longitudinal forces: the tire slip is increased and decreased over a certain time period which is essential for antilock braking systems (ABS) control algorithms.



FIGURE 7 Calculation of tire forces.

The HSRI tire model determines the forces from the kinematic state of the TCP, i.e. from the tire slip. When the vehicle is in rest, the tire slip is not defined and another approach has to be used. For small velocities of the TCP, e.g. below 0.1 m/s, the motion of the TCP is integrated and tire forces are calculated by means of a spring-damper model simulating static tire deformations.

Toolbox

For simplifying the process of modeling a vehicle, a toolbox with many vehicle subsystems was developed. These can be assembled in an easy and intuitive way. In general, a model should always be as simple as possible but yet meet all requirements for the corresponding driver training. This principle allows to implement comprehensive vehicle models that are still suitable for real-time systems. Figure 8 shows a selection of the available mechanical and control subsystems. The vehicle components are based on a set of basic transmission, force and control elements. The concept of the toolbox allows to provide different models for the same vehicle subsystem, e.g. different suspension models. According to the training goals or the available computer performance, a simple model or a more sophisticated model can be selected when assembling the vehicle. Developing new subsystem models is easy when following the interface specification for the mechanical or the control components.



FIGURE 8 Toolbox with mechanical components.

In modern vehicles, several electronic control components, e.g. antilock braking system (ABS), traction control system (TCS) or vehicle dynamics control (VDS), are used to facilitate the driving task (27 - 29). Since a driving simulator has to provide appropriate control mechanisms, a set of vehicle control components was developed that can be linked to the dynamics model between the driver input signals, i.e. the steering wheel, pedals and switches, and the corresponding mechanical components. Since the complete vehicle dynamics is available from the simulation, most of the implemented control algorithms are much simpler than in a true vehicle, where many controller input data, e.g. the tire slip or the oversteering angle, have to be estimated from sensor signals.

Example

For illustrating the capabilities of the vehicle toolbox, a cross-country truck with four-wheel drive is modeled. Figure 9 shows the most important components of the vehicle. The engine torque is transmitted to the wheels by means of a power divider and two differentials which can all be locked to improve the traction on uneven terrain. The model has 16 degrees of freedom, six for the motion of the chassis with respect to the inertial frame, each one for every wheel suspension and wheel rotation, one for the gearbox (neutral gear) and one for the clutch.



FIGURE 9 Model of a cross-country truck with four-wheel drive.

Many of the subsystems provide an input from the driver's cabin, e.g. the position of a pedal, lever or switch, that has to be processed by the real-time simulation. The pedal and steering wheel input angles pass a control component that allows remapping of the angle by means of a function or tabular data. ABS control and traction control elements demonstrate the integration of mechanical and control subsystems.

TERRAIN MODELING

Terrain Profile

Elevation and normal direction of the terrain is determined from the visual database either by querying the image generator in real-time or by accessing a virtual database that was prepared offline (*30*). In order to give the driver a realistic information on the type and the roughness of the terrain, a much higher resolution than that of the visual terrain modeling is required.

An efficient approach is based on a random elevation profile that is added to the elevation obtained from the visual database. However, this random pattern has to be $C^{(1)}$ continuous in space and constant in time, i.e. queries at the same position should always return the same elevation. Furthermore, the random elevations shall be normally distributed with zero mean value and a s depending on the type of the terrain.

To fulfill these requirements, a regular grid with spacing *D* for both the *x*- and *y*-direction is defined where a random elevation is calculated for each grid point (x/y). The normal distributed random values are computed according to the *polar method* (31). From two uniform deviates *a* and *b* in [0,1], two random values r_1 and r_2 are defined as

$$r_1 = \sqrt{-2\ln(a)}\cos(2\mathbf{p}b)$$

$$r_2 = \sqrt{-2\ln(a)}\sin(2\mathbf{p}b)$$
(8)

where *a* and *b* can be obtained from a predefined random array *rand* with *n* elements:

$$a = rand[1 + (x \text{ DIV } D) \text{ MOD } n]$$

$$b = rand[1 + (y \text{ DIV } D) \text{ MOD } n]$$
(9)

Elevation and normal direction of the random pattern are then calculated by evaluating a B-spline patch based on the 16 surrounding grid points. Each terrain type is described by a certain grid spacing D and the standard deviation s of the random elevations of the pattern. Table 1 shows the corresponding values used for some terrain types.

Terrain Type	Grid Spacing D [m]	Random Elevation <i>s</i> [m]
Asphalt	4.0	0.02
Meadow	2.0	0.05
Sand	0.8	0.06
Gravel	0.3	0.08

 TABLE 1
 Random elevation profiles for some terrain types

True terrain and road profile data are obtained by measurement (32, 33). Usually, mean spatial frequency and amplitude values of a road or terrain profile may be within a wide range even for the same type of ground. The values given in Table 1 are deduced from power spectral density data (33) and from road profiling data (34) according to the *International Roughness Index* (IRI).

Ground Properties

When driving on an elastic terrain, e.g. clay, sand or snow, the longitudinal and lateral forces are limited by the terrain properties. When a certain maximum force is reached, the soil starts to fail. The maximum shear stress t_{max} is determined according to the Mohr-Coulomb criterion (33)

$$\boldsymbol{t}_{\max} = c + f_n \cdot \tan \boldsymbol{f}$$

where f_n is the normal stress, c the cohesion of the soil and $\tan f$ the internal shearing resistance angle, i.e. a soil property representing the internal friction between the particles. The shear stress in a point of the contact patch

(10)

depends on the particular shear displacement s_D of that point, i.e. the distance between the point from its location on the unloaded tire.

The relationship of shear displacement and shear stress is determined from measurements (33, 35). It can be approximated with two linear segments, one for the increasing shear stress and one for constant shear stress when the soil fails (Figure 10).



Shear Displacement $s_{D}[m]$

FIGURE 10 Relationship of shear displacement and shear stress.

K is referred to as the shear deformation modulus. It represents the shear deformation corresponding to the maximum shear stress. For the approximation of the shear displacement, we may write

$$\mathbf{t} = \begin{cases} \frac{\mathbf{t}_{\max} \cdot s_D}{K} & s_D < K \\ \mathbf{t}_{\max} & s_D \ge K \end{cases}$$
(11)

According to the procedure used for the HSRI tire model (see above), the contact region of the tire is divided into an area of tire deformation ($s_D < K$) and an area of soil failure ($s_D \ge K$). The total longitudinal and lateral tire force is then determined by integrating Equation 11 over the whole contact region. Table 2 shows the characteristic ground parameters for snow and several kinds of soil.

Terrain type	c [kPa]	$\tan f$ [deg]	<i>K</i> [m]
Dry sand	1.04	28	0.025
Heavy clay	68.95	34	0.006
Loam	3.1	29.8	0.01
Snow	6	20.7	0.04

 TABLE 2
 Ground parameters for some terrain types 1

reference: (33)

For soft soils, the tire also sinks into terrain. The implemented tire model considers sinkage by increasing the rolling resistance. Any other forces resulting from sinkage, e.g. the increase of the steering forces, are not yet implemented.

IMPLEMENTATION

The vehicle components and the algorithms used to solve the system dynamics were implemented with the objectoriented programming language C++. The approach of abstract base classes and virtual methods complies all requirements of a modular, open and extendable toolbox. Most of the subsystem models represent a simplification of the corresponding component of the true vehicle. This makes it possible that even complicate vehicle models with many degrees of freedom are suitable for real-time applications. Nevertheless, all vehicle models are able to simulate the characteristic behavior of the true vehicle. Thus, they always ensure the training goals.

The overall topology of a vehicle model is typically tree-structured where the vehicle body is the root element and the wheels are the leaves. The motion of the vehicle body with respect to the inertial frame is described by each three prismatic and revolute joints. In such systems, considerable optimizations are possible for the calculation of the mass matrix. When setting a pseudo-acceleration to a joint, only the subtree starting with that joint has to be recalculated while all other accelerations keep unchanged.

Usually, multibody dynamic systems are integrated with adaptive step size control or with predictor-corrector methods. These integrators are varying the number of system evaluations to optimize stability and precision of the solution. They can be hardly adapted for real-time applications. For that reason, an Adams-Bashforth method (21) was used which requires exactly one solution of the system for each simulation step. Since the powertrain components and the tires have much higher eigenfrequencies than the chassis and the suspensions, it is indicated to use a higher integration frequency for these vehicle subsystems. However, high and low frequency parts of the vehicle are coupled through the wheels and the tires. Multi-rate integration was applied to decouple and integrate the systems with different frequencies. For the vehicle chassis and the suspensions, an update frequency of 60 Hz was used while the control components, the powertrain and the tire models were calculated with 300 Hz.

SIMULATION

For examining the vehicle dynamics simulation, the different models were integrated into the *ADAMS* driving simulator (6). The system was equipped with either a chassis of a true passenger car or a replica of a truck cabin (see Figure 11).



FIGURE 11 ADAMS driving simulator for police car (left) and cross-country truck with a six DOF motion platform (right).

The computer generated images were projected through three front channels with a total angle of 180 degrees and two rear-view mirrors. The virtual environment database comprised several classes of roads including a freeway and an off-road driving area with different ground types, a steep hill and a river with a low water passage. For the truck cabin, a motion platform with six degrees of freedom (DOF) was available.

The simulation computer was a *PowerPC* 750 with 400 MHz running the real-time operating system *VxWorks*. The base simulation frequency was 60 Hz. Each simulation cycle comprised an input phase reading pedals, levers and switches in the driver's cabin, a simulation phase calculating the vehicle dynamics, the generic traffic and performing the trainee assessment, and finally an output phase controlling the cabin instruments, the steering wheel control loader, the image generator and the motion system.

RESULTS

Validation

To validate the vehicle toolbox and the algorithms implemented to solve the dynamics, results from the simulation of a heavy two-axle truck were compared with acceleration measurements of a true vehicle. A special measuring device with several linear acceleration sensors was installed on the assistant driver's seat. From these measurements, all linear and rotational accelerations of the cabin were calculated with respect to a coordinate system located in the driver's eye point.

Figure 12 shows the longitudinal acceleration for a full braking maneuver with ABS control on dry asphalt. The initial speed was 60 km/h. The time from the initial speed to full rest is about the same for both the measurement and the simulation (~ 3.8 s) and even the subsequent cabin oscillations of the simulation are close to the measurements.



FIGURE 12 Longitudinal acceleration for full braking maneuver with ABS control.

The lateral dynamics was verified by means of a full braking maneuver with ABS control while driving a curve with maximum steering angle. The initial speed was 10 km/h and the radius of the path of the cabin was approximately 7 m. Figure 13 depicts the lateral acceleration. Obviously, the damping characteristics of the suspensions of the model and the true vehicle are very similar.



FIGURE 13 Lateral acceleration for full braking maneuver with ABS control while driving a curve.

Of course, for some maneuvers, it is difficult to compare measurement and simulation since the on-road or off-road conditions and the driver actions cannot be reproduced very precisely with the simulation. Nevertheless, the results from the validation procedure indicate that the vehicle models assembled from components of the generic toolkit show the same characteristic behavior as the true vehicles.

Differential

When driving off-road or on roads covered with ice or snow, the differential has a considerable influence on the handling characteristic of a vehicle. The differential uniformly distributes the driving torque among the left and right wheel of the driving axle. At the same time, it allows different rotational velocities of the wheels when driving in curves. However, the differential has a major disadvantage. Its total driving torque is given by the wheel with the lower friction coefficient. To overcome this problem, differentials can either be locked or self-locking differentials are used that produce an equalizing torque between the wheel axles. The most common techniques are the torque-sensing (TORSEN) and the viscous type differential. For details on the design of differentials, the reader is encouraged to see reference (*36*).



FIGURE 14 Wheel speed when accelerating on **m**split (asphalt / ice): unlocked differential (left) and self-locking differential (right).

The vehicle toolkit comprises models for various kinds of differentials. To illustrate the driving characteristics of these components, a traditional differential was compared with a viscous self-locking differential that produces an equalizing torque based on the relative velocity of the wheels. For examining the functionality of the differential, the

vehicle model was accelerated on a **m**-split ground with the left wheel on ice and the right wheel on dry asphalt. Figure 15 shows the rotational speed of the left and the right wheel. At the beginning, the left wheel starts skidding with both differentials. While the self-locking differential is equalizing the motion of the wheels, the unlocked differential is hardly slowing down the skidding wheel.

Even more convincing than the speed of the wheels is the resulting acceleration of the vehicle which corresponds to the total traction force that can be applied to the ground. Figure 16 depicts the vehicle speed for a locked differential, for a self-locking viscous differential and for an unlocked differential on a **m**-split ground.



FIGURE 15 Vehicle speed when accelerating on msplit (asphalt / ice).

The best acceleration is achieved with the locked differential. Indeed, it is the same as with both wheels on dry asphalt, provided that the total driving force can be passed to the ground with only one wheel. With the unlocked differential, both wheels are skidding and the resulting vehicle acceleration is very poor.

Time Measurements

In real-time systems, it is of particular importance to know the computation time of a software module for the typical case as well as for the worst case. To determine the computational complexity of vehicle models based on the generic toolkit, different vehicles were examined. The simulation time was measured for a base cycle comprising one integration step of the chassis and the suspensions and five integration steps for the control components, the powertrain and the tires (multi-rate integration). For a two-axle vehicle with one driven axle, the simulation time was 1.59 ms, for a semitrailer truck with five axles it was 2.72 ms. The worst case arises when a transient process caused by friction has to be handled. Then, the simulation time is doubled since an additional evaluation of the system is required.

CONCULUSION

Vehicle Modeling

Several different vehicle models, e.g. passenger cars, heavy trucks with semitrailer or full trailer and light-armored vehicles, were model and integrated into the *ADAMS* driving simulator. The toolbox presented in the previous sections enabled fast prototyping and simulation of the vehicles without detailed knowledge on multibody dynamics. The simulation is suitable for real-time applications and the computation time is quite short even for vehicles with many axles. Acceleration measurements and reports from professional drivers have shown that the vehicle simulations are close to reality and suitable for the corresponding driver training.

Simulation

A study with a high-speed police car simulator (10) has shown that a motion system providing cues on the longitudinal and lateral acceleration of the vehicle is indispensable for a driving simulator. Without any motion system at all, braking or driving through curves is very difficult since the only information on the vehicle motion is obtained from the computer-generated images.

For on-road driving, a seat motion system with two degrees of freedom provided the best results since the acceleration cues can be built up much faster than with a large system moving the whole cabin. Typical driving scenarios comprise sequences of accelerating and braking or left and right turns. It is not possible to generate the corresponding motion cues with a classical six degrees of freedom motion system without an intermediate phase of wrong cues. In research, motion systems with an additional x-y carriage are used to overcome this problem.

However, for cross-country vehicles, the motion system has to represent rather the slope and the roughness of the ground than the accelerations resulting from the vehicle dynamics. Here, a conventional motion platform where the cabin is mounted on a hexapod perfectly fulfills the requirements.

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REFERENCES

- 1. Bernasch, J. and S. Haenel. The BMW Driving Simulator used for the Development of a Driver-biased Adaptive Cruise Control. *Proc. DSC'95 Conference*, Sophia Antipolis, France, 1995.
- 2. Grezlikowski, H. and V. Schill. With Low Cost to High End a PC approach at the DaimlerChrysler Driving Simulator. *Proc. DSC2000 Conference* (CD-ROM), Paris, France, 2000.
- 3. Reymond, G., A. Heidet, M. Canry and A. Kemeny. Validation of Renault's Dynamic Simulator for Adaptive Cruise Control Experiments. *Proc. DSC2000 Conference* (CD-ROM), Paris, France, 2000.
- 4. O. Olofinboba. A Unique High-fidelity Wrap-around Driving Simulator for Human Factors Research Applications. *Proc. SPIE*, Vol. 2740, 1996, pp. 52-59.
- 5. Bailey, A. C., A. H. Jamson, A. M. Parkes and S. Wright. Recent and Future Development of the Leeds Driving Simulator. *Proc. DSC'99 Conference* (CD-ROM), Paris, France, 1999.
- 6. Thöni, U. A. and H. Düringer. ADAMS: An Advanced Driving and Maneuvering Simulator For a Variety of Training Needs. *Proc. 17th I/ITSEC Conference* (CD-ROM), Orlando, FL, 1995.
- 7. Deister, M. Driving Simulators for Trucks and Busses. *Proc. '99 I/ITSEC Conference* (CD-ROM), Orlando, FL, 1999.
- 8. Filipo, A.: TRUST: the Truck Simulator for Training. *Proc. DSC2000 Conference* (CD-ROM), Paris, France, 2000.
- 9. Tomaske, W. A. Modular Automobile Road Simulator *MARS* for on and off-Road Conditions. *Proc. DSC'99 Conference* (CD-ROM), Paris, France, 1999.
- 10. Thöni, U. A. Design and Trials of a High-Speed Police Car Simulator. *Proc.* 12th ITEC Conference (CD-ROM), Lille, France, 2001.
- 11. Welles, R. T. and M. Holdsworth. Tactical Driver Training using Simulation. *Proc. 2000 I/ITSEC Conference* (CD-ROM), Orlando, FL, 2000.

- 12. Allen, R. W. and T. J. Rosenthal. *Requirements for Vehicle Dynamics Simulation Models*. SAE-Paper 940175, 1994.
- 13. Godbole, D. and S. Karahan. Automotive Powertrain Modeling, Simulation and Control Using Integrated System's CASE Tools. SAE-Paper 940180, 1994.
- 14. Sayers, M. W. and D. Han. A Generic Multibody Vehicle Model for Simulating Handling and Braking. *Journal* of Vehicle System Dynamics, Supplement 25, 1996, pp. 599-613.
- 15. Sayers, M. W. and M. Riley. *Modeling Assmptions for Realistic Multibody Simulations of the Yaw and Roll Behavior of Heavy Trucks*. SAE-Paper 960173, 1996.
- 16. Allen, R. W., J. Rosenthal and J. R. Hogue. *Modeling and Simulation of Driver/Vehicle Interaction*. SAE-Paper 960177, 1996.
- Zanten, A. T., R. Erhardt, A. L. Lutz, W. Neuwald and H. Bartels. Simulation for the Development of the Bosch-VDC (SAE-Paper 960486), *Investigations and Analysis in Vehicle Dynamics and Simulation*, SAE SP-1141, 1996, pp. 151-161.
- Plöchl, M. and P. Lugner. Braking Bahaviour of a 4-Wheel-Steered Automobile with an Antilock Braking System. *Vehicle System Dynamics*, Supplement 25, 1996, pp. 547-558.
- 19. Schröder, C., K.-U. Köhne and T. Küppers. Fahrdynamiksimulation schwerer Nutzfahrzeuge. *Automobiltechnische Zeitschrift ATZ*, Vol. 99, No. 6, 1997, pp. 322-328.
- 20. Walker, M. W. and D. E. Orin. Efficient Dynamic Computer Simulation of Robotic Mechanisms. ASME Journal of Dynamic Systems, Measurements and Control, Vol. 104, 1982, pp. 205-211.
- 21. Eich-Soellner, E. and C. Führer. Numerical Methods in Multibody Dynamics. Teubner, Stuttgart, 1998.
- 22. Hiller, M. Multiloop kinematic chains. *Kinematics and Dynamics of Multi-Body Systems*, Eds. J. Angeles and A. Kecskeméthy, CISM Courses and Lectures No. 360, Springer-Verlag, Wien, New York, 1995, pp. 75-165.
- 23. Feretti, G. Systematic Dynamic Modelling of Mechanical Systems Containing Kinematic Loops. *Journal of Mathematical Modelling of Systems*, Vol. 2, No. 3, 1996, pp. 212-235.
- 24. Eberhard, P. and U. Neerpasch. Interactive Modelling of Multibody Systems with an Object Oriented Data Model. *Journal of Mathematical Modelling of Systems*, Vol. 2, No. 1, 1996, pp. 55-68.
- 25. Dugoff, H., P. S. Fancher and L. Segel. An Analysis of Tire Traction Properties and their Influence on Vehicle Dynamics Performance. SAE-Paper 700377, 1970.
- 26. Crolla, D. A. and A. S. A. El-Raza. A Review of the Combined Lateral and Longitudinal Force Generation of Tyres on Deformable Surfaces. *Journal of Terramechanics*, Vol. 24, No. 3, 1987, pp. 199-225.
- Tomizuka, M. and J. K. Hedrick. Advanced Control Methods for Automotive Applications. *Vehicle System Dynamics*, Vol. 24, 1995, pp. 449-468.
- 28. Leffler, H., R. Auffhammer, R. Heyken and H. Röth. *New Driving Stability Control System with Reduced Technical Effort for Compact and Medium Class Passenger Cars.* SAE-Paper 980234, 1998.
- 29. Tseng, H. E., B. Ashrafi, D. Madau, T. Allen Brown and D. Recker. The Development of vehicle stability control at Ford. *IEEE/ASME Transactions on Mechatronics*, Vol. 4, No. 3, Sept. 1999, pp. 223-234.
- Papelis, Y., S.Allen and B.Wehrle. Automatic Correlated Terrain Database Generation and Management for Ground Vehicle Simulators. *Modeling and Simulation Technologies Conference*. Amer. Institute of Aeronautics and Astronautics. AIAA paper 99-4197, 1999.

- 31. Knuth, D. E. *The Art of Computer Programming*. 2nd Edition, Vol. 2: Seminumerical Algorithms. Addison-Wesley, 1981.
- 32. Laib, L. Measurement and Mathematical Analysis of Agricultural Terrain and Road Profiles. *Journal of Terramechanics*, Vol. 14, No. 2, 1997, pp. 83-97.
- 33. Wong, J. Y. Theory of Ground Vehicles. John Wiley and Sons, Inc. New York, 1993.
- 34. Sayers, M. W. and S. M. Karamihas. *The Little Book of Profiling*. Online: <u>http://www.umtri.umich.edu/erd/roughness/litbook.html</u>, University of Michigan, 1996.
- 35. Schmid, I. Interaction of Vehicle and Terrain Results from 10 Years Research at IKK. *Journal of Terramechanics*, Vol. 32, No. 1, 1995, pp. 3-26.
- 36. Lechner, G. and H. Naunheimer. Fahrzeuggetriebe, Springer-Verlag. Berlin, Heidelberg, 1994.