

Flexible Real-Time Simulation of Truck and Trailer Configurations

2011-01-2264

Published
09/13/2011Michael Peperhowe, Kusnadi Liem and Hagen Haupt
dSPACE GmbH

Copyright © 2011 SAE International

doi:10.4271/2011-01-2264

ABSTRACT

Real-time simulation of truck and trailer combinations can be applied to hardware-in-the-loop (HIL) systems for developing and testing electronic control units (ECUs). The large number of configuration variations in vehicle and axle types requires the simulation model to be adjustable in a wide range. This paper presents a modular multibody approach for the vehicle dynamics simulation of single track configurations and truck-and-trailer combinations. The equations of motion are expressed by a new formula which is a combination of Jourdain's principle and the articulated body algorithm. With the proposed algorithm, a robust model is achieved that is numerically stable even at handling limits. Moreover, the presented approach is suitable for modular modeling and has been successfully implemented as a basis for various system definitions. As a result, only one simulation model is needed for a large variety of track and trailer types. The number of simulated axles of the track and the trailer can be changed during run time without model adaptations. In addition, the same model can be used to simulate single track configurations as well as truck-and-trailer combinations. The model's modularity makes it useful for the whole development process of electronic stability control (ESC): from model-based-design (MBD) to software-in-the-loop (SIL) and finally to HIL systems.

INTRODUCTION

HIL systems have become a state-of-the-art technology for developing and testing ECUs in the automotive field [20]. The virtual testing of controller functionality is increasing rapidly, especially for track applications. Due to higher numbers of axles, gear types, suspension systems, and trailer combinations, efficient variant handling is indispensable [5]. One truck may contain over 30 networked ECUs. Due to the

complexity on the controller side and the large variety of truck and trailer configurations on the plant side, repeatable test sequences are necessary to ensure test coverage in all combinations. This requires flexible dynamic simulation models which can easily be adapted to these needs. If a model is configurable to different variants just by parameterization rather than by changing the model structure, loops can be created to run the same tests for all variants automatically [4].

There are some general requirements that need to be fulfilled by the models used in HIL systems. First, they have to be real-time-capable on the selected HIL system hardware. This means that the reading of actuator signals, the model calculations themselves, and the output of sensor signals have to be performed in one simulation step. In addition to having real-time capability, the model for an HIL system must be able to generate physically correct, consistent outputs (sensor signals) according to the actual inputs (actuator signals) [14].

In truck-trailer simulation, one main focus is on the hitch coupling. Several approaches to modeling the coupling between the track and trailer, such as a spring-damper system [18], DAE formalism [4, 1, 18], minimal coordinates using Jourdain's principle, etc., [17], Kane's method [10] and d'Alembert's principle [24], are suitable for generating truck-trailer system models. However, each of these approaches has a certain drawback. [14] outlines the advantages and disadvantages of these approaches.

In this paper the mathematical models of truck and trailer are generated independently using Jourdain's principle of virtual power. As the truck and trailer have similar mechanical topologies, modeling their systems leads to similar implementations. The equations of motion are decomposed so that the axles are configurable without changing the model structure. Then the truck-trailer coupling is formulated using

the articulated body algorithm [7]. The combination of these two formulations leads to an efficient, robust implementation which is flexible and capable of real-time simulation. This algorithm ensures high numerical stability of the hitch simulation for different driving maneuvers, especially at the handling limits.

MODULAR MODELING

For vehicle dynamics simulation, the truck and trailer mechanics are realized as multibody systems. The truck and trailer each consist of a sprung mass (body), and the axles (components) are represented by additional unsprung masses (Figure 1).



Figure 1. Truck and trailer bodies and components

MINIMAL COORDINATES

To generate the equations of motion in minimal coordinates, n independent generalized velocities:

$$\mathbf{u} = [u_1, u_2, \dots, u_n]^T, \quad (1)$$

have to be chosen, corresponding to the number of DOFs in the system. Applying the principle of virtual power, the equations of motion in minimal coordinates $\mathbf{q}$ can then be written as:

$$\begin{aligned} \mathbf{V}(\mathbf{q}) \cdot \dot{\mathbf{q}} &= \mathbf{u} \\ \mathbf{M}(\mathbf{q}) \cdot \ddot{\mathbf{q}} &= \mathbf{Q}(\mathbf{q}, \mathbf{u}), \end{aligned} \quad (2)$$

where $\mathbf{V}$ is a so-called Hamel matrix [9] and defines the kinematical correlation which maps the first time derivative of the generalized coordinates to the generalized velocities, $\mathbf{M}$ is the generalized mass matrix, and $\mathbf{Q}$ the vector of generalized forces. The equations of motion for the truck and the trailer are built in the same way.

DECOMPOSITION OF EQUATION OF MOTION

The mass matrix $\mathbf{M}$ and the force vector $\mathbf{Q}$ can be divided into body (subscript “B”) and axle (component, subscript “C”) related parts (Figure 1):

$$\begin{bmatrix} \mathbf{M}_B & \mathbf{M}_{BC} \\ \mathbf{M}_{BC}^T & \mathbf{M}_C \end{bmatrix} \cdot \begin{bmatrix} \dot{\mathbf{u}}_B \\ \dot{\mathbf{u}}_C \end{bmatrix} = \begin{bmatrix} \mathbf{Q}_B \\ \mathbf{Q}_C \end{bmatrix}. \quad (3)$$

Expanding and inserting the equations from (3) into each other yields to two separate equations that can be solved successively:

$$\tilde{\mathbf{M}} \cdot \ddot{\mathbf{u}}_B = \tilde{\mathbf{Q}} \quad (4)$$

$$\ddot{\mathbf{u}}_C = \mathbf{M}_C^{-1} \cdot (\mathbf{Q}_C - \mathbf{M}_{BC}^T \cdot \ddot{\mathbf{u}}_B), \quad (5)$$

where

$$\tilde{\mathbf{M}} = \mathbf{M}_B - \mathbf{M}_{BC} \cdot \mathbf{M}_C^{-1} \cdot \mathbf{M}_{BC}^T, \quad (6)$$

$$\tilde{\mathbf{Q}} = \mathbf{Q}_B - \mathbf{M}_{BC} \cdot \mathbf{M}_C^{-1} \cdot \mathbf{Q}_C. \quad (7)$$

ARTICULATED BODY ALGORITHM

The articulated body algorithm (ABA) is a propagation method for solving forwards dynamics problems [7]. The algorithm can be divided into two major steps: backward assembly and forward acceleration. Figure 2 shows the schematics of the ABA for an open kinematic tree consisting of three bodies, which are connected by joints. The mass matrices and force vectors are represented by $\mathbf{M}$ and $\mathbf{Q}$, respectively, and the acceleration vector $\mathbf{a}$ denotes either the accelerations of the bodies ($\mathbf{a}_j$) or the relative accelerations between two bodies ($\mathbf{a}_{ji}$)

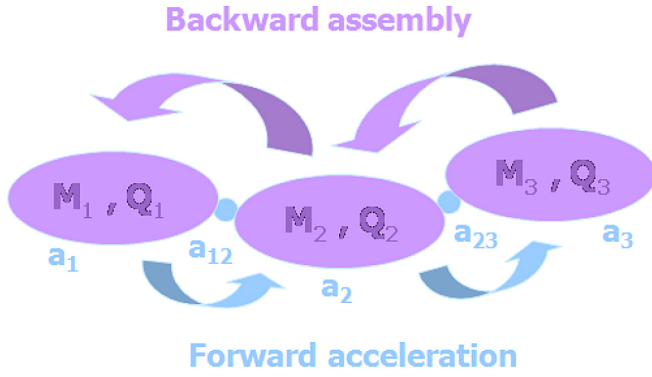


Figure 2. Articulated body algorithm

Starting at the last body, articulated mass matrices and force vectors can each be obtained from the successors as follows:

$$\begin{aligned} \mathbf{M}_3^A &= \mathbf{M}_3 \\ \mathbf{M}_2^A &= \mathbf{M}_2 + f_1(\mathbf{M}_3^A) \\ \mathbf{M}_1^A &= \mathbf{M}_1 + f_2(\mathbf{M}_2^A) \end{aligned} \quad (8)$$

$$\begin{aligned} \mathbf{Q}_3^A &= \mathbf{Q}_3 \\ \mathbf{Q}_2^A &= \mathbf{Q}_2 + f_3(\mathbf{M}_3^A) \\ \mathbf{Q}_1^A &= \mathbf{Q}_1 + f_4(\mathbf{M}_2^A) \end{aligned} \quad (9)$$

where f_i represents the transformations according to the joint type between body and successor. The result of backward assembly is an articulated body equation of motion. When the first body is reached, the acceleration of the first body can be derived by matrix inversion of [equation \(10\)](#):

$$\mathbf{M}_1^A \cdot \mathbf{a}_1 = \mathbf{Q}_1^A. \quad (10)$$

The articulated body equation of motion describes the acceleration response of a single rigid body, in which the dynamic effects of a specific set of other bodies and joints are taken into account. Articulated body masses depend only on the masses of the member bodies and the constraints that are imposed by the member joints. In the forward acceleration, the accelerations of the members are calculated one by one by relative kinematics.

RELATIVE KINEMATICS

To combine the independent equation sets of the truck and the trailer ([equation \(2\)](#)), body-fixed reference coordinate systems “ $\mathbf{K}_V$ ” and “ $\mathbf{K}_T$ ” (subscript “V” denotes the vehicle, i.e. truck, and “T” denotes the trailer) are attached to the truck and trailer body, respectively. The global frame of reference is referred to as “ $\mathbf{K}_E$ ”. The vehicle reference frame lies at the

center point of the front axle, and the trailer reference frame is fixed to the hitch point of the trailer as shown in [Figure 3](#).

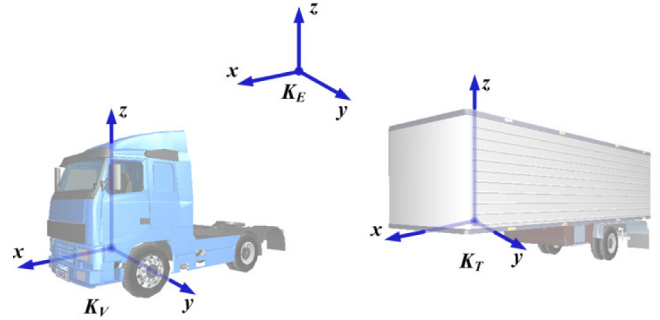


Figure 3. Truck and trailer reference coordinate system.

The hitch point is modeled as a ball joint, and the relative kinematics of the hitch position are introduced. The position vector of the hitch point with respect to the global reference system can be written as:

$${}^E \mathbf{r}_T = {}^E \mathbf{r}_V + {}^E \mathbf{R}_V \cdot {}^V \mathbf{r}_{VT}, \quad (11)$$

where ${}^E \mathbf{r}_T$ is the position vector of the trailer hitch point, ${}^E \mathbf{r}_V$ the position vector of the vehicle, ${}^E \mathbf{r}_{VT}$ the position vector of the hitch point relative to the vehicle reference system, and ${}^E \mathbf{R}_V$ the orthogonal transformation matrix which transforms vectors from coordinate system “ $\mathbf{K}_V$ ” to “ $\mathbf{K}_E$ ”. The index in the left upper corner of a vector defines the coordinate system used.

The velocity relationship can be obtained by differentiating the position [equation \(11\)](#) with respect to time. The relative kinematics of the trailer's linear and angular velocities can be written as:

$$\begin{aligned} {}^V \mathbf{v}_T &= {}^V \mathbf{v}_V + {}^V \boldsymbol{\omega}_V \times {}^V \mathbf{r}_{VT}, \\ {}^V \boldsymbol{\omega}_T &= {}^V \boldsymbol{\omega}_V + {}^V \mathbf{R}_T \cdot {}^V \boldsymbol{\omega}_{VT}, \end{aligned} \quad (12)$$

where ${}^V \boldsymbol{\omega}_V$, ${}^V \boldsymbol{\omega}_T$ are the angular velocity of the vehicle and trailer, respectively, and ${}^V \boldsymbol{\omega}_{VT}$ is the relative angular velocity between the vehicle and trailer. The trailer's absolute linear and angular accelerations $\mathbf{a}_T$, $\boldsymbol{\alpha}_T$ can be expressed in the vehicle reference system or in the trailer reference system. With respect to the vehicle system, they are

$${}^V \mathbf{a}_T = {}^V \boldsymbol{\omega}_V \times {}^V \mathbf{v}_T + {}^V \dot{\mathbf{v}}_V + {}^V \dot{\boldsymbol{\omega}}_V \times {}^V \mathbf{r}_{VT}, \quad (13)$$