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# TRUCK BOND GRAPH MODELLING INTEGRATING BOTH MULTIBODY AND FUNCTIONAL APPROACHES

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## ABSTRACT

This paper deals with the construction of a 6-wheel heavy-duty truck chassis multibond graph model. The novelty of this work resides in the combination of both multibody and functional approaches to build the bond graph representation. The relevance of this combined approach is assessed by numerical simulation results comparison w.r.t. a reference multibody model.

**Keywords:** Bond graph, Vehicle dynamics, Functional modelling, Multibody modelling.

## 1 INTRODUCTION

The heavy-duty truck industry is undergoing significant technological developments. These developments aim at improving truck performances in terms of vehicle dynamics behaviour, safety, and energy efficiency leading to disruptive innovations. Functions that used to be treated separately, such as steering, braking systems, suspension, power train and engine, must now be more and more treated in synergy. Therefore, improving vehicle design requires a deep understanding of its architecture, its various systems and their complex interconnections. In particular, the chassis, to which all vehicle subsystems are connected, is of central interest.

This paper addresses the construction of a six wheels heavy-duty truck model. The truck has a central drive axle and a steered front axle. It has 3D motions and the different parts are considered as rigid bodies. The model of the link between the truck body (sprung mass) and the spindle (unsprung mass) is based on a functional approach,

leading to a domain knowledge model (namely vehicle dynamics). It consists of a mathematical model derived from the kinematical relationships between some key velocities and the degrees of freedom. This differs from the “organic” approach used in generic multi-body modelling, which considers all the bodies and kinematic joints constituting the chassis [1].

The modelling is also based on the bond graph language, and more precisely the multibond graph representation, especially suited for multibody mechanical systems [2]. While detailed concepts of bond graphs are not covered here, readers are directed to [3] for further insights. Numerous studies have explored bond graph modelling in this context, yielding models of various complexities such as the quarter-vehicle [4], bicycle models [5] [6], and the full four-wheel model [7]. The model, offering a versatile approach, can be adapted for various purposes and can be simplified to an intermediate model while preserving its functional aspects.

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The novelty of this work resides in the combination of both multibody and functional approaches to build the bond graph representation. To a large extent, the functional approach falls into an analytical approach of the bond graph construction of multibody system [3].

The paper is organized as follows. Section 2 introduces the chassis model, its bodies, as well as an analysis of the overall energy structure using a word graph bond representation. Section 3 details the functional approach and its integration into the bond graph formalism. The overall bond graph representation is then presented. Section 4 presents a comparison of the numerical simulation results obtained with MS1 software for the bond graph model and the results of a reference multibody model. From an effort viewpoint, section 5 interprets the bond graph model, contextualizing the results in relation to some phenomena contributing to the vehicle dynamics, such as efforts feedback via the axle. The conclusion is given in the last section.

## 2 SYSTEM ANALYSIS

### 2.1 Chassis description

The truck chassis is a multibody system containing several components, namely the body, spindles, wheels and all the mechanical joints between these elements. In the terminology used in the document, the truck body refers to the sprung mass of the chassis and the spindle (by misuse of language, the term "spindle" refers not only to the actual solid spindle, but is used to designate the entire unsprung non-rotating steering mass) to the unsprung mass. Some of these components are simply duplicated in structural terms, but using different parameters. The study uses a Galilean reference frame  $\mathcal{R}_0$  as the absolute frame (figure 1a). The truck body  $S_1$  is a rigid body with a local reference frame  $\mathcal{R}_1$  at its center of mass  $G$ , featuring mass matrix  $[M_1]$  and inertia tensor  $[\bar{I}_{G,S_1}]$ . The spindle  $S_2$  has a reference frame  $\mathcal{R}_2$  attached to the wheel center  $A_2$ , with mass matrix  $[M_2]$  and inertia tensor  $[\bar{I}_{A_2,S_2}]$ . The orientation of  $\mathcal{R}_2$  relative to  $\mathcal{R}_1$  is determined by an Euler decomposition of type ZXY, steer/camber/self-rotating, with the corresponding

angles  $\delta_2$ ,  $\varepsilon_2$  and  $\eta_2$ . The wheel  $S_3$  is connected to the spindle  $S_2$  via a revolute joint, and its reference frame is  $\mathcal{R}_3$  attached to  $A_2$ . The wheel has mass matrix  $[M_3]$  and the inertia tensor is  $[\bar{I}_{A_2,S_3}]$ . The wheel/ground contact point is denoted  $B_3$ . In our study, road interaction efforts, denoted as  $F_{tire}^2$  and  $T_{tire}^2$ , are applied at the wheel center  $A_2$  and expressed in  $\mathcal{R}_2$ . The steering system considered is an hybrid electro-hydraulic power steering (EHPS) system [8].

### 2.2 General hypothesis

The modelling is based on the following assumptions:

- The wheel center  $A_2$  serves as the center of gravity for the subsystem  $S_2 + S_3$ ;
- The chassis model permits the representation of axle kinematics, influenced by wheel travel (current and opposite) and the rotation of the steering box output axis for the front axle;
- Elasto-kinematic effects and body torsion are not considered;
- All the geometric and inertial quantities associated with the spindle take into account all the elements actually connected to it.

### 2.3 Word bond graph representation

Figure 1b shows a word bond graph representation of the truck model. The figure exhibits the word truck body with six multibonds coupling it to the "spindle+wheel" words via the wheel center velocities. The steering system is linked to both front "spindle+wheel" words via a single power bond, representing steering interaction. The words "spindle+wheel" for each axle are connected to each other via two opposite single power bonds, linking the kinematics of each half-axle. Figure 1b shows the steering and transmission sub-systems, while other sub-systems are part of the "spindle+wheel" word. For sake of clarity, signal bonds between different words are omitted in this word bond graph representation.

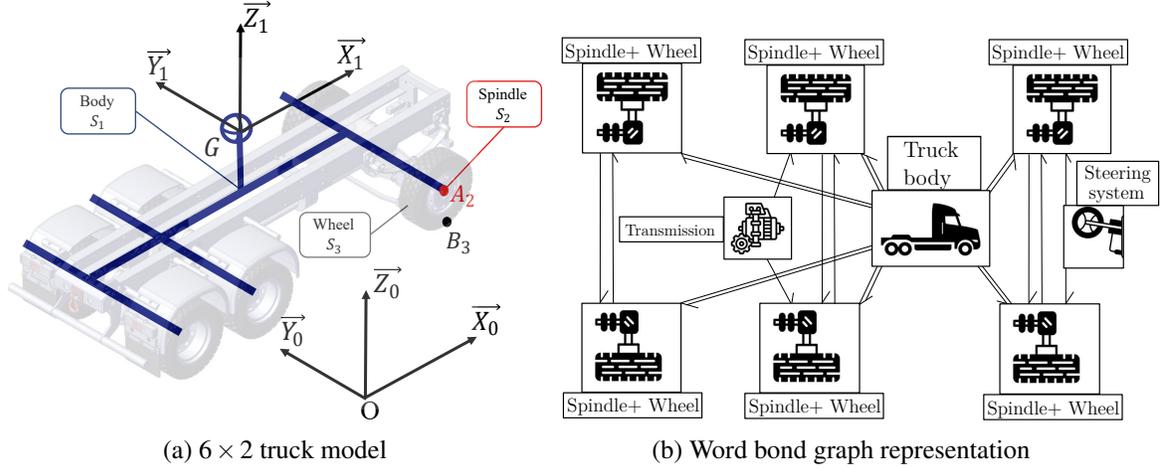


Figure 1: Truck chassis

### 3 CHASSIS BOND GRAPH MODELLING

This section focuses on establishing the body/spindle link by kinematic corrections for wheel center and rim orientation. We will then introduce a bond graph model for a steering half-axle within the chassis functional modelling framework. The calculations and representations primarily concentrate on the left front half-axle of the truck, and are analogous for the other half-axes.

#### 3.1 Chassis degrees of freedom

The system degrees of freedom (DOFs) are determined by the number of bodies and the nature of joints determining for kinematic constraints. The specific joint between the body and each spindle element constraints 5 mobilities to depend solely on a unique DOF, here chosen as the vertical wheel travel (denoted  $z$ ). The complete relative motion between the truck body and a spindle is modelled with a functional representation that depends on the current and opposite vertical wheel travels,  $z$  and  $z_{opp}$  respectively (this allows to define the interdependence of the left and right suspensions), and the wheel rotation about the vertical axis (namely the steering reflected directly by the output  $\alpha$  of the steering box). For a non-steered axle, the kinematic link only depends on the current and opposite vertical wheel travels.

The joints between the wheel and the spindle, as well as between the output axis of the steering box and the truck body are revolute joints. These considerations lead to a global multibody chassis model with 19 DOFs,

$$\left\{ \begin{array}{l} \text{Truck body (1 body) : 6 DOF} \\ \text{Spindles (6 bodies) : } 6 \times 1 \text{ DOF} \\ \text{Wheels (6 bodies) : } 6 \times 1 \text{ DOF} \\ \text{Steering box body (output axis) : 1 DOF} \end{array} \right.$$

#### 3.2 Kinematic corrections - Functional approach

To describe the kinematics of the actual wheel center, we introduce the following key points:  $A_1$ , a reference point ( $A_1$  is coincident with the position of the wheel center at the height of reference of the axle system.), representing the wheel center as if it was fixed to the truck body; and  $A_2$ , its kinematically corrected position, reflecting the true wheel center. The kinematic correction is associated with the relative displacement between  $A_1$  and  $A_2$ , with

respect to  $\mathcal{R}_1$

$$\overrightarrow{A_1 A_2} = \begin{pmatrix} x(z, z_{opp}, \alpha) \\ y(z, z_{opp}, \alpha) \\ z \end{pmatrix}_1. \quad (1)$$

Using the velocity composition law, the absolute velocity of  $A_2$  expressed in  $\mathcal{R}_1$  writes:

$$\begin{aligned} \overrightarrow{V}(A_2/0) &= \overrightarrow{V}(A_2, 1/0) + \overrightarrow{V}(A_2/1) \\ &= \overrightarrow{V}(G/0) + \overrightarrow{A_2 G} \wedge \overrightarrow{\Omega}(1/0) + \begin{pmatrix} \dot{x} \\ \dot{y} \\ \dot{z} \end{pmatrix}, \end{aligned} \quad (2)$$

where  $\overrightarrow{V}(A_2, 1/0)$  and  $\overrightarrow{V}(A_2/1)$  are respectively the driving and relative velocity.

The time derivative of the position vector (1) yields:

$$\begin{pmatrix} \dot{x} \\ \dot{y} \\ \dot{z} \end{pmatrix}_1 = \underbrace{\begin{pmatrix} \frac{\partial x}{\partial z} & \frac{\partial x}{\partial z_{opp}} & \frac{\partial x}{\partial \alpha} \\ \frac{\partial y}{\partial z} & \frac{\partial y}{\partial z_{opp}} & \frac{\partial y}{\partial \alpha} \\ 1 & 0 & 0 \end{pmatrix}}_{J_t} \begin{pmatrix} \dot{z} \\ \dot{z}_{opp} \\ \dot{\alpha} \end{pmatrix}, \quad (3)$$

where  $J_t$  refers to the Jacobian matrix of  $(x, y, z)^T$  as a function of  $(z, z_{opp}, \alpha)^T$ . Index  $t$  refers to the translation dynamics. Then, plugging (3) into (2) gives:

$$\overrightarrow{V}(A_2/0) = \overrightarrow{V}(G/0) + \overrightarrow{A_2 G} \wedge \overrightarrow{\Omega}(1/0) + J_t \begin{pmatrix} \dot{z} \\ \dot{z}_{opp} \\ \dot{\alpha} \end{pmatrix}. \quad (4)$$

We are now looking at the kinematic corrections related to the rim plane orientation. Using Euler angles decomposition (ZXY type) for the rotation

of the spindle relative to the body, the spindle instantaneous rotational velocity with respect to  $\mathcal{R}_1$  frame writes:

$$\overrightarrow{\Omega}(2/1) = [A_{Euler2}^1] \begin{pmatrix} \dot{\delta}_2(z, z_{opp}, \alpha) \\ \dot{\epsilon}_2(z, z_{opp}, \alpha) \\ \dot{\eta}_2(z, z_{opp}, \alpha) \end{pmatrix}, \quad (5)$$

where  $[A_{Euler2}^1]$  is the transformation matrix from *Euler's* frame (Euler frame for the truck body/spindle link) to  $\mathcal{R}_1$  frame.

The absolute instantaneous rotation velocity of the spindle expressed in  $\mathcal{R}_1$  is written:

$$\overrightarrow{\Omega}(2/0) = [A_{Euler2}^1] \begin{pmatrix} \dot{\delta}_2 \\ \dot{\epsilon}_2 \\ \dot{\eta}_2 \end{pmatrix} + \overrightarrow{\Omega}(1/0). \quad (6)$$

In the same way as for the translation part:

$$\begin{pmatrix} \dot{\delta}_2 \\ \dot{\epsilon}_2 \\ \dot{\eta}_2 \end{pmatrix} = \underbrace{\begin{pmatrix} \frac{\partial \delta_2}{\partial z} & \frac{\partial \delta_2}{\partial z_{opp}} & \frac{\partial \delta_2}{\partial \alpha} \\ \frac{\partial \epsilon_2}{\partial z} & \frac{\partial \epsilon_2}{\partial z_{opp}} & \frac{\partial \epsilon_2}{\partial \alpha} \\ \frac{\partial \eta_2}{\partial z} & \frac{\partial \eta_2}{\partial z_{opp}} & \frac{\partial \eta_2}{\partial \alpha} \end{pmatrix}}_{J_r} \begin{pmatrix} \dot{z} \\ \dot{z}_{opp} \\ \dot{\alpha} \end{pmatrix}, \quad (7)$$

where  $J_r$  is the Jacobian matrix of  $(\delta_2, \epsilon_2, \eta_2)^T$  as a function of  $(z, z_{opp}, \alpha)^T$ . Index  $r$  refers to the rotation dynamics. Then, plugging (7) into (6), yields:

$$\overrightarrow{\Omega}(2/0) = [A_{Euler2}^1] J_r \begin{pmatrix} \dot{z} \\ \dot{z}_{opp} \\ \dot{\alpha} \end{pmatrix} + \overrightarrow{\Omega}(1/0). \quad (8)$$

The Jacobian matrices,  $J_r$  and  $J_t$ , embed the kinematic tables, forming the functional representation for defining axle system geometry. These ma-

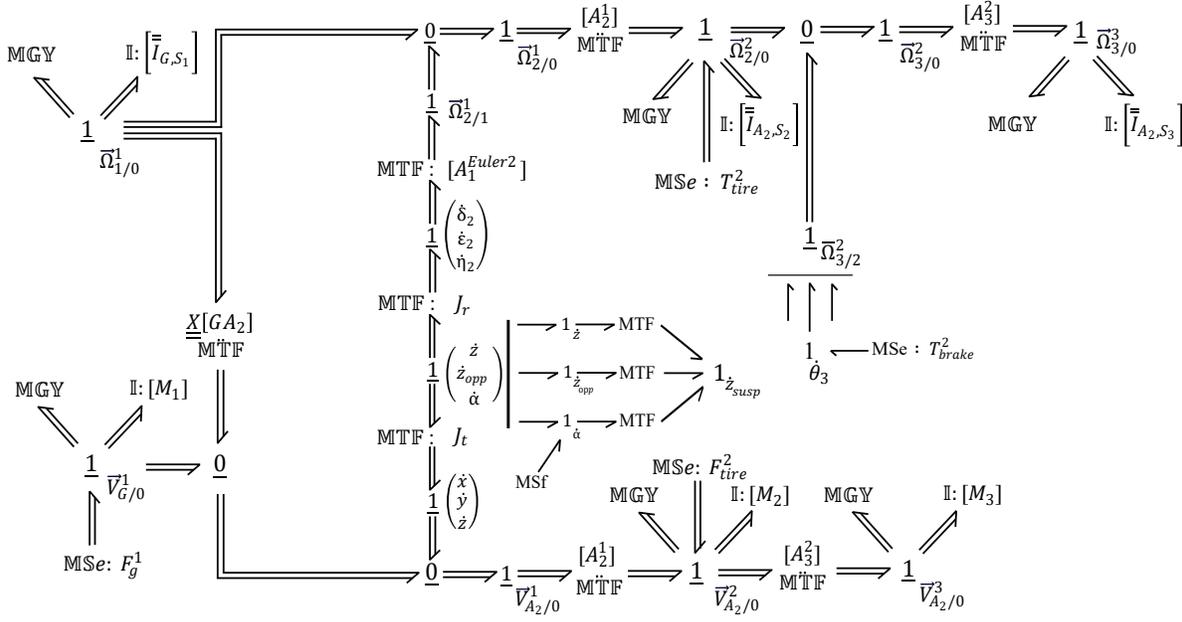


Figure 2: Front left axle bond graph model

trices, derived from measurements or multibody software simulation, detail the variations in track width, wheelbase, steering angle, camber angle, and self-rotating angle in relation to  $(z, z_{opp}, \alpha)^T$ .

### 3.3 Multibond graph model of a half-axle

It is worth noticing that the kinematic relationships are power conserving. For instance, the equation (4) is a flow balance equation and thus has 0-junction array representation. Repeating the process, the junction structure of the whole system is built (figure 2).

Two 1-junction arrays on the left-hand side correspond to, respectively, the truck angular velocity vector ( $\vec{\Omega}_{G/0}^1$ , at the top), and the truck centre of mass velocity ( $\vec{V}_{G/0}^1$ , at the bottom). Superscript 1 specifies the frame of expression. The rotational and translational dynamics of the body are represented by pairs of  $I$  and  $MGY$  multiport elements, the latter forming Euler junction structures. They are associated with Newton's laws expressed in the body frame. Gravity  $F_g^1$  is modelled by a modulated effort source on the translational 1-junction array.

The connection between the body and the spindle is obtained graphically using equations (4) and

(8). It is embedded in the two 0-junction arrays on the middle vertical bonds. This is the functional part of the bond graph representation.  $MTF$  multiport element characterize either frame transformation  $[A]$ , jacobian matrices  $J$  or antisymmetric matrix  $\underline{X}[GA_2]$ .

The rotational and translational dynamics of the spindle (resp. wheel) are taken into account through pairs of  $I$  and  $MGY$  multiport elements connected to the 1-junction arrays of the spindle (resp. wheel). Spindle (resp. wheel) angular velocity vector  $\vec{\Omega}_{2/0}^2$  (resp.  $\vec{\Omega}_{3/0}^3$ ), is at the top, and  $A_2$  velocity vectors,  $\vec{V}_{A_2/0}^2$  and  $\vec{V}_{A_2/0}^3$ , are at the bottom.

Road interaction efforts are considered at the wheel centre  $A_2$ , expressed in  $\mathcal{R}_2$ . Relative rotation between the wheel and the spindle, denoted  $\theta_3$ , represents the wheel own rotation. The braking torque  $T_{Brake}^2$  is applied to this DOF. Signal bonds between the different modulated elements are omitted in the bond graph representation. Regarding their complex geometry, modelling suspension systems in a functional framework is challenging. It is relevant to treat all the elements (springs, dampers, end stoppers, anti-roll bar, etc) globally as a ratio between the relative body/spindle vertical motion along  $\vec{Z}_1$  and that along the axis of each suspension element. The functional equations for suspension

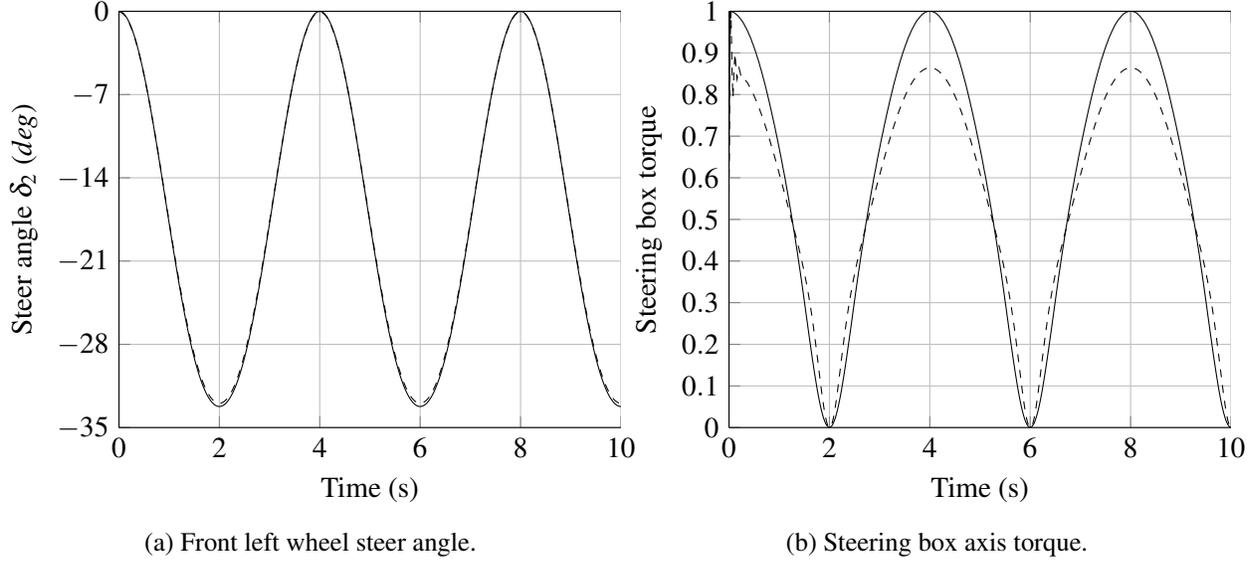


Figure 3: Dashed curve (MBT model) and solid curve (MS1 model).

velocity and suspension force, in dual form, are as follows:

$$\dot{z}_{susp} = \begin{pmatrix} \frac{\partial z_{susp}}{\partial z} & \frac{\partial z_{susp}}{\partial z_{opp}} & \frac{\partial z_{susp}}{\partial \alpha} \end{pmatrix} \begin{pmatrix} \dot{z} \\ \dot{z}_{opp} \\ \dot{\alpha} \end{pmatrix} \quad (9)$$

$$\begin{pmatrix} F_z \\ F_{z_{opp}} \\ F_\alpha \end{pmatrix} = \begin{pmatrix} \frac{\partial z_{susp}}{\partial z} \\ \frac{\partial z_{susp}}{\partial z_{opp}} \\ \frac{\partial z_{susp}}{\partial \alpha} \end{pmatrix} F_{susp} \quad (10)$$

and thus the bond graph representation follows.

#### 4 SIMULATION RESULTS AND VALIDATION

Simulation results are now presented using the MS1 software [9]. The following results are used to validate the bond graph model by comparison with simulations of an existing Volvo Group reference model obtained with a MultiBody-oriented Tool (MBT). This reference model considers the steered front axle. It contains all unsprung front masses, including front wheels, suspensions, and steering box. The various settings and parameters

of the MBT are based on real commercial truck data. The parameters and values are therefore not presented in this paper.

The simulation performed corresponds to a parking manoeuvre where the vehicle is steady. This real-life situation is used to evaluate the vehicle steering system and, more specifically, the effect of the axle kinematics during the steering, as well as the effort feedback on the steering system. In our case, the effort feedback is evaluated in terms of the torque about the output axis of the steering box.

A flow source is used to impose the output angular velocity of the steering box ( $\dot{\alpha} = 0.5 \cos(\frac{\pi}{2}t - \frac{\pi}{2})$ ). Steering angle  $\delta_2$  of the left front wheel versus time is plotted (Figure 3a).

Figure 3b then shows the torque measured at the steering box output axis, corresponding to the effort feedback. It should be noted that the curve values have been normalised.

The results in these figures are relatively good in terms of the shapes and the amplitudes of the curves. The small differences between the MS1 and MBT results are due to the interpolation of the kinematic tables and the evaluation of the inertia of the spindles and the wheels. The difference at the start of the simulation (Figure 3b) is due to the crushing of the tyre, which acts as a high stiffness

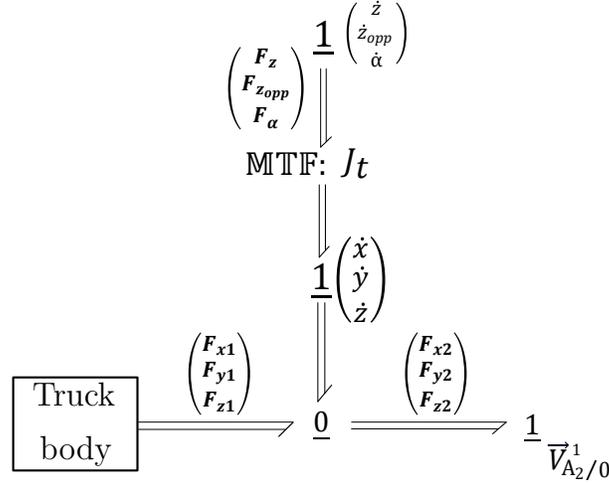


Figure 4: Partial bond graph of the translational body/spindle specific joint

vertical spring, not taken into account in the MS1 model.

## 5 EFFORT INTERPRETATION OF THE MULTIBOND GRAPH REPRESENTATION

The purpose of this section is to give an interpretation the model from an effort perspective, establishing connections between the modelled elements and the context of vehicle dynamics.

The efforts applied to the “spindle+wheel” , either by pneumatic/ground contact forces, braking torque or gyroscopic effects, have an impact on the body dynamics, passing through the axle parts. Analysing the joint forces on the spindle and their influence on body and axle movements reveals numerous phenomena [10]. To illustrate, we focus on the translational aspect of the body/spindle connection using a partial bond graph representation (Figure 4).

We introduce the following force vectors expressed in frame  $\mathcal{R}_1$ :

- $(F_{x1} F_{y1} F_{z1})^T = (F_{x2} F_{y2} F_{z2})^T$ : Truck body/spindle joint force;
- $(F_z F_{z_{opp}} F_\alpha)^T$ : Truck body/spindle joint force in the axle DOFs space  $(z, z_{opp}, \alpha)$ .

Equation (11) summarizes the relationships between these force vectors:

$$\begin{pmatrix} F_{x1} \\ F_{y1} \\ F_{z1} \\ F_z \\ F_{z_{opp}} \\ F_\alpha \end{pmatrix} = \begin{pmatrix} \boxed{1} & 0 & 0 \\ 0 & \boxed{1} & 0 \\ 0 & 0 & \boxed{1} \\ \frac{\partial x}{\partial z} & \frac{\partial y}{\partial z} & 1 \\ \frac{\partial x}{\partial z_{opp}} & \frac{\partial y}{\partial z_{opp}} & 0 \\ \frac{\partial x}{\partial \alpha} & \frac{\partial y}{\partial \alpha} & 0 \end{pmatrix} \begin{pmatrix} F_{x2} \\ F_{y2} \\ F_{z2} \end{pmatrix}. \quad (11)$$

These considerations illustrate a number of key points in vehicle dynamics and highlights the role of the kinematic table from an effort point of view:

- Within the dashed border in equation (11) matrix, the terms indicate that joint force directly influences the truck body for acceleration, braking ( $F_{x1} = F_{x2}$ ), and steering ( $F_{y1} = F_{y2}$ );
- The terms in the dotted border highlight the contribution of orthogonal  $(x, y)$ -forces to the vertical  $z$ -forces in the axle. This represents the Brouhiet effects (The Brouhiet effect relates to forces, due solely to the kinematics of the axles, inducing effort increases on the actual relative move-

ment of the spindle due to force components in other directions). The term  $\frac{\partial x}{\partial z}$  corresponds to an anti-dive effect during braking and anti-camber effect under acceleration, stemming from account the longitudinal force and the kinematic table;

- The double-bordered terms depict forces transferred to the steering system showing the connection between the axle and the steering box. These effects enable the driver to sense, at the steering wheel, the forces generated at ground level. The elements  $MTF : \frac{\partial x}{\partial \alpha}$  and  $\frac{\partial y}{\partial \alpha}$  represents the kinematic effects of the axle, contributing significantly to vehicle stability and enhancing driver perception.

## 6 CONCLUSION AND PERSPECTIVES

In this paper, a 6-wheel truck chassis multibond graph model has been presented. An integrated functional/multibody modelling approach is used to model the body/spindle joint. The presented model enables the consideration of the longitudinal, transversal, and vertical effects of the chassis. Also, some effects traditionally considered in vehicle dynamics (such as Broulhiet effects) can be, by means of the functional approach, embedded in the bond graph representation. As perspectives, it could be interesting to consider the elasticity of the axles in the model as well as a more accurate evaluation of the spindle inertia. These improvements are prospects for future work.

Although not presented in the paper, this approach is part of an overall methodology dedicated to assist engineers in designing mechatronic systems from dynamic and energy criteria [11] [12]. In the truck industry context and their mechatronic systems, this methodology could address issues such as steering design, considering the vehicle life situations.

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