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Mechatronic Bond Graph Modelling of an Automotive Vehicle

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SUMMARY

Automotive manufacturers have created vehicle models but these are somewhat complex and use black boxes with numerous tables and mappings. Each model is generally valid for its own field of use and can thus produce significant differences in results outside its specific field. Moreover vehicle behaviour requires detailing and analysis during precise manoeuvres called "life situations". These "life situations" require different models to be set up.

A complete vehicle model is necessary when attempting to reproduce simulated realistic behaviours for different manoeuvres. High modelling standards are also sought when vehicle systems such as steering, suspension or braking are approached.

This paper presents bond graph modelling of an automotive vehicle with sixteen degrees of freedom. Although not presented here, one purpose of such a model is dimensioning of different vehicle systems, in certain life situations, using an inverse approach. The vehicle body is modelled by a rigid body with six degrees of freedom. Each wheel has two degrees of freedom, one corresponding to vertical suspension deflection, the other to wheel axial rotation. Each front wheel has one additional degree of freedom corresponding to steering mobility. Vehicle components modelled and connected to the body model are thus four suspensions, wheels and steering system.

Simulations of this model have been undertaken on 20Sim. They reveal good correlation with a model simulated on AMESim by automotive manufacturer PSA Peugeot Citroën, considered as the validation reference for this study.

Keywords: Vehicle dynamics, bond graph modelling, EPS, suspension, tyre, systemic and phenomenological analyses.

1. INTRODUCTION

Intermediary simpler models often need to be studied prior to automotive vehicle modelling and to understand properly vehicle dynamics. These intermediary models include the so-called quarter vehicle model, the longitudinal two-wheel model,

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the transverse two-wheel model and the bicycle model. Such models enable a number of key notions in the field of vehicle dynamics to be gradually considered. Model complexity level depends basically on the model objective. For example, the bicycle model corresponds to a longitudinal two-wheel model, in which longitudinal, lateral and yaw motions are all considered. Steering is included in this model, which allows both longitudinal and lateral vehicle dynamics to be analysed. In relation to the bond graph, several authors have proposed representations of this type of model, in particular Karnopp [1], who derived a bond graph representation of a bicycle model starting from a generic vehicle body representation. Margolis [2] introduced advanced physical phenomena such as vehicle-trailer coupling, transverse wheel compliance and vehicle frame bending, prior to conducting stability analysis based on a linear bicycle model. Karnopp and Wuh [3] used a simplified bicycle model of a vehicle + driver performing a lane change driving manoeuvre. Both the vehicle power steering system and tyres were taken into account. The aim was to study how a power steering system affects driver feel for the reaction moment caused by the force exerted on the wheels.

Another category of two-wheel models includes consideration of vehicle heave and pitch motions. For example, Louca *et al.* [4] developed a bond graph representation of such a model, incorporating engine and drive train models. The purpose of their study was model reduction to achieve enhanced design-related insight into the model, without decreasing model predictability.

A number of authors have extended two-wheel bond graph models to include additional motions. For example, Assadian *et al.* [5] included the steering function in a longitudinal two-wheel model (involving pitch and heave motions) with the aim of designing braking, steering, and suspension system controllers.

The model presented in this paper belongs to the four-wheel vehicle model category but, again, various versions may be discussed. In general, they all consider a vehicle body model with six degrees of freedom model, but they may differ in relation to the assumptions made for axle kinematics, suspension systems and, more generally, for different model-integrated systems. Mera *et al.* [6] and Vera *et al.* [7] presented a bond graph model for a 2WD (Wheel-Drive) and a 4WD vehicle respectively. These four-wheel models have eighteen degrees of freedom for the vehicle body and unsprung masses, in which longitudinal, vertical and pitch motions were considered. A bond graph model of the drive train was associated for acceleration manoeuvre analysis as well as 2WD and 4WD vehicle comparison. More recently, Mera *et al.* [8] presented a series of three four-wheel vehicle models aimed at conducting roll axis influence analysis. More specifically, their third model featured a Mc Pherson suspension for each wheel. Pacejka and Tol [9] presented a series of models that integrate gradually certain degrees of freedom. Using a truck example, these authors applied initially a bicycle model, then introduced heave and pitch motions and finally considered all three-dimensional motions. Margolis and Asgari [10] also presented a series of models of varying complexity depending on their purpose. These models were a quarter

model, a heavy pitch model, a bicycle model, a simplified four-wheel model and a full four-wheel model. The four-wheel model took into account the vehicle body, suspension systems involving horizontal bushings and vertical deflection, and tyres based on Pacejka longitudinal and the cornering force formulae. Finally, Margolis and Shim [11] developed a four-wheel vehicle model featuring three-dimensional rotation and translation motions. This model included actuated suspension models involving only vertical deflections, tyre models based on Pacejka formulae, an electric brake system model and an electric power steering model. The vehicle body model was simplified following vehicle kinematics approximation.

This paper proposes an automotive vehicle model featuring a number of integrated systems such as suspensions, tyres and power steering. The simulation objective is to reproduce, for example, braking in curved motion. The model can be used for dimensioning vehicle mechatronic systems considered. Use of the model in a dimensioning context is not included in this paper, whose purpose is to demonstrate the modelling approach broken down into first systemic and subsequently phenomenological analysis. These analyses correspond to technological component and energy structure description levels respectively. The paper also set out to demonstrate that bond graph representation can be advantageously used in the vehicle dynamics field. This aspect is further emphasised when considering the vehicle-integrated mechatronic systems.

The model presented in this paper is a four-wheel vehicle model including suspension systems, tyres and electric power system. Unlike a similar model considering general three-dimensional rotational and translation motions (e.g. [11]), the vehicle body part of the bond graph representation is not simplified and a highly non-linear general model is therefore created. The model complements other comparable four-wheel vehicle models in terms of complexity (e.g. [6], [7] and [8]), yet differs in that it includes different mechatronic system models or is used for different purposes.

The paper is structured in 5 sections. Section 2 introduces the technological component analysis level. Section 3 deals in depth with model break-down and introduces the energy structure analysis level. A word bond graph representation and a bond graph representation correspond to both analysis levels. Section 4 introduces two simulations obtained using 20Sim software [12] and compared for validation purposes with another model developed using AMESim software [13] at PSA Peugeot Citroën. The latter model was considered as the reference model for the validation purposes. Section 5 concludes the paper.

2. SYSTEMIC ANALYSIS AND WORD BOND GRAPH REPRESENTATION

Systemic analysis breaks down the model into ten main components, specifically vehicle body, suspension systems, wheels and steering system. Some of these components are simply duplications in structural terms, but using different parameters. This is the

Fig. 1. Four-wheel vehicle model

case for both front and rear axle suspension systems and wheels. We now present these components.

2.1. Vehicle body

The vehicle body is assumed to be rigid and it has six degrees of freedom, specifically three translations and three rotations with respect to an inertial reference frame denoted R_0 . Figure 1 shows the defined vehicle body reference frame R_v with the centre of mass G located as its origin. \vec{x} , \vec{y} , \vec{z} are roll, pitch and yaw axes respectively, with the corresponding Cardan angles ϕ , θ and ψ denoting the successive rotation angles with respect to the inertial reference frame [14]. The vehicle body is connected to suspension systems at A_{fr} , A_{fl} , A_{rr} , A_{rl} , front-right (fr), front-left (fl), rear-right (rr) and rear-left (rl) attachment points respectively.

2.2. Suspension systems

Main suspension system functions are to maintain the wheels in contact with the ground, transmit tyre forces, and filter road excitations. In conjunction with axles, suspension systems form the link between wheels and vehicle body. There are different axle-suspension system types (Mc Pherson, pseudo Mc Pherson, trailing-arms, multi-arms,) and their kinematics are somewhat complex to model in a multibody system context. Simple suspension systems have been considered here to maintain clarity. Suspension deflection is assumed to be along a vertical axis. These suspension systems are coupled dynamically with the vehicle body, through attachment points A_{fr} , A_{fl} , A_{rr} , A_{rl} , and with the wheels. However, "Brouilhet effects" [15] govern part of the dynamic forces due to rear axle kinematics between the wheels and the vehicle body. "Brouilhet effects" are also present in the front axle but they are neglected.

2.3. Wheels

The tyre is an important wheel component: it not only forms the road/vehicle interface, but also fulfils both directional and longitudinal driving functions. Tyre forces therefore play an important part in the overall vehicle dynamics. Wheel kinematics is somewhat complex and simple assumptions are again made. Suspension systems couple tyre vertical dynamics with vehicle body dynamics. Both longitudinal and cornering efforts and self-aligning torque are modelled, in general, by Pacejka's formulae. Points at which road/tyre forces are transmitted to the vehicle body are denoted C_{fr} , C_{fl} , C_{rr} , C_{rl} : these correspond to wheel centres. Elastic kinematical phenomena are

Fig. 2. Word bond graph representation of vehicle

also modelled [16]: these correspond to wheel-axle structure and joint global elasticity. An Anti-Roll Bar (ARB) located between the wheels of each axle is also considered. Finally, a brake torque model is incorporated in the wheels model to take into account front-rear braking distribution.

2.4. Steering system

The steering system considered is an Electrical Power System (EPS). This comprises two actuating lines, one controlled by the driver and the other by electrical power assistance. The electrical actuator is a DC motor coupled to the steering column pinion. It was important to include this component in the model because the projected dimensioning study focuses on its design with respect to a number of assistance level specifications. The steering system is coupled to the wheels through a kinematical chain featuring rack and connecting rods.

2.5. Word bond graph representation

Figure 2 shows word bond graph representation of the automotive vehicle model considered in this paper. The figure shows the word Vehicle Body with four multibonds coupling the wheels through attachment points C_{fr} , C_{fl} , C_{rr} , C_{rl} . Wheel/vehicle body coupling corresponds to road/tyre interactions transmitted to the vehicle body. These interactions are not included in suspension words because only vertical dynamics is considered for these components. Single port coupling between suspension words and wheel words represents corresponding vertical dynamics coupling. Rear and front Anti-Roll Bars couple with rear and front suspensions respectively through a single power port. The Electrical Power Steering system word is coupled with both front wheel words through a single power bond and this corresponds to steering interaction. For the sake of clarity, signal bonds between the different words have not been represented in this word bond graph representation.

3. PHENOMENOLOGICAL ANALYSIS AND BOND GRAPH REPRESENTATION

Modelling assumptions for different phenomena are now given for the components described above. Energy interpretation is converted into the bond graph formalism. Phenomenological analysis allows introduction of a bond graph representation for each component word.

3.1. Vehicle body

As stated above, the vehicle body is considered to possess six degrees of freedom due to its general motion in 3-dimensional space. Both kinetic and gravity phenomena are considered. Multiple references may be found for setting up the corresponding multibond graph vehicle body representation. One method is based on an approach involving the Newton-Euler equation. Figure 3 shows the multibond graph vehicle body representation. The reader is referred to for example [17], [18] and [19] for a more detailed explanation of this representation.

Brief explanations are nevertheless given here in relation to physical interpretation of this representation. In this case, multibond graph construction is essentially based on the vehicle body kinematical relationships. This creates the representation junction structure. Two 1-junction arrays on the representation middle line correspond to vehicle angular velocity vector $\underline{\Omega}_V^0$ (rotational dynamics at the top) and vehicle centre of mass velocity vector \underline{V}^0 (G) (translation dynamics at the bottom) respectively. These velocities allow any vehicle body point velocity to be derived. In particular, each suspension and wheel attachment point velocity is obtained using a TF multiport element and 0-junction array. This corresponds to the following kinematical relationship for a rigid body: $\underline{V}^0(\mathbf{M}) = \underline{V}^0(\mathbf{G}) + \underline{\mathbf{X}}(\underline{\mathbf{GM}}) \cdot \underline{\Omega}_V^0$.

The second term in this relation is a cross product matrix expression, in which $\underline{\mathbf{X}}(\underline{\mathbf{GM}})$ is an anti-symmetric matrix made up of position vector $\underline{\mathbf{GM}}$ components with respect to the vehicle body reference frame (M representing any attachment point).

The part of the vehicle body multibond graph located between MTF columns features flow and force vectors projected onto the vehicle body reference frame. Suspension flow and effort vectors are expressed in the inertial reference frame \mathbf{R}_0 , so this requires MTFs characterised by transformations between inertial and vehicle body reference frames.

Vehicle body / wheel coupling is broken down into a multibond and single bond. On one hand, the dimension 2 multibond corresponds to both longitudinal and cornering forces transmitted to the vehicle body and, on the other hand, the single bond corresponds to self-aligning torque, which is also transmitted to the vehicle body. The associated self-aligning torque flow is the vertical component of the vehicle body angular velocity vector. It is coupled to a 1-junction, at which single bonds starting from wheel words are connected.

While TF multiport elements are characterised by a constant matrix, MTFs characterised by frame transformation are modulated by vehicle body orientation Cardan angles with respect of the inertial reference frame. These angles are obtained following algebraic transformation and integration of the vehicle body angular velocity vector components. The algebraic transformation is itself modulated by the same angles. Specific frame transformation between vehicle body and front wheels (resp. A_{FRw}^v and A_{FLw}^v) requires two additional angles but no vehicle body yaw angle. This allows

Fig. 3. Multibond graph representation of the vehicle body

Fig. 4. Bond graph representation of a) the suspension word b) the Anti-Roll Bar word

both longitudinal and transversal wheel directions to be indicated. Additional angles are steering angles from the EPS word and induced steering angles from front left and front right wheel words. The latter angles result from elastic kinematical phenomena and will be included with the front wheel components. For the sake of multibond graph representation clarity, signal bonds have not all been shown.

Rotational and translation dynamics are represented by a pair of I and MGY multiport elements forming Euler Junction Structures [1], [20]. They are associated with Newton's laws expressed in relation to the vehicle body reference frame. Concerning the translation dynamics, gravity is modelled by a modulated effort source corresponding to vehicle body gravity components within the vehicle body reference frame.

3.2. Suspension systems

Passive suspensions have been modelled. Figure 4 shows the bond graph suspension representation (Figure 4a). Dynamics is assumed solely on the vertical axis and suspension actions are denoted F_z . Spring/damper suspension component phenomena correspond to C-type energy storage and energy dissipation; a pair of C and R elements thus represents these phenomena. Their underlying laws are assumed to be non-linear and are given by the automotive manufacturer. Phenomena associated with Anti-Roll Bars are simply represented by a C element modelling C-type energy storage (Figure 4b). Anti-Roll Bars are assumed to operate based on left/right suspension deflection differences. The stiffness characterising these components is different for rear and front axles and represents bar torsion stiffness. "Brouilhet effects", modelled for the rear axle result from braking torques and correspond to forces exerted on the vehicle body through the axle (e.g. trailing arm) [15]. However, these have been neglected for the front axle. In the bond graph representation, "Brouilhet effects" are simply represented by an effort source modulated by longitudinal and cornering road/tyre interactions and wheel radius. They contribute to suspension dynamics.

3.3. Wheels

Figure 5 shows the details of the bond graph representation of vehicle wheels. Wheel vertical dynamics (left-hand side of Figure 5) is uncoupled from longitudinal and cornering dynamics (right-hand side of Figure 5). Suspension force and wheel dynamic radius only are used to modulate longitudinal and cornering dynamics. Suspension force modulates both the effort source representing wheel braking torque and the MR

element used for modelling road/tyre interactions. Wheel dynamic radius, measured on the C-type representing tyre vertical stiffness corresponds to variations in wheel radius. This modulates the MTF representing coupling between wheel axial rotation and longitudinal dynamics.

Kinetic phenomenon and weight, represented by an I element and an effort source respectively, are considered as unsprung mass for vertical dynamics purposes. Linear vertical stiffness and damping phenomena, represented by a pair of C and R elements, are also considered for the tyre. The road profile is introduced through a flow source that can be modulated by the longitudinal vehicle velocity. This allows driving conditions for the road profile to be considered.

Three phenomena are considered for wheel longitudinal and cornering dynamics purposes, specifically road/tyre interactions, elastic kinematical phenomena and braking action. Road/tyre interactions are based on Pacejka formulae [21] and [22]. These formulae express longitudinal F_x , cornering forces F_y and self-aligning torque M_z in terms of wheel vertical load, tyre sliding angle, longitudinal slip rate and camber angle. Longitudinal and cornering forces are assumed to be uncoupled in the represented model. Moreover, cornering force and self-aligning torque are considered uniquely dependent on vertical load and sliding angle, whilst longitudinal force is considered uniquely dependent on vertical load and longitudinal slip rate. These forms of Pacejka formulae constitute characteristic relations for the corresponding MR element. Signal bonds modulating this element make all dependencies available for formula calculation.

Elastic kinematical phenomena are also considered and result from wheel-axle structure and joint global elasticity [16]. These phenomena are considered from an overall standpoint, so they are represented by an "ad hoc" component, characterised by corresponding mathematical relations. One consequence of this is an additional wheel steering angle, which induces wheel kinematics modifications. The induced steering angle also alters frame transformation in the vehicle body word. Induced steering is only considered for the front axle.

Finally, the braking system is modelled very simply because it comprises an effort source modulated by longitudinal vehicle acceleration, allowing braking strategy to be computed. Braking torque is applied to the wheel for which the rotational kinetic phenomenon has been represented by an I element.

Figure 5 shows clearly the wheel word link with suspension, vehicle body and EPS links. The suspension word link represents vertical dynamics coupling. Links with the vehicle body word correspond to transmission of longitudinal, cornering forces and self-aligning torque to the vehicle body. Finally, EPS coupling corresponds to transmission of a torque M_s made up of both self-aligning torque and cornering force through a mechanical trail in the axle. This coupling is only included for the front axle.

Fig. 5. Bond graph representation of the wheel word

3.4. Power steering system

The key point in relation to power steering system modelling was its correct coupling with the full vehicle representation and thus its proper feedback along with the forces concerned. Figure 6 shows the bond graph representation for the power steering system. In this system, two actuating lines both act in parallel on the wheels, implying that their action is superposed. The principal actuating line from the steering function standpoint is the line controlled by the driver. The secondary actuating line is controlled by and assists the principal line in torque terms, when demanding manoeuvres are performed.

The principal actuating line model (top left in Figure 6) includes the driver (steering) wheel, column and torsion rod down to the column pinion. The driver is assumed to generate steering wheel motion. This is represented by a modulated flow source. The 'driver' block consists of the time function corresponding to the life situation required for the simulation or dimensioning study. Wheel and column rotational kinetic phenomena are consolidated in an I element. Viscous friction in the column is taken into account through inclusion of a linear R element. The torsion rod is fitted with a sensor, whose corresponding torque measurement is used as DC motor control input in the assistance actuating line. Energy storage associated with the torsion rod is represented by a C element. In relation to the principal actuating line model, column pinion rotational kinetic phenomena are represented by an I element, whilst dry friction on this column pinion is represented by a non linear R element.

The assistance actuating line model (bottom left in Figure 6) includes a DC motor with a supply voltage derived from the assistance law. Supply voltage is represented by a modulated effort source, whilst automotive manufacturer tables provide the assistance law. A pair of I and R elements represents the motor electrical part, specifically induction coil storage energy phenomena and conductor dissipation phenomena due to Joule effect. A gyrator element represents electromechanical coupling. The following I and R elements represent kinetic phenomena associated with rotor inertia rotation and friction (both viscous and dry) respectively. A reduction gear couples the DC motor to the principal actuating line at column pinion level. Reduction gear efficiency is considered by including a modulated R element allowing energy to be divided into portions lost and effectively transmitted to the rest of the system.

The remainder of the EPS comprises mechanical transmission to the wheels. A transformer represents rack and pinion reduction. Reduction efficiency is again considered and represented by a modulated R element. A pair of I and R elements represents kinetic phenomena associated with the rack in translation and rack friction (viscous and dry). Modulated transformers represent transmission of rack motion to wheel

Fig. 6. Bond graph representation of EPS system

Fig. 7. a) Lateral acceleration of vehicle model during a steering operation, and b) Vehicle model vertical road/wheel forces for a steering wheel angular increment, followed by curved braking

motions. These modulated transformers are characterised by crank-rod kinematical-type relationships. Modulations depend on left and right hand-side steering angles respectively. These steering angles are also used for transformer modulations in the vehicle body word. Finally, I elements represent kinetic energy phenomena associated with both left and right pivots in rotation.

Simulation results are now presented. The following plots are both used to validate the bond graph model with respect to simulations undertaken using the AMESim tool.

4. SIMULATION RESULTS

The 20Sim-based bond graph model is compared, under simulated conditions, to an existing PSA Peugeot Citroën reference model under AMESim software. Different constitutive laws and parameters have been provided by automotive manufacturer PSA Peugeot Citroën and are thus not presented in this paper. Two simulations were performed, corresponding to a simple steering operation and a steering operation followed by a curved braking operation respectively.

For the simple steering operation, the vehicle initial velocity was 80 km/h and a 35 degree steering angle was applied after 1 s. Lateral acceleration (in g) versus time was observed. Figure 7a results are relatively good in relation to both curve shape and magnitude. In particular, the plot shows increasing lateral acceleration up to a stationary value corresponding to the vehicle in curve.

For the steering + braking operation, the vehicle initial velocity was again 80 km/h, a 20 degree steering angle was applied after 1 s and curved braking was applied after 5 s. Figure 7b shows vertical forces at the road/wheel interface. The Figure 7b plot again reveals relatively good correlation between 20Sim and AMESim simulation results. These indicate that, for the steering operation, both left-hand side vehicle wheels are loaded, while both right-hand side vehicle wheels remain unloaded. When braking occurs, both front wheels are loaded, while both rear wheels remain unloaded.

Two simulations only are presented here and used to validate the bond graph model. Many other model outputs might have been tracked to test the bond graph. However, the life situations implemented and the observed output are reckoned sufficiently representative to consider the bond graph model valid with respect to the AMESim model.

5. CONCLUSION

Vehicle model analysis was not the objective of this paper, rather its purpose was to demonstrate that bond graph representation can compete with other modelling tools in the field of vehicle dynamics. Owing to its multiple description levels, a bond graph dictates a hierarchical analysis structure. It starts from systemic analysis revealing the technological component description level, but it also demonstrates a phenomenological description level, obtained from phenomenological analysis. Moreover, a bond graph offers potentially a mathematical description level through the concept of causality, which allows different bond graph element constitutive relations to be structured into a simulation model or even a state space representation [23].

In spite of the relatively good results presented, model improvement may be sought. One of the main directions for model improvement would be ensured by detailing axle kinematics. This would enable *ad hoc* elements, such as "Brouilhet effects" or elastic kinematical phenomena, to be replaced by multibody models. These improvements are prospects for future work. It was stated in the introduction to this paper that using a bond graph model represents a dimensioning perspective with respect to dynamic and energy criteria. Dimensioning is based here on a methodology, developed at the Laboratoire d'Automatique Industrielle of the National Institute of Applied Sciences, Lyon and designed for setting up design assistance tools. It involves an inverse approach allowing direct application of specifications for selecting or validating actuation line components suited to dynamics and energy requirements ([24], [25], [26], [27]). A key characteristic of this methodology is also use of the bond graph tool. In conjunction with the bicausality concept [28], [29], bond graph representation of a mechatronic system enables both direct and inverse models to be cleverly derived. This methodology offers interesting features for solving design problems confronted by automotive manufacturers. It shows that vehicle mechatronic system design issues can be dealt with more precisely and with greater relevance [30], [31], [32]. An important contribution of this methodology in an automotive context is the possibility of overall dimensioning, because mechatronic systems are usually coupled in vehicle life situations [33].

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