



HAL
open science

Sizing of an Electric Power Steering system on dynamic and energetic criteria

Olivier Mechin, Wilfrid Marquis-Favre, Serge Scavarda, Pierre Ferbach

► **To cite this version:**

Olivier Mechin, Wilfrid Marquis-Favre, Serge Scavarda, Pierre Ferbach. Sizing of an Electric Power Steering system on dynamic and energetic criteria. ICBGM'03, International Conference on Bond Graph Modeling and Simulation, Jan 2003, Orlando, United States. pp.137-147. hal-02084664

HAL Id: hal-02084664

<https://hal.science/hal-02084664>

Submitted on 29 Mar 2019

HAL is a multi-disciplinary open access archive for the deposit and dissemination of scientific research documents, whether they are published or not. The documents may come from teaching and research institutions in France or abroad, or from public or private research centers.

L'archive ouverte pluridisciplinaire **HAL**, est destinée au dépôt et à la diffusion de documents scientifiques de niveau recherche, publiés ou non, émanant des établissements d'enseignement et de recherche français ou étrangers, des laboratoires publics ou privés.

Sizing of an Electric Power Steering system on dynamic and energetic criteria

Mechin O. * and **, Marquis-Favre W.*, Scavarda S.*, Ferbach P.**

* LAI-Institut National des Sciences Appliquées de Lyon

Bât. Antoine de St Exupéry

25, avenue Jean Capelle

F-69621 Villeurbanne Cedex, France

Wilfrid.Favre@lai.insa-lyon.fr

** PSA Peugeot Citroën

Direction des Organes

18, rue des Fauvelles

F-92256 La Garenne-Colombes Cedex, France

Olivier.mechin@mpsa.com

Keywords : Bond graph, modeling, inverse method, sizing, automotive design EPS system.

Abstract :

This paper presents the methodology to size a mechatronic system on dynamic and energetic criteria. The methodology is based on the establishment of the inverse model from the bond graph representation of the system by using the bicausality concept. By means of an automotive example, we illustrate the methodology and we present a solution to extend it to more complex problem.

Introduction

The automotive industries use and develop more and more electronic systems. Since many years, cars have been equipped in many systems such as ABS, ESP...They offer higher possibility due to their ability to improve performances compared to passive systems [1]. Up to now, steering systems were unaffected by this evolution to electronic for cost and geometric size reasons. We remark that technical improvements have allowed to create Electronic Power Steering (EPS) systems which will replace hydraulic ones in the future. However, these systems are more and more complex because they are at a time electric system and mechanic system i.e. mechatronic. The consequences of this evolution to electronic are that classical conception schemes are themselves transformed. In the design phase, as well as determining geometric shapes and dimensions, now appears the problem of determining the fitted actuators in terms of effort, velocity and power to obtain the attended objectives. Furthermore, the

automotive market restrictions impose to the carmaker to reduce delays and costs of design. In this context, to define and to establish correctly specifications and attended goals by the systems are one of the key point of the design phase.

The challenge that a system designer has to deal with is to define, to choose a structure for the system and to size the system of components according to specifications. To help the designer in his work, the Laboratoire d'Automatique Industrielle from INSA de Lyon has developed a methodology that gives information on model components in terms of admissible power. This methodology consists of an inverse approach where the objectives must be described in terms of dynamics imposed on a given load and where the system structure is clearly defined. Furthermore the original aspect of this methodology is the use of bond graph language [2-3] as the modeling and analysis tool but also the bicausality concept to establish the inverse model [4]. The sizing phase uses energetic and dynamic criteria which are more adapted to define the specifications of the system. These needs thus expressed have logically driven us to opt for the bond graph representation as modeling tool.

In the following sections, we will present the way specifications are defined and on an example of the EPS, we will apply the methodology to size the actuating system as the vehicle does a lane change test. We will talk briefly about the bond graph model of the vehicle (characteristics, hypotheses...) and we will present in more detail the bond graph model of the Electric Power Steering system. Then, we will establish the inverse model bond graph and we will apply the methodology to validate and to size the actuator of an EPS as the vehicle does a lane change.

Subsequently, the results of simulations are presented followed by conclusions.

I. Methodology

In this section, we deal with our approach to consider the problem of component sizing and the different steps of the methodology. At first, we recall the process of inversion in the bond graph context and then we explain how a structural analysis of the inverse model gives information about the trajectory that may be imposed. Once the system is established in an inverse form, and given the direct model output, the different stages of the actuating line can be validated in terms of power and power variables, and the direct model input can then be computed.

The sizing problem considered here consists in validating an actuating system from dynamic and energetic considerations with respect to the desired output trajectory to be realized.

I.1 Inverse model and bond graph

Without losing generality in the approach, let us consider an actuating line to size and its load represented by the Fig 1a causal bond graph. We represent the direct model can be expressed in state space representation by the equations 1 and can be derived from its causal bond graph (Fig 1a). In this case the computation of the output y and the power variables e_A and f_A on the load power bond necessitate knowing not only the input control but also the actuating line structure and parameters (that are *a priori* not known in the sizing problem).

$$\begin{aligned} \dot{X}_{\text{dir}} &= f_{\text{dir}}(X_{\text{dir}}, u) \\ y &= g_{\text{dir}}(X_{\text{dir}}, u) \\ e_A &= h_{\text{dir}}(X_{\text{dir}}, u) \\ f_A &= i_{\text{dir}}(X_{\text{dir}}, u) \end{aligned} \quad [1]$$

where X_{dir} is the state vector of the direct model, $y(t)$ is the output, $e_A(t)$ and $f_A(t)$ are the power variables required from the actuator and u the input control.

The corresponding input/output inverse model can be described by equations 2 and derived from the Fig 1b inverse bicausal bond graph (as it will be shown later in this paper). In this case the computation of the power variables e_A and f_A only depends on the desired output y and some of those derivatives (that can now be imposed from the specifications), and the load structure and parameters (that are to given in the actuating line sizing problem).

$$\begin{aligned} \dot{X}_{\text{inv}} &= f_{\text{inv}}(X_{\text{inv}}, y, y^{(1)}, \dots, y^{(\alpha)}) \\ u &= g_{\text{inv}}(X_{\text{inv}}, y, y^{(1)}, \dots, y^{(\beta)}) \\ e_A &= h_{\text{inv}}(X_{L_{\text{inv}}}, y, y^{(1)}, \dots, y^{(\alpha)}) \\ f_A &= i_{\text{inv}}(X_{L_{\text{inv}}}, y, y^{(1)}, \dots, y^{(\alpha)}) \end{aligned} \quad [2]$$

where X_{inv} is the state vector of the minimal-order inverse model, $X_{L_{\text{inv}}}$ is the state vector of the load inverse model, and $y^{(1)}$ or $y^{(\beta)}$ are the 1st and β^{th} time derivative of the output y .

In this context of an inverse approach, the bond graph language is very convenient as it represents the structure and the energy transfers in a dynamic system. It also provides a unified approach for modeling systems in different energy domains. Furthermore, the bond graph representation offers many possibilities in graphical analysis to help the designer like for instance the causality and bicausality concepts that respectively enable a direct model and an inverse model to be derived.

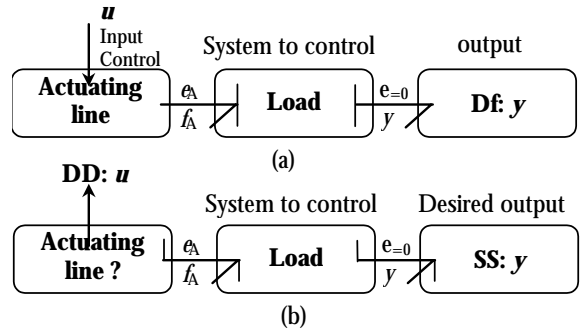


Fig. 1: Generic (a) direct and (b) inverse model of an actuating system

I.2 Bond graph and bicausality

An acausal bond graph model represents the physical structure and the constraints between power variables in the system, independently of the equation formulation. The causal strokes enable a graphic organization of the mathematical equations in the model. The main idea of the bicausality is to consider the causal stroke as two half causal strokes corresponding to the effort and flow variables at a power port. The “standard” causality (Fig. 1a and b) shows how power variables are coupled in their assignment. On the contrary the bicausality concept (Fig. 1c and d) is a way for independently propagating the effort and flow variables in the bond graph model.

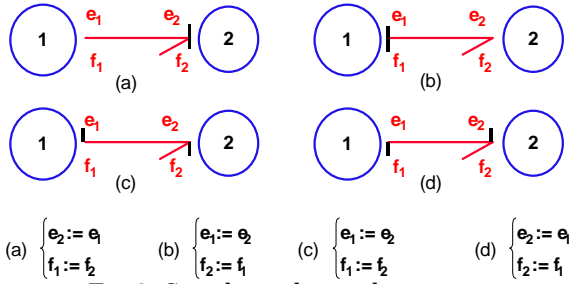


Fig. 2: Causality vs bicausality concept

More generally bicausality as causality is one way for graphically organizing the equations. The bicausality concept is an extension and a generalization of the causality one. The computational possibilities have been enlarged thereby enabling problems such as parameter identification, state estimation or system inversion to be considered with the bond graph approach.

Bicausality necessitates the definition of new elements such as a double source *SS* and a double sensor *DD* extrapolating the classical source elements *Se* and *Sf*. These elements enable the initialization of a bicausality assignment and the termination of a bicausal path in the bond graph representation. With a bicausality assignment the orientation of the model equations naturally leads to the inverse model [6-8].

I.3 Inversion and bond graph

Using inversion with bond graphs was first discussed by Gawthrop [4] and the causal configuration for inversion was proposed. A Sequential Causality Assignment Procedure for Inversion (SCAPI) that leads to an appropriate representation of inverse bond graphs model was proposed in the case of SISO system and then extended to MIMO square Systems [5,8]. Before introducing the inversion procedure, some key definitions are now given to help understand the different steps of the procedure.

Definition 1: an input-output causal path is a series of variables successively connected according to the causality assignment between an input variable and an output variable. Disjoint input-output causal paths are input-causal paths that share no variable in their series of variables.

Definition 2: the length of a set of input-output causal paths is the sum of the causal path lengths. The length of a causal path is the net number of integral operations on the path.

The bond graph based inversion procedure is given below in the case of MIMO systems [8]:

1. In the bond graph in integral causality, determine the minimal length set of disjoint input-output causal paths.
2. Propagate bicausality from the output *SS* elements to the corresponding *DD* input elements.
3. Achieve the causality assignment of the bond graph using conventional causality assignment rules.

Steps (1) and (2) of the procedure only guarantees the minimal number of differentiation necessary to solve the input (similarly to classical inversion methods). Therefore, on the inverse bond graph model, the trajectories (resulting from specifications) must be differentiable enough (structural dynamic properties) to ensure the determination of input variables. By the graphical study of the causal bond graph, we can determine easily the set of minimal length paths and their numbers but also to determine the conditions that trajectories must verified when the model is complex.

On the following academic SISO example, the procedure is applied to help the reader understand the way the inverse model is established. More information and all theoretical developments are presented and discussed in references [5-8].

Let us consider the example of a DC motor with a load (Fig. 3a). The voltage u is the input and the velocity v is the output. Step 1: the causal bond graph of the system is given in Fig. 3b. The lowest length u/v causal path, which is also the unique causal path between u and v , is represented by the dotted line (Fig. 3b). 2 *I*-type elements have an integral causality and none has a derivative causality. The length of the u/v causal path is 2. This relative degree indicates that the desired trajectory for the velocity v must be a function of time of class C^2 for solving the inverse model. Steps 2 and 3 (Fig. 3c): the *Se* and *Df* elements are respectively replaced by a *DD* element (effort and flow detector) an a *SS* element (effort and flow source). The bicausality is then propagated from the *SS* element to the *DD* element along the previous step causal path. It is worthwhile to note that the 2 *I*-type elements initially in integral causality on this causal path have changed their causality.

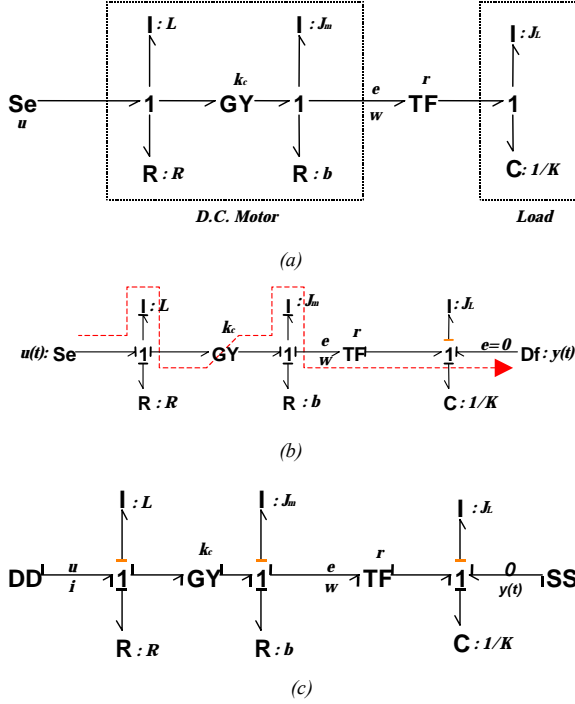


Fig. 3: Definition step by step of inverse model of DC Motor with load

The systems of equations 3 (direct state space representation) and 4 (u/v inverse state space representation) illustrate the analytical exploitation of respectively Figure 3b and 3c causal and bicausal bond graphs after a preliminary analysis of zero order loop. In particular they show how the second derivative of the desired output is necessary to compute the input and which part of the model and which parameter are necessary to compute the intermediary power variables e and ω .

$$\begin{cases}
 \dot{p}_1 = -\frac{R}{L} p_1 - \frac{k_c}{J_m} p_2 + u \\
 \dot{p}_2 = \frac{J_m k_c}{L(J_m + r^2 J_L)} p_1 - \frac{b}{J_m + r^2 J_L} p_2 - \frac{J_m r K}{J_m + r^2 J_L} q_3 \\
 \dot{q}_3 = \frac{r}{J_m} p_2 \\
 y = \frac{r}{J_m} p_2 \\
 e = \frac{k_c r^2 J_L}{L(J_m + r^2 J_L)} p_1 - \frac{b r^2 J_L}{J_m (J_m + r^2 J_L)} p_2 + \frac{J_m r K}{J_m + r^2 J_L} q_3 \\
 \omega = \frac{1}{L} p_2
 \end{cases} \quad [3]$$

$$\begin{cases}
 \dot{q}_3 = y(t) \\
 e = J_L \dot{y}(t) + K q_3 \\
 \omega = \frac{1}{L} y(t) \\
 u = \frac{L(J_m + r^2 J_L)}{k_c r} \dot{y}(t) + \left[\frac{Lb}{k_c r} + \frac{R(J_m + r^2 J_L)}{k_c r} \right] y(t) + \left[\frac{k_c}{r} + \frac{LrK}{k_c} + \frac{Rb}{k_c r} \right] y(t) + rKq_3
 \end{cases} \quad [4]$$

I.4 Application to design problems

The sizing problem can be stated at different hierarchical stages corresponding to the selection and validation of existing components. In this paper we consider only the selection and the validation of components and so the problem is to help determine the appropriate components with respect to the prescribed performances. Models resulting from inversion are used in order to give information about the physical variables of the component in a system such as effort, flow or power variables. The evolution of one (or several) typical variable(s) of a system that is (are) compatible with the trajectories of the imposed output variables can then be determined. Some performance criteria enable the definition of the trajectories that are to be followed by the load.

One way to characterise each component is to consider it by its power, its effort and its flow maximum values (and/or its maximum displacement, and/or acceleration). Compared to all these variables (functions of time) calculated on the inverse model, we obtain indications about the most appropriate component for a given technology that reaches the prescribed output objectives (Fig. 4). Thus, we can easily check whether its physical limitations are (not) reached and even crossed them momentarily on an effort-flow diagram. The key idea is to compare variables computed from the inverse bond graph model with the technical characteristics of a given component. An other significant advantage of this dynamical sizing is that it takes into account the dynamic behavior of the system.

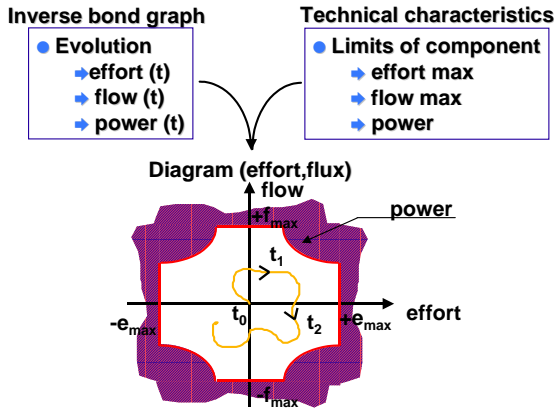


Fig. 4: Validation of components in an effort-flow diagram

II. Automotive application

The aim of the EPS system is to control the handwheel torque that the driver can feel. The handwheel torque is an essential means for the driver for informing him on the vehicle behavior and so for helping him to drive the car [9]. The sizing phase of the power system must enable to define the most adapted system for controlling this torque in all situations. This handwheel torque control will be specific restrictions of these directions as mechanic or electric power limitations need to optimize the requirements imposed by specifications. Therefore, the first step consists of defining and establishing criteria in order to describe attended objectives by the power system and by the conductor. Thus, to define pertinent specifications allows to estimate precisely energetic needs of the power system to size. For that purpose, many tests on vehicle have been made and showed that the sizing tests for the power system of the direction are:

- ✓ Parking maneuvers
- ✓ Lane change tests

We have represented in effort velocity diagram the behavior of direction system but also the specifications apply to the value of the handwheel torque for a specific handwheel speed and rack effort. For these maneuvers, the handwheel torque must be inferior to T_1 and T_2 values defined by the carmaker [10].

This diagram helps the designer to choose and to validate his actuator. However, vehicle tests show that this type of specifications is not sufficient and needs to be clarified. Our approach consists in

studying the sizing problem an dynamic and energetic aspects to define precisely our goals for the driver and for the actuator, and to improve the way the specifications are defined. In the paper, we will limit ourselves to study of the EPS in lane change test for different handwheel torque.

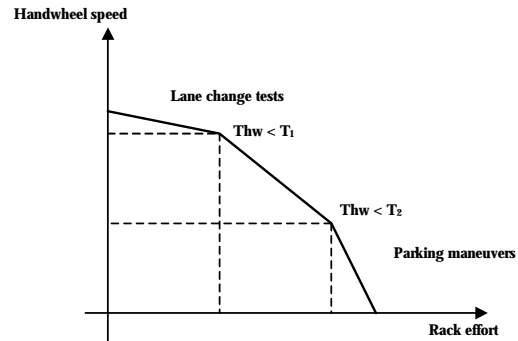


Figure. 5: EPS specifications

As we explained in previous section, the lane change test is one of the sizing test used for steering system. Therefore, we have developed a bond graph model of a car including a steering system in order to apply the methodology in this type of situations. In the next section, the vehicle model used in simulation will be briefly described and we will present the EPS system to size in more detail. Then, we will apply the proposed methodology to obtain the inverse bond graph model. Subsequently, the results of simulations will be presented followed by conclusions.

II.1 Lane change test

In our study, we consider a change lane test to size the power system of the steering system. A change lane test consists in driving the car at initial speed between many blocks without using the brake pedal and the throttle position.

We achieve this maneuver at different speeds with a driver model representing the driver's reaction to the vehicle response. All simulations are performed as closed loop steering control maneuvers. The use of a driver model allows to always obtain the same behavior of the car in this type of maneuvers and compared different technologies in the same situation. The driver model want to represent the behavior of a human driver when he arrived in front of an obstacle[11]. The driver saw to a distance d who allowed him to anticipate his position and he went then to correct the handwheel angle

accordingly. In particular, we can define the length d to limit the rate of change of the steering angle to 1000 deg/s, which approximately corresponds to the maximum rate that can be generate by human drivers.

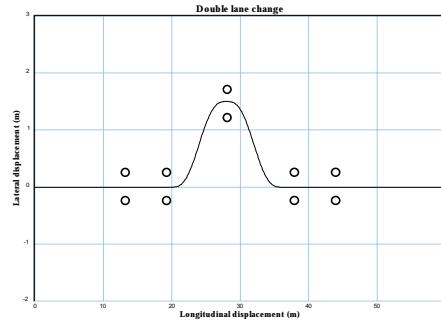
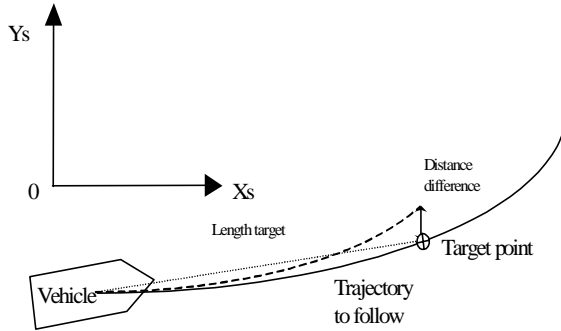
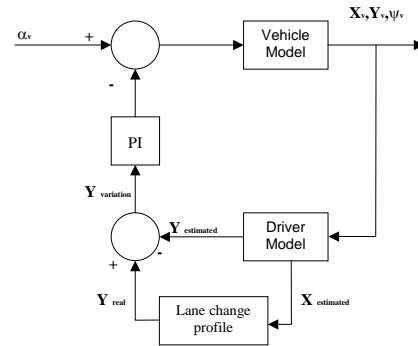


Figure.6 : Lane change test profile



(a) Principle of a driver model



(b) Schema of driver model

Figure.7 : Driver Model

II.2 Vehicle bond graph model

Hypotheses

The model has a total of 14 degrees of freedom, excluding the dynamics of subsystem. Vehicle body is modeled as a rigid body, which can perform three translations and three rotations. In our model, we suppose that all the suspensions are independent, so that each wheel has two degrees of freedom: rotation about the lateral axis and translation to the vertical axis. There are two type of inputs to the model: drivers inputs, such as steering wheel angle, brake pedal... and environmental inputs, such as the road, the road geometry (in our model, we suppose we are on a flat road situation.).

In our model, we take into account:

- ✓ The effects of suspension jacking forces. These are vertical components of the forces in lateral links of suspension. As a result, they have a tendency to

modify the height of the center of the gravity during hard cornering [12,13].

- ✓ Nonlinearities in suspension stiffness characteristics, including the bushings, and in damping characteristics modeled by look up tables.
- ✓ The Pacejka's model to represent the tires. The tire model supposes that the lateral stiffness and the longitudinal stiffness are independent [14].

II.3 EPS bond graph model

In this section, we present in detail the bond graph model of the EPS. The hardest problem has been to connect this model to the vehicle model in order to obtain the right back up value of the effort to the direction system. The modeling of the EPS takes into account the handwheel, the direction column, the power system, and the gear-rack set. Mechanical efficiencies of the speed reducer and the gear-rack set have been taken into account too. On the other hand, thermal phenomena have not been modeled for simplicity reasons

1. Back up effort modeling

The back up effort to the direction is made by kinematic caster. In our model, it is modeled by a *TF* element which is a parameter vehicle. One part of the efforts applied are due to the cornering forces f_y of the tire, the other part are due to self alignment torque M_z of the tire.

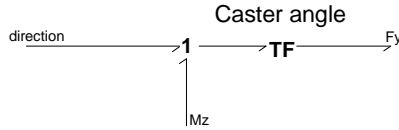


Figure.8 : Scheme of the back up effort

2. Kinematic direction modeling

The axle kinematic has been represented by the Janteaud diagram indeed for given position of the rack, the right and left wheel have a different orientation. The modeling of the kinematic recalls a crank system for having an equation of the θ pivot angle function of and the x rack displacement.

By this way, we obtain an equation $\theta = f(x)$ from which we deduce a relation between the velocity variables $\dot{\theta} = g(\dot{x}, \theta)$. The relation is represented by a *MTF*-element.

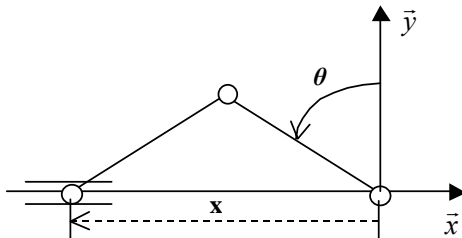


Figure.9 : Principle of a crank system



Figure.10 : Bond graph element of the axle kinematic

(1) Handwheel Sensor Column :

The handwheel inertia has been brought back to the column inertia (*I*-element). The twisting beam is represented by a *C* element. The dry frictions modeled by a *R* element and *I* element for the low part of the column.

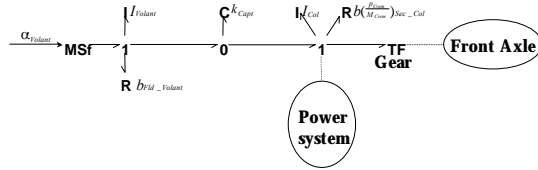


Figure.11 : Bond graph of handwheel

(2) Gear –rack set :

The gear-rack set allows to transform a translation movement in translation movement with a *TF*-element. The efficiency of the gear-rack set is taken into account in the *MR*-element ($\rho=0.9$). On the rack, we have modeled the viscous and the dry frictions (*R*-elements) and the inertia (*I*-element). The pivots are represented by a rotational inertia (*I*-elements).

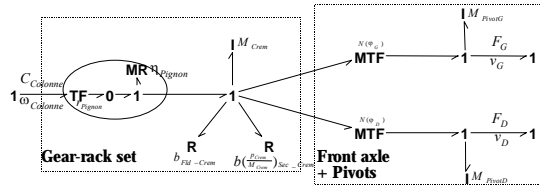


Figure.12 : Bond graph of gear-rack set

3. Power system modeling

Motor-speed reducer group :

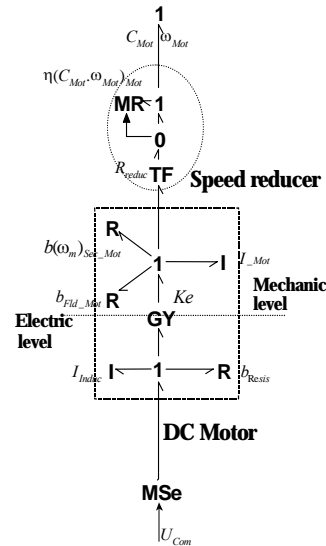


Figure.13 : BG model of a DC motor

Speed reducer :

The speed reducer part is modeled as the same way as gear-rack set with a *TF*-element and *MR*-element for the efficiency. But in a speed reducer, there is a direct and indirect efficiency when the motor is in a driving phase ($\rho=0.88$) or when the

motor is in a receiving phase ($\rho=0.7$). In this case, we use an arctangent function to allow the transition to these different states. In particular, we avoid numerical problems during simulations.

Motor:

The motor of the studied EPS is a DC-motor supplied in tension by a *Mse*-element in which electric and mechanic level are modeled. Thermal phenomena are not represented in our model. The mechanical part takes into account the mechanical inertia (*I*-element) and the viscous and dry frictions of the motor (*R*-elements). The electric part is modeled by a resistor *R*-element and the inductor *I*-element.

II.4 Inverse bond graph model

In the previous sections, we have presented the bond graph model of the car and of the direction system. As we said before, the goal of an EPS system is to control the handwheel torque in every situation with the help of a power system (here a DC motor). Using the methodology, we will show how the inverse bond graph of the steering system is established and which information is given by the structural analysis.

Step (1)

On the causal bond graph, the input *u* (motor input) represented by a *MSe*-element and the output *Cv* (handwheel torque) represented by a *DD*-element are defined respectively as the input and the output variables. The study of disjoint input and output causal paths shows a unique causal path represented by dotted line. The minimal order of the input-output is equal to three.

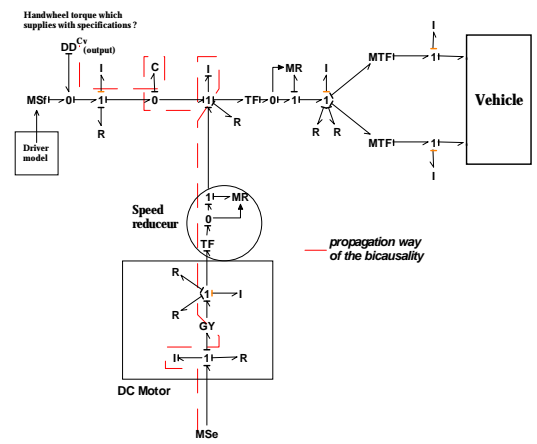


Figure.14 : Input/output causal path in the causal bond graph

Remarks :

Concerning the bond graph model of the vehicle and the bond graph model of the steering system, we can refer the reader to many publications and works on these topics[15-17].

Step (2) and Step (3)

Using bicausality and the conventional causality assignment, we obtain the inverse bond graph model (Fig.15). With the *SS*-element, we impose the handwheel torque and we remark that the *C*-element and the two *I*-elements, in integral causality (Fig.15), are now in derivative causality. The consequence is that the prescribed torque must be at least differentiable three times to solve the inverse problem.

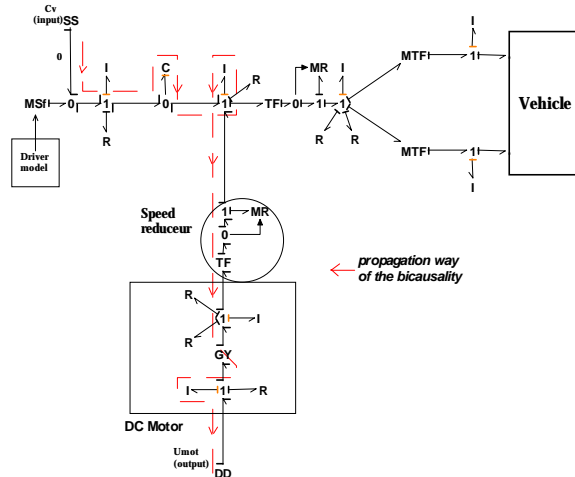


Figure.15 : Inverse bond graph model

Remarks :

In our study, storage elements on the input-output causal path are already in derivative causality (Fig.15). When we assign the bicausality, these elements stay still in derivative causality (Fig.16). The order of the input-output causal path is the number of storage element who has changed of causality between these two configurations. This order is the minimum order when a minimum number of storage elements have changed of causality. Then, we talk about the relative degree of an input/output pair to indicate the class of functions the prescribed trajectory must belong to for solving the inverse problem without obtaining Dirac function.

II.5 Appropriateness of automotive specifications to the methodology

In our approach, the main difficulty lies in defining the correct trajectory of the handwheel torque to impose to the inverse model. Indeed, the

inverse methods need to establish trajectories function of time and differentiable enough. But in our case, this type of curve is unworkable because it must be fitted to the handwheel angle at any time. The simulations results are particularly sensitive to the shape of this trajectory and the least variation gives us a large range of values. This assessment has led us to consider the problem differently.

In our case, we consider that the driver as an actuating system as the DC motor. Thus, we have two actuators that they must work together, one controlled in velocity and the other one in effort. The specifications impose that the torque apply on the driver, does not cross a given value. The proposed solution is to define a distribution function between the rack effort and the handwheel torque checking specifications.

The distribution function is given as follow :

$$C_{vm} = \frac{2}{\pi} * \arctan(\text{slope} * \frac{\pi}{2 * C_v} * \text{effort}) * C_v$$

This relation is always a function of time but time is now an implicit variable. We can not only limit a maximal torque but also define precisely the rate between the handwheel torque and the rack effort.

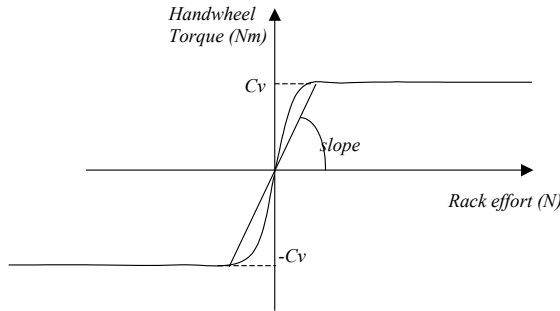


Figure.16 : Distribution function.

This device allows to extend the methodology to this type of problem which was not up to now studied. This solution gives a mean of improving these specifications and is directly usable because it needs specific curves established by the carmaker.

We obtain Fig.17 the inverse bond graph completed by the distribution function and the aim is now to determine the performances attended by the power system during a change lane test. These results must enable the designer to get a precise idea of the required power to developed by the power system whatever technology you take. Once the

technology chosen, the designer can check the appropriateness or inappropriateness between the performances and the technical characteristics of the actuating system (i.e. saturation in effort, flow or power). In particular, the inverse approach enables to study the sizing problem at different hierarchical levels and to identify mechanical or electrical components imposing limitations.

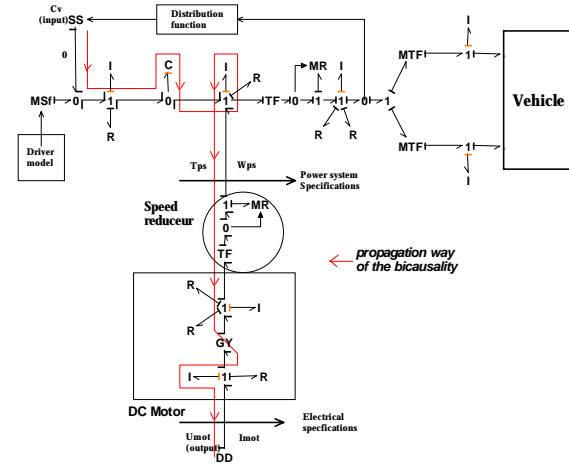


Figure.17 : Inverse bond graph model with the distribution function

III. Simulation results

In this section, we present the simulation results obtained from the inverse bond graph model on different diagrams. The lane change test is one of the tests used to size an EPS. We simulate the dynamic behavior of a car in this situation at a medium speed. The distribution function is established for growing values of the handwheel torque with the same slope.

In the effort-flow diagram (Fig.18), we obtain the dynamic evolution of the power variable (torque-angular velocity) for the lane change test. The results point out the angular velocity does not cross 14 rad/s (i.e. 800°/s) and the torque of the actuating system changes between 20Nm and 18Nm (the arrow indicates the increasing value of the handwheel torque). The mechanical power goes from 125W to 140W. All these results enable the designer to specify the performances attended by the actuating system. The dotted frame represents the mechanical characteristics of an actuating system (maximal torque, maximal angular velocity and isopower curve). The methodology gives the possibility to check easily the appropriateness between the actuator and the required performances.

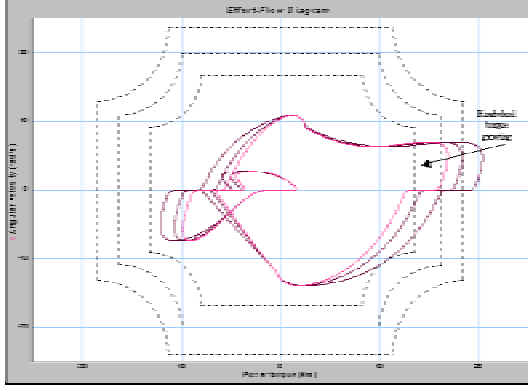


Figure.18 : Mechanical performances attended by the power system

In our study, the actuator is a DC motor and the following stage of our approach is to check his electrical performances. By this way, we can validate step by step the actuating system and identify components imposing limitations. In the diagram (Fig.19), we obtain the dynamic evolution of the power variable (Tension-intensity). The results show that the tension is around 10V and the intensity between 27A and 25A (the arrow indicates the increasing value of the handwheel torque). The electrical power goes from 240W to 270W. The dotted frame represents the electrical characteristics of the DC motor and defines the admissible functioning domain. We remark that the evolutions of the power variables remain in this area and show that the DC motor is well sized.

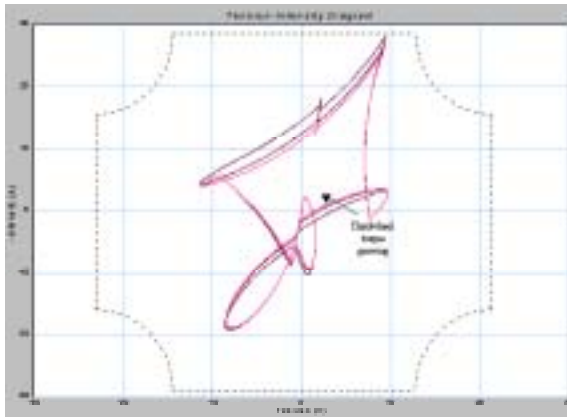


Figure.19 : Electrical performances of the DC motor

IV. Conclusion

In this paper, we have first presented a methodology for sizing mechatronic systems. This methodology is based on an inverse

approach and the bond graph language. It enables the system to be studied and to be characterized independently of the control law part. Also the inverse approach enables a progressive validation of each power stage of the actuating line without *a priori* assumption on what is to size. The bond graph language fits to the needs of the designer to study and to size system on dynamic and energetic criteria.

Then, through the automotive example, we have applied the methodology to design or to size the actuating system of an EPS system during a change lane test. We have proposed a solution to establish the input trajectory for the inverse model in a complex situation such as a change lane test. In particular, it allows not only to extend the methodology to this type of problem but although to bound easily the carmaker's specifications with the methodology. Thus, the simulation results have given precise information on the performances attended by the power system and have help the designer to establish pertinent specifications concerning the power system. Furthermore, the methodology is a mean to test and to validate an actuating system for a given objective at different hierarchical levels and to identify components imposing limitations.

References

- [1] Zanten, A. T. van., *Improvement of Road Vehicle Handling by Mechatronic Systems.*, Proceedings of 1st IFAC Conference on Mechatronic System, Darmstadt, Germany, September 18-20, 2000.
- [2] Karnopp, D. C., Margolis, D. L., Rosenberg, R. C., *System Dynamics : A Unified Approach*, New York : John Wiley & Sons, 1990, 514p.
- [3] Dauphin-Tanguy, G., *Les bond graphs*, Hermès Science Publications, Paris, 2000
- [4] Gawthrop, P. J., *Bicausal bond graphs*, Proceedings of the ICBGM'95, 2nd Int. Conf. on Bond Graph Modeling and Simulation Simulation, Las Vegas, USA, January 1995, pp83-88.
- [5] Fotsu-Ngwompo, R., Scavarda, D. Thomasset, *Inversion of linear time-invariant siso system modelled by bond graph*, J. Franklin Institute 333(B) (2) (1996) pp 157-174.

- [6] Fotsu-Ngwompo, R., S. Scavarda, D. Thomasset, *Structural inversibility and minimal inversion of multiavriable linear systems-a bond graph approach*, In Proceedings of the 1997 International Conference on Bond Graph Modeling and Simulation (ICBGM'97), Simulation Series, J. J. Granda, G. Dauphin-Tanguy (Eds.), Vol.29, Society for Computer Simulation, Phoenix, Arizona, USA, 1997, pp 109-114
- [7] Fotsu-Ngwompo, R., S. Scavarda., *Dimensioning problems in system design using bicausal bond graphs*. Simulation Practice and theory, vol. 7, 1999, pp. 577-587.
- [8] Fotsu-Ngwompo, R., S. Scavarda, D. Thomasset. *Physical model-based inversion in control systems design using bond graph representation. Part 1: theory and Part 2: Applications*. Proceedings of the IMECHE Part I Journal of Systems & Control Engineering, vol. 215, no. 2, 2001, pp. 95-103 and pp. 105-112
- [9] Badawy, A. *Modelling and analysis of an Electric Power Steering System*, Automotive Engineering, September 1998, vol 106, n°9, p25-31
- [10] Mechin, O.; W., Marquis-Favre, S. Scavarda, M. Ebalard,. F., Guillemard,. *Aide à la validation des composants d'un système mécatronique dans l'automobile* Proceedings of the SIA Congress Vehicle Dynamics 2001, Lyon, France, pp. 119-126
- [11] Afonso M., et al. *Modélisation du comortement d'un conducteur désirant suivre une trajectoire* Proceedings of the SIA Congress Vehicle Dynamics, 1993, pp. 51-58
- [12] Halconruy,T., *Les liaisons au sol*, Paris : ETAI, 1995, 200p
- [13] Wong, J;Y., *Theory of Ground Vehicles*, Wiley, New York, 1978, 435p
- [14] Bakker, E. et al., *A New Tyre Model with Application in vehicle Dynamics Studies*, Paper Society of Automotive Engineers, 890087, 1989
- [15] Bos, A. M., *Modelling multibody systems in terms of multibond graphs*, Ph. D. Thesis : Electrical Engineering :University of Twente, Enschede, Netherlands, 1986, 218p.
- [16] Karnopp, D., *Bond graph for Vehicle dynamics*, Vehicle Systems Dynamics, vol5, 1976, p.171-184
- [17] Margolis, D., Shim, T., *A Bond graph model incorporating sensors, actuators, and vehicle dynamics for developing controllers for vehicle safety*, Journal of The Franklin Institute , n°338, 2001, p.21-34