MODELING AND SIMULATION OF A HYDRAULIC POWER ASSISTED STEERING SYSTEM THROUGH BOND GRAPHS

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ABSTRACT
The hydraulic power assisted steering (HPAS) system is one of the most sensitive vehicle interfaces to the driver perception. Comfort and performance parameters such as ride, handling, tactile transfer functions and overall noise levels are directly affected by its performance. The modeling of a HPAS system using the bond graph technique makes possible the combination of hydraulic and mechanical components. This allows physical and design variables such as fluid compressibility and hoses diameters to be evaluated simultaneously. HPAS should be used as a design and tuning tool to develop different system configurations before prototype test build, representing an improvement in terms of product development time and cost for both component and vehicle levels.

Keywords: hydraulic power assisted steering, bond graph, design development, vehicle design

1. INTRODUCTION
Computer-Aided Design and system analysis aims to find mathematical models that allow emulating the behavior of components and facilities. The high competitiveness in industry, the little time available for product development and the high cost in terms of time and money of producing the initial prototypes means that the computer-aided design and analysis of products is taking on major importance. On the other hand, in most areas of engineering the components of a system are interconnected and belong to the different domains of physics (mechanics, electrics, hydraulics, thermal...). When developing a complete multidisciplinary system, it needs to integrate a design procedure to ensure that it will be successfully achieved.

Engineering systems require an analysis of their dynamic behavior (evolution over time or the path of their different variables). This is especially important in automotive products (Pacejka 1985, Martinus 1986), railway dynamics, machine tools, robotics (Gawthrop and Jones 1993, Anex and Hubbard 1984) and aeronautics. Modeling a complete system with particular attention to detail in the specific component intended for analysis enables concepts relative to the component to be analyzed as well as their influence on the rest of the system (Karnopp and Rosenberg 1968, Gordon 1969, Bekey 1977).

The purpose of modeling and simulating dynamic systems is to generate a set of algebraic and differential equations or a mathematical model. This always leads to a description of the represented system that is never ambiguous.

In order to perform rapid product optimization iterations, the models must be formulated and evaluated in the most efficient way. Automated environments contribute to this. Freeing engineers of the tedious task of producing equations is vital. In addition, this automation prevents the inevitable human error and leads to a rapid evaluation of the different alternatives of a particular component.

Although subsystems are widely modeled on component level by their manufacturers, overall HPAS system use to be initially designed based only on parameters as package and costs despite engineering design variables (Kumar et al. 1999). As result the process to reach performance targets begun lately, often ending on rework labor and efficiency loss.

In this paper a theoretical HPAS system is modeled and analyzed using the Bond Graph (C. Vera and Félez 1994, Vijayak and Barak 2002) technique as an assembly, using the components described in figure 1:

1. Fluid pump	5. Return line / fluid reservoir
2. Pressure line	6. Right tie rod
3. Steering gear box rotary valve	7. Left tie rod
4. Rack and pinion mechanism	8. Steering column

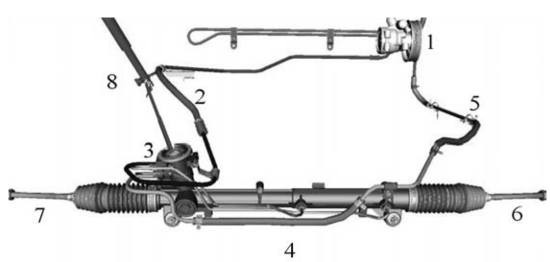

Figure 1: HPAS Assembly

2. OBJECTIVES

The main purpose of this work is to model a HPAS that works as a base for future real correlated models. Even though its theoretical behavior all system components are represented and can be tuned according to design and construction parameters. Each tuning configuration results successfully on a particular system performance.

3. BOND GRAPH MODELING

The HPAS system can be represented by using Bond Graph technique as shown in figure 2. Components are analyzed for their compliance, resistance and inductance energy elements.

The model has a source of flow *Q* that represents the fluid pump. The pressure line is represented by the elements *Kpr* and Rpr that are the radial stiffness (expansibility by volume unit) and the resistance (head loss) of the pressure hose. The return hose is represented as the pressure hose, with an element *Krt* and an *Rrt* that are the radial stiffness and the head loss of the return hose. At the end of the return hose an effort source represents the fluid reservoir.

The rotary valve is represented by four resistances (*Rli*, *Rri*, *Rlo* and *Rro*) that are the right and left, in and out passes of the rotary valve. For each resistance pair there is a compliance element (*Klc* and *Krc*) which represents the compressibility at the respective hydraulic chamber of the valve. The transformer *TFps* represents the area of the hydraulic piston section into the rack component, which converts hydraulic pressure in longitudinal force.

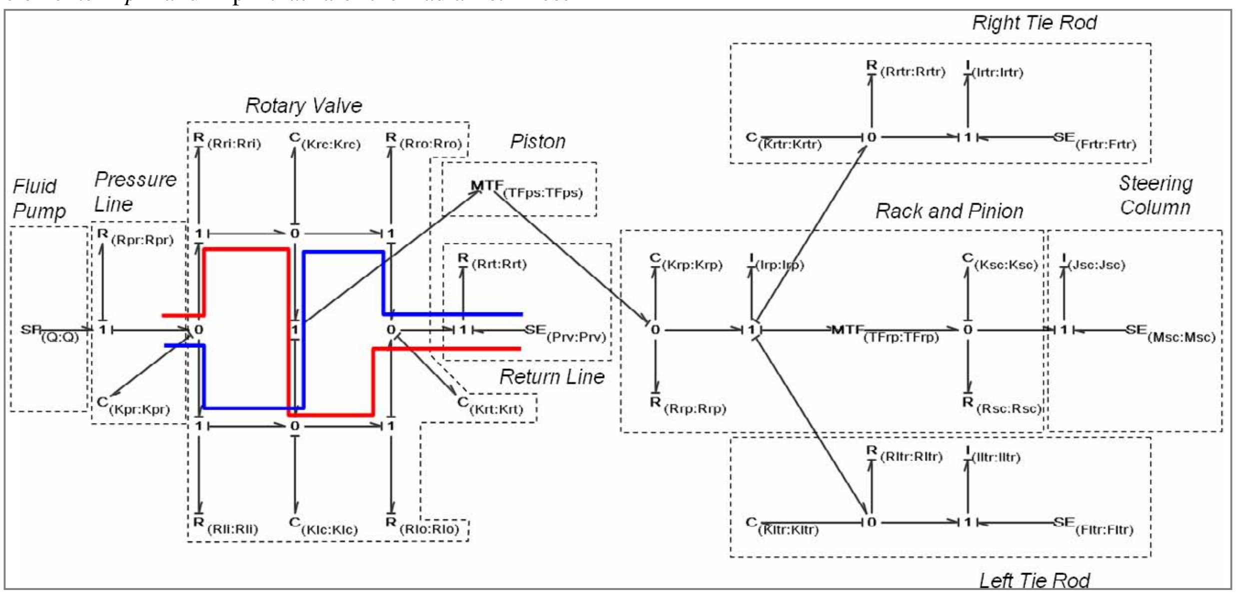

Figure 2: HPAS Bond Graph Model

The red line represents the case in which the steering system is turned to right direction. Under this condition the valves *Rri* and *Rlo* are opened, while *Rli* and *Rro* stay closed.

The blue line represents the case in which the steering system is turned to left side. Valves *Rli* and *Rro* are opened, while *Rri* and *Rlo* stay closed. Both cases are illustrated and detailed on figures 3 and 4.

Both tie rods, left and right side, receive longitudinal forces as input, represented on the model by the effort source (*Fltr* and *Frtr*); compliances elements *Kltr* and *Krtr* represents the axial stiffness of each tie rod. *Rltr* and *Rrtr* represent the resistances on the unions between the tie rods and the knucles of the vehicle. Iltr and Irtr represent the mass of the tie rods.

Rack and pinion mechanism is represented by the transformer *TFrp* that transfers steering wheel rotational motion on the steering column to longitudinal motion on the steering rack component. *Krp* represents the stiffness of the rack, *Rrp* the resistance of the mechanism and Irp the mass of the assembly.

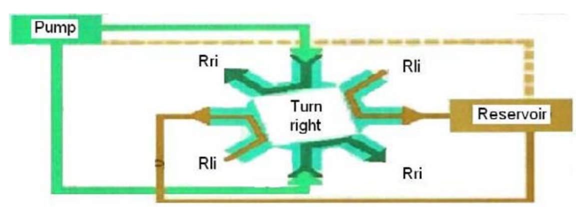

Figure 3: Rotary Valve Turning to Right

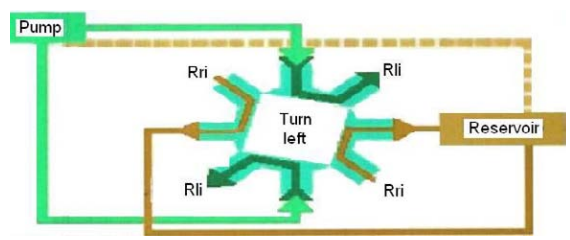

Figure 4: Rotary Valve Turning to Left

The steering column torsion stiffness is represented by the *Ksc* element. *Rsc* represent the resistance on the steering column, and *Jsc* the moment of inertia of the steering column and wheel assembly. The effort source *Msc* represents the momentum applied by the driver on the steering wheel.

4. RESULTS

As the theoretical model has the purpose of a model design technique developing, parameters were chosen aiming turn easy the understanding of the results. Total simulation time is of 10s. The list below shows all variables and values.

Therefore, it is possible to obtain the results in a simple way by evaluating flows and efforts that join and connect the components of the model. To obtain the simulation of the model, *Bondin* © software will be used (Romero et al. 2009). This software allows obtaining the evolution of the characteristic parameters of the model as well as letting them be compared.

To carry out the model validation, the values of the parameters used in the simulation are listed in the following table.

Table 1: Parameter values

Kpr = 100.000 N/m^3	Rrtr = 1 (N·s)/m
Rpr = 0.0001 (Pa·s)/m^3	Irtr = 0.001 kg
Krc = 100.000 N/m^3	Frtr = 0 N
Klc = 100.000 N/m^3	Kltr = 100000 N/m
Krt = 100.000 N/m^3	Rltr = 1 (N·s)/m
Rrt = 0.0001 (Pa·s)/m^3	Iltr = 0.001 kg
Prv = 0 Pa	Fltr = 0 N
TFps = 1	TFrp = 0.5
Krp = 100.000 N/m	Ksc = 10000 (N·m)/rad
Rrp = 1 (N·s)/m	Rsc = 0.1 (N·m·s)/rad
Irp = 0.001 kg	Jsc = 0.0001 kg·m^2
Krtr = 100000 N/m	Msc = 0 N·m

In order to simulate the behavior law of the valve and the beginning of the fluid pump, different values of the necessary resistances and flow *Q* has been introduced.

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if ((t>0) and (t<=5)){ Q=1.2^t }
if ((t>5) and (t<=10)){ Q=1.2^(10-t) }

if (t<=5){ Rri=10^-4 }
if ((t>5) and (t<=10)){ Rri =10^3 }

if (t<=5){ Rro=10^3}
if ((t>5) and (t<=10)){Rro=10^-4 }

if (t<=5) { Rli=10^3}
if ((t>5) and (t<=10)){ Rli=10^-4 }

if (t<=5) { Rlo=10^-4}
if ((t>5) and (t<=10)){ Rlo=10^3 }
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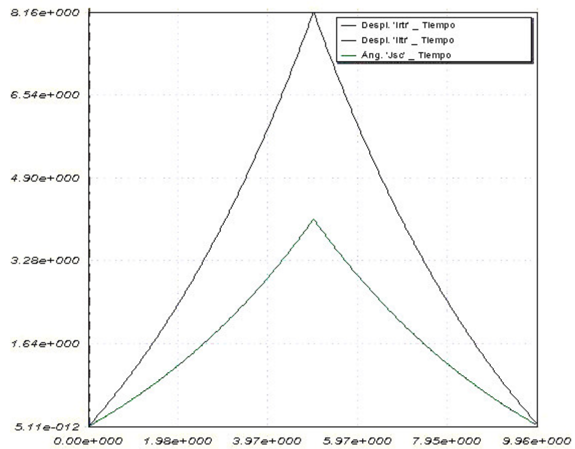
High stiffness values and low resistance and inductance values were used aiming approximate the model of an ideal configuration, with minimum losses.

Fluid flow (*Q*) increases from 0s to 5s, and after this decreases on the same rate until 10s. The rotary valve, controlled by their resistances, is turned right from 0s until 5s, and after this turned left until 10s.

The response of the system to the flow input was measured on the tie rods and steering column as displacement and turning angle. Following figure shows the plotted results.

Displacement [m]
Angular Displacement [rad]



time [s]

Figure 5: Tie Rod Displacement and Steering Column Rotation on Time

Hydraulic fluid accumulated volume is shown on table 2 with 1s of interval between 0s and 10s.

Table 2: Incremental Supplied Flow

Time [s]	Volume [m^3]
0	0.00
1	1.20
2	2.64
3	4.37
4	6.44
5	8.93
6	11.00
7	12.73
8	14.17
9	15.37
10	16.37

As expected, as 16.37m^3 was input along 10s, and considering that *TFps* = 1, what means that the area of the piston is 1m^2, both tie rods were displaced until 8.16m at 5s and came back to zero position at the end of the 10s, as shown on figure 5.

Steering column showed a proportional (1:2) rotation when compared with tie rods displacement due to *TFrp* = 0.5 which represents the rack and pinion mechanism relation.

Slight differences between model and calculated results (0.3% on this case) are expected due to system stiffness properties considered on the model.

5. CONCLUSIONS

The displacements and rotation found on tie rods and steering column shows that the model has worked as expected.

Rotary valve model have successfully directed hydraulic fluid flow, resulting on a simultaneously movement between both tie rods and steering column rotation.

Pressure and return lines should be separated on different parts on the model, each one with its own compliances and resistances simulating different head losses of the hoses. As this is not the focus of this paper, the assumption used was to do not model these variables.

Compliances, resistances and inductances must have their values upgraded to real magnitudes on a perspective of correlation and validation of a real system based on this theoretical model, which can be used as a initial step on the development and validation of a real based model.

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