

Vehicle Simulation for the Development of an Active Suspension System for an Agricultural Tractor

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ABSTRACT

The design of suspension systems for heavy-duty vehicles covers a specific field of automotive industry. The proposed work focuses on the design development of a front controllable suspension for an agricultural tractor capable to satisfy the system requirements under different operating conditions. The design of the control algorithms is based on the developed multibody model of the actual tractor, including the pitch motion of the sprung mass, the anti-dive effects during braking and forward-reverse maneuvers and the non-linear dynamics of the actuation system. For an advanced analysis, a novel thermo-hydraulic model of the hydraulic system has been implemented. Several semi-active damping controls are analyzed for the specific case study. Therefore, the most promising damping strategy is integrated with other control functions, namely a self-leveling control, an original control algorithm for the reduction of the pitch motion, an anti-impact system for the hydraulic actuator and an on-line adaptation scheme, which preserves an optimal damping ratio of the suspension, even against large variations in operating conditions.

INTRODUCTION

The main role of suspension systems is improving vehicle ride comfort and handling [1]. For off-highway vehicles there are other significant requirements such as the adaptation of the vehicle height, the minimization of the suspension working space (i.e. the suspension elongation) and the reduction of the infrastructure damage [2],[3]. A passive suspension system with constant characteristics (i.e. stiffness and damping factors) would hardly satisfy all these conflicting requirements, therefore, a wide number of controllable suspensions have been proposed ([2],[4],...,[11]). Suspension systems can be classified into passive and active systems. The active systems can be further divided into two categories: fully active and semi-active systems. The fully active, by means of an electro-hydraulic actuator, have the ability to dissipate, store and introduce energy into the vehicle system. Semi-active suspensions employ dampers or springs with variable characteristics to generate the control force. Such type of devices can only absorb and dissipate the energy during the motion of the vehicle body. These systems can be classified according to the damper configuration used: dampers with a controllable orifice, dampers with controllable fluids (electro-rheological and magneto-rheological) and friction dampers. Among the many

different types of controlled suspensions, semi-active are recognized as the best compromise between cost and performance for the off-highway vehicles: they only employ shock absorbers with controllable damping coefficient. On the other hand, the main limit of semi-active suspensions is the "passivity constraint": it is impossible to deliver forces on the body having the same direction of the suspension elongation speed [5]. This work is focused on the development of controllable suspension systems for heavy-duty vehicles, in particular for agricultural tractors. In this field, the research activity is not complete, as confirmed by the lack of scientific literature and for the few examples of commercial application for this kind of vehicles present in the market. For off-highway vehicles, the load conditions on the front and rear axles can vary considerably and have an effect on the dynamic behavior of the vehicle. Moreover, in many cases (such as tractors in agriculture), only the front axle is provided with a suspension, while the rear one is rigid [3]. Typical applications of suspensions in off-highway industry include the cabin suspension (known as secondary suspension system) and the front axle suspension (known as primary suspension system) [2]. Up to now, the performance improvements have been reached through new solutions developed for the secondary systems, while the primary systems are generally implemented with passive systems, due to economical motivations and their limited energy demand. Obviously, such technical solutions partially satisfy the system requirements. Consequently, the employment of a controllable suspension system is a potential way to reach the desired dynamic performances [3],[6],[7],[8]. In particular, due to the big variety of load and operating conditions, some additional control features (i.e. leveling and pitch control) could drastically improve the vehicle performances. As a matter of fact, the largest manufacturers of agricultural equipment are looking for solutions to control suspension stiffness and damping, technique widely used in the automotive industry. The main objective of this study consists in the design of a cost-effective controllable suspension for a primary suspension system, capable to adapt the dynamic behavior of an agricultural tractor, under different road and load conditions. This goal is reached by an optimized blending of known and innovative control algorithms.

The paper is structured as follows. The first part deals with the multiphysics (mechanical and thermo-hydraulic) model of the tractor. The frequency responses of the overall vehicle subjected to road inputs are analyzed in different conditions. In the second section the application of different semi-active algorithms to the specific case study are described, with particular attention both to the damping laws and the sensor location issues. The results are delivered in the form of frequency response and time response plots. Indeed, a fully functional suspension system should provide not only ride comfort and handling enhancement. Therefore, as explained in the third section, the damping control is integrated with additional features: self-leveling control, pitch control, an adaption scheme of the damping ratio and an original anti-impact system for the hydraulic actuator. The final

section of the work deals with typical simulations to evaluate the overall system performances, such as maneuvers with various profiles in terms of vehicle velocity and load conditions.

VEHICLE MULTIBODY MODEL

The study of a controllable suspension requires a suitable prototype capable to describe the actual vehicle dynamic. From this point of view, the employment of an agricultural tractor represents the best choice. On the other hand, such solution would be very expensive, time consuming and unsafe. Moreover, the experimental study would require specific components, often not readily available. In order to speed up the control evaluation process, a multibody model of the actual tractor has been developed, which includes the pitch and heave motions of the sprung mass, the vertical displacement of the front unsprung mass and the anti-dive effects during braking and forward-reverse maneuvers. Such virtual simulator includes a thermo-hydraulic model and the non linear dynamics of the actuation system (i.e. friction phenomena and the mechanical end-stops). The multiphysics model, developed in Matlab environment has been calibrated and tested by experimental data ([3],[12],[13]), also by using AMESim/Matlab co-simulation technique.

MULTIBODY MODEL - The dynamic behavior of an agricultural tractor with a front suspended axle can be described by a 3 Degrees of Freedom (DoF) model, composed of two principal bodies (Figure 1): the first is the sprung mass of the tractor (m_s) and the second body represents the unsprung body (unsprung mass m_{us} , two wheels and front axle, Figure 2). The longitudinal motion of the wheel centre due to chassis and suspension motions is not included being negligible with respect to other DoF. The suspension consists of a rigid axle connected to the sprung body of the tractor through an articulated quadrilateral mechanism. The suspension control arms are described by a fictitious rod connecting the unsprung body center of mass to its Instantaneous Center of Rotation (ICR), as shown in Figure 2. The moment of inertia of the unsprung body is neglected.

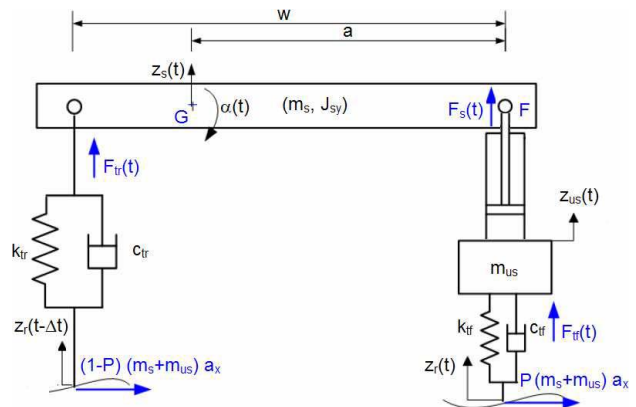


Figure 1: Tractor multibody model

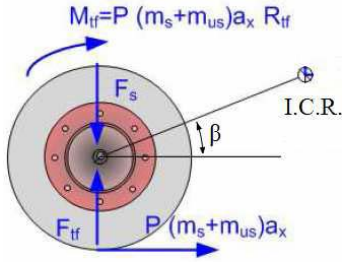


Figure 2: Schematic of the unsprung body: fictitious rod and Instantaneous Center of Rotation (ICR)

The equations of motion are given by:

$$\begin{cases} m_{us}\ddot{z}_{us} = [m_{us} - P(m_s + m_{us})] a_x \tan(\beta) - F_s + F_{tf} \\ m_s\ddot{z}_s = -[m_{us} - P(m_s + m_{us})] a_x \tan(\beta) + F_s + F_{tr} \\ \frac{J_g\ddot{\alpha}}{a_x} = [m_{us} - P(m_s + m_{us})] \tan(\beta)(b_x + a) - m_s h_g - m_{us} R_{tf} + \frac{F_{tr}(w-a) - F_s a}{a_x} \end{cases} \quad (1)$$

Where a_x represent the longitudinal acceleration of the vehicle, P the ratio between the front axle and the total traction forces, h_g the height of the sprung body center of mass and R_{tf} the nominal radius of the front wheels. The force delivered by the suspension F_s is defined by the thermo-hydraulic model of the actuation system, as explained in the following section.

HYDRAULIC SUSPENSION SYSTEM: OPERATING PRINCIPLES - In Figure 3 it is shown the hydraulic schematic of the suspension system [14]. The hydraulic connections are:

- P port = hydraulic pressure line from the tractors hydraulic system;
- T port = tank line (return line);
- LS port: Load Sensing signal line towards vehicles hydraulic system.

When suspension mode is selected (proportional solenoid valves 3 and 4 energized), the pressure in the system will balance the external load acting on the cylinder. Damping effect is provided by the damping units (7,8). The load leveling function compensates the suspension deflection induced by front axle load changes. By doing so, the suspension system will always have the same stroke available for the bound (compression) or rebound (extension) action. The load leveling system will level the suspension by the information that is provided from the displacement sensor installed on the front axle. The system will raise the tractor by activating the raise solenoid valve (solenoid valve 1) and lower it by activating the lowering one (solenoid valve 2). Additional damping features (i.e. pitch control, anti-impact control and the damping adaption scheme) are implemented by acting on the current (PWM control) to proportional solenoid valves (3,4).

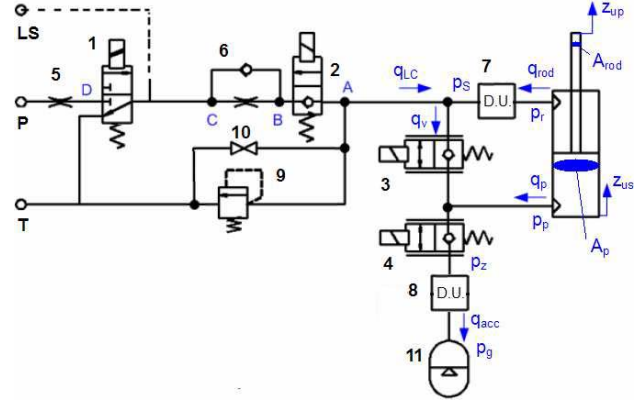


Figure 3: Schematic of the hydraulic suspension unit. 1-Raise solenoid valve (3-way, 2-position), 2-Lowering solenoid valve (2-way, normally closed), 3-4-Lockout (proportional) solenoid valves (2-way, normally closed), 5-Raise orifice, 6-Lowering orifice and check valve, 7-Rebound damping unit, 8-Compression damping unit, 9-High pressure relief valve, 10-Manual override, 11-Hydraulic accumulator.

THERMO-HYDRAULIC MODEL OF THE ACTUATION SYSTEM - The thermal dynamics of the hydro-pneumatic suspension unit can be firstly described by three variables: temperature, pressure and volume of the gas in the accumulator. The thermo-hydraulic model is obtained by applying the first law of thermodynamics, the law of mass conservation and the ideal gas law:

$$\begin{cases} \frac{dT_g}{dt} = \frac{K_{acc}(\gamma-1)}{Rn} (T_{env} - T_g) - \frac{T_g(\gamma-1)}{V_g} \frac{dV_g}{dt} \\ V_g(t) = V_g(0) - \int_0^t q_{L.C.}(t) dt - A_{rod}(z_{up} - z_{us}) \\ p_g(t) = \frac{nRT_g(t)}{V_g(t)} \end{cases} \quad (2)$$

where:

- p_g, V_g, T_g = pressure, volume and temperature of the gas in the accumulator;
- T_{env} = environmental temperature;
- K_{acc} = heat transfer coefficient (global);
- R = gas constant;
- n = amount of gas (moles);
- γ = polytrophic index.

Figure 4 shows the transient response of the suspension unit when the hydraulic cylinder is subjected to a step displacement. As expected, the process approaches firstly an adiabatic law, while the steady state behavior can be described by an isothermal process. In fact, during fast processes temperature compensation does not occur and after an adequately long period of time the isothermal process reflects the state of the system. The thermal model has been further enhanced by adding another DoF that is the temperature of the hydraulic system (hydraulic fluid, hydraulic cylinder, valves and manifold):

$$\begin{cases} C_{idr} \frac{dT_{idr}}{dt} = K_{acc-idr}(T_g - T_{idr}) + \sum_i W_{diss,i} \\ \quad + K_{idr-env}(T_{env} - T_{idr}) \\ C_g \frac{dT_g}{dt} = K_{acc-idr}(T_{idr} - T_g) - p_g \frac{dV_g}{dt} \\ \quad + K_{acc-env}(T_{env} - T_g) \\ V_g(t) = V_g(0) - \int_0^t q_{L.C.}(t) dt \\ \quad - A_{rod}(z_{up} - z_{us}) \\ p_g(t) = \frac{nRT_g(t)}{V_g(t)} \end{cases} \quad (3)$$

where T_{idr} is temperature of the hydraulic unit, K_{ij} is the heat transfer coefficients at the i/j interface, $W_{diss,i}$ represents the power dissipated by the i^{th} component (hydraulic resistances, friction phenomena in hydraulic cylinder,...) and C_{idr} and C_g are respectively the heat capacities of hydraulic unit and accumulator gas.

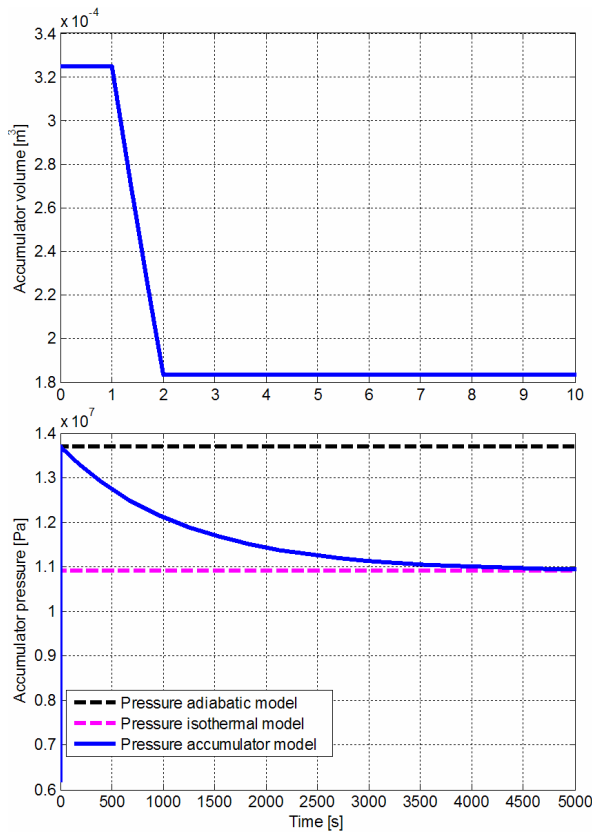


Figure 4: Transient response of the suspension unit thermo-hydraulic model (1 DoF)

FRICTION MODEL - The non linear behavior of the suspension system is partially due to the friction forces acting on the piston-cylinder assembly. For a hydraulic system, this phenomenon also depends on pressure in the cylinder chambers. In particular, experimental evidence of the hysteresis phenomena suggests to employ a Coulomb model characterized by a friction force magnitude dependent on the accumulator pressure. As shown in Figure 5, for small sweep amplitudes the friction forces acting on the piston-

cylinder assembly lock the suspension in the low frequency range.

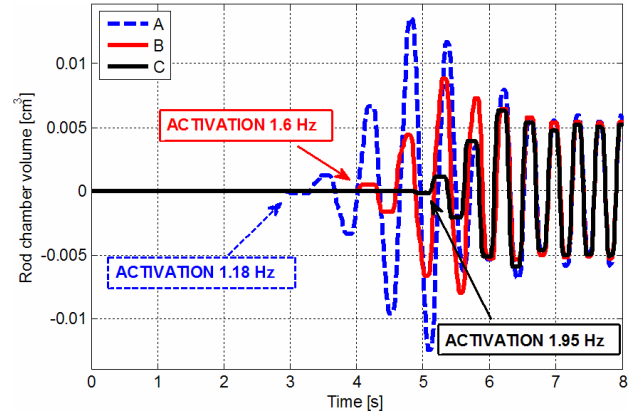


Figure 5: Rod chamber volume of the hydraulic cylinder when the vehicle is subjected to a sweep input profile (different friction force magnitude: low (A), nominal (B) and high (C))

The developed multiphysics model permits to evaluate the vehicle dynamic behavior for different operating conditions: the Frequency Responses (F.R.) of the vehicle are presented in the following sections. Furthermore, such virtual simulator is used as a design tool during the development process of the different control features, namely (adaptive) damping control, leveling, pitch and anti-impacts features.

SEMI-ACTIVE CONTROL STRATEGIES

A simplified model of the vehicle dynamics (Quarter Car Model - QCM, Figure 6) was used to develop the semi-active Skyhook, Groundhook and Hybrid control strategies, widely described in [3],[4],[5],[9],[10],[11],[15].

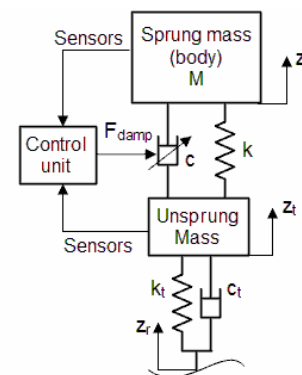


Figure 6: Quarter Car Model

The Skyhook concept employs a fictitious damper located between the sprung mass and the stationary sky as a way of suppressing the vibratory motion of the sprung mass. The Skyhook control reduces the resonant peak of the sprung mass quite significantly and thus achieves a good ride quality without increasing the

transmissibility at high frequencies. The Groundhook control assumes an additional virtual damper between the unsprung mass and the ground to decreasing the dynamic tire force. Hybrid strategy merges the advantages of Skyhook and Groundhook controls, therefore can offer benefits to both sprung and unsprung mass dynamics. It is possible to implement Skyhook, Groundhook and Hybrid strategies by measuring different signals (sprung and unsprung mass accelerations and suspension elongation) and controlling a variable damper placed between the two bodies, as shown in Figure 6. In this work a linear approximation of the Skyhook concept (CVD, Continuously Variable Damper) is applied: the damping value $c(t)$ can be given in the interval $[c_{min}, c_{max}]$. The control algorithm is given by:

$$\begin{cases} c(t) = \text{sat}_{[c_{min}, c_{max}]} \left[\frac{\alpha c_{max} (\dot{z} - \dot{z}_t) + (1-\alpha) c_{max} \dot{z}}{\dot{z} - \dot{z}_t} \right] \\ \quad \text{if } \left[\frac{\alpha c_{max} (\dot{z} - \dot{z}_t) + (1-\alpha) c_{max} \dot{z}}{\dot{z} - \dot{z}_t} \right] \geq 0 \\ c(t) = c_{min} \text{ if } \left[\frac{\alpha c_{max} (\dot{z} - \dot{z}_t) + (1-\alpha) c_{max} \dot{z}}{\dot{z} - \dot{z}_t} \right] < 0 \end{cases} \quad (4)$$

Where sat_g is a saturation operator. The typical value for α is 0.5 that corresponds to equally divide the damping coefficient between the sprung body and the wheel. The approximation of the Groundhook concept, here used, is given by ([11],[15]):

$$\begin{cases} c(t) = c_{max} \text{ if } \dot{z}_t(\dot{z} - \dot{z}_t) \leq 0 \\ c(t) = c_{min} \text{ if } \dot{z}_t(\dot{z} - \dot{z}_t) > 0 \end{cases} \quad (5)$$

Finally, the Hybrid concept is implemented by Eq. 6, where the variable α is the relative ratio between the Skyhook and Groundhook components of the damping force ([11],[15]).

$$\begin{cases} c_{sky}(t) = c_{max} \text{ if } \dot{z}(\dot{z} - \dot{z}_t) \geq 0 \\ c_{sky}(t) = c_{min} \text{ if } \dot{z}(\dot{z} - \dot{z}_t) < 0 \\ c_{gnd}(t) = c_{max} \text{ if } \dot{z}_t(\dot{z} - \dot{z}_t) \leq 0 \\ c_{gnd}(t) = c_{min} \text{ if } \dot{z}_t(\dot{z} - \dot{z}_t) > 0 \\ c(t) = \alpha c_{sky}(t) + (1 - \alpha) c_{gnd}(t) \end{cases} \quad (6)$$

A conventional passive isolator can only be influenced by local variables such as positions, velocities or accelerations. Although the concept of semi-active control has been illustrated above using only a very basic two DoF system, it is important to note that in a multi DoF system the damper can be made to react to signals originating at locations remote from the damper position. In particular, it is possible to use accelerometers to measure the pitch and heave accelerations and to command forces which damp the pitch and heave motion, therefore reacting to absolute rather than relative motion. Obviously, such solution would be optimal if both the front and rear axles are provided with a suspension system. In this case the damping ratios and the relative resonances of the pitch and heave motions are independently adjustable. For the particular case study, due to plant-equipment constraints (Figure 7), it is only possible to drive the damper in order to obtain a

compromise between pitch and heave motion control objectives. The semi-active strategy, called "Mix strategy", is given in Eq. 7 according to the passivity constraint. Two different weight coefficients, b_1 and b_2 , are applied in order to decide how much the strategy damp pitch or heave motions and the relative damping ratios.

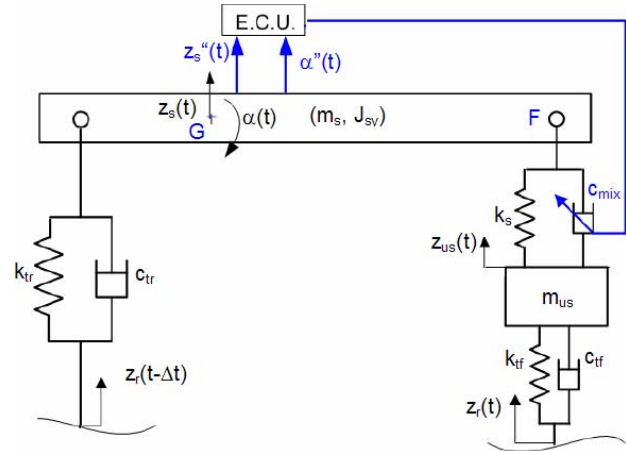


Figure 7: Schematic for the control of pitch and heave motion (tractor with a front axle equipped with a controllable damper)

$$\begin{cases} F_{damp}(t) = b_1 \dot{z}_s - b_2 \dot{\alpha}_s \\ c_{mix}(t) = \frac{F_{damp}(t)}{v_{cil}} \text{ if } F_{damp}(t)v_{cil} \geq 0 \\ c_{mix}(t) = c_{min} \text{ if } F_{damp}(t)v_{cil} < 0 \end{cases} \quad (7)$$

SEMI-ACTIVE CONTROL PERFORMANCE - The different semi-active controls are tested and compared by means of the developed multiphysics model described in the previous section. In particular, in order to assess the system performances different objectives are used in the evaluation:

- Comfort objective: the minimization of the RMS (Root Mean Square), APC (Average Power Contents) and RMSF (Root Mean Square Frequency - weighted, [16]) values of the body accelerations and the reduction of the transmissibility magnitude between road profile and body accelerations (pitch and heave motion).
- Handling objective (driveability): the reduction of the transmissibility magnitude between road profile and dynamic tire forces. Moreover, the minimization of the RMS value of dynamic tire contact forces and tire displacements.
- Working space requirement: the reduction of the transmissibility magnitude between road profile and suspension elongation. Moreover the minimization of the RMS value of the suspension elongation.

The system measurements are the sprung or unsprung mass accelerations and the suspension elongation (Figure 6). From this point of view, the sensor location issue is not a trivial problem. In particular, in order to