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Simulation of a Vehicle's Operation Controlled by Active Suspensions

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ABSTRACT

The current cars are designed to go more quickly and they must adapt to all extreme situations of urgencies to avoid accidents. Active systems of safety already exist to reduce the task of the drivers but several investigations are still made. The purpose of this research carried out is to improve the existing systems or to propose ones more effective. In this work, a proposal is made to improve control by using the suspensions. These suspensions are controlled by a synthesized LQ law which proved its effectiveness on the braking distance of a vehicle. These same suspensions are tested in this work for an operation of the roundabout type. For this, a model of vehicle in three dimensions is presented to simulate this configuration of control.

Key words: Vehicle dynamics, load transfer, vibratory comfort, steering angle, vehicle's trajectory

INTRODUCTION

For a few years, new active systems of safety have appeared on the market. Among them, the systems of control dynamic of stability (ABS, ESC...) mark a projection in the field of active safety, mainly for the vehicles whose spinning of the driving wheels involves instability of the vehicle (Cho *et al*., 2008).These instabilities on the vehicles are hardly controllable by a driver with a lack of experience. A good balance of the frame and motricity by the nose gear wheels guarantee the stability of the vehicle in most of the driving situations (Baffet *et al*., 2008).

In the same objective, several works were proposed. Among them, Chu and Jones, 2008 investigated steady-state nonlinear cornering behaviour of vehicles under lateral load transfer. To this end, a criterion has been proposed to demonstrate how the handling behaviour is changed in the high lateral acceleration region. Several methods exist to control the frame of a vehicle. Andreasson and Bunte (2006) proposed to approach Global Chassis Control (GCC) where the available degrees of freedom are used to execute the desired vehicle motion while minimizing the utilization of the tyre's grip potential.

In certain critical situations caused by a deformation of the roadway or an excessive speed for example, it can happen that the vehicle will be more difficult to control. Under these conditions, Demic (1994) minimized the vibratory loads. It optimizes simultaneously the vertical vibrations of the driver's seat, vibrations of the steering wheel and normal reactions in the contact surface of the tyre and road.

The presence of an active suspension system helps to the control of the vehicle. This system of suspension synthesized was tested by Ori *et al.* (2011) to improve the braking distance. In this study, this law is tested in another configuration of control.

It is often unrealistic to assume that all vehicle states and the disturbances acting on it can be measured (Venhovens and Naab, 1999). This is why simulation is useful to have information on the evolution of the vehicle's states during an operation. Many different vehicle models have been developed for use in various vehicle control systems. An accurate and realistic vehicle model is essential for the development of effective vehicle control systems (Shim and Ghike, 2007).

To this end, a complete model of vehicle is presented. It includes 22 degrees of freedom and just in a realistic way to all vehicle dynamics. An operation of the round point type is simulated in this project to test the effectiveness of the law of control.

Model and equations of the vehicle

Rigid mass of the vehicle: This model (Ori, 2001) includes the rigid mass of the vehicle's body and the four unsprung masses. It has 22 degrees of freedom, (x_G, y_G, z_G, θ, φ, ψ) for the vehicle's body, (x_{ij}, y_{ij}, z_{ij}) for each unsprung mass and (w_{ij}) for each wheel in rotation. We can simulate the dynamic behaviour of a vehicle in real time with this model (Fig. 1). Thus, it facilitates experimentation on controlled frame systems using a control simulator, integrating the reactions of a real driver.

The indication i = 1 is for the front wheel-axle unit, i = 2 is the rear wheel-axle unit. The indication j = 1 is for the left side and j = 2 for the right side.

To determine the general equations of the model's movements in 3D, we used Lagrange equations. Thus the equations of the vehicle's body are:

$$I_{x'x'} \frac{\partial^2 \theta_G}{\partial t^2} = \sum_{i=1}^{2} (l_{i1} F_{zi1} - l_{i2} F_{zi2}) + \sum_{i,j=1}^{2} h_{ij} F_{yij} + M.g.h \times \sin\theta$$

$$I_{y'y'} \frac{\partial^2 \varphi_G}{\partial t^2} = \sum_{j=1}^{2} (L_2 F_{z2j} - L_1 F_{z1j}) - \sum_{i,j=1}^{2} h_{ij} F_{xij}$$

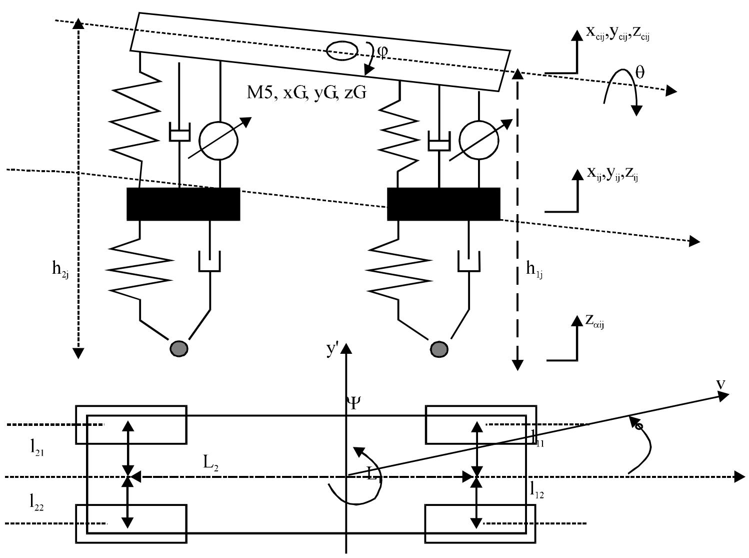

Fig. 1: Model of the complete vehicle

$$I_{z'z'}\frac{\partial^2 \psi_G}{\partial t^2}=\sum_{i=1}^{2}(l_{i2}F_{xi2}-l_{i1}F_{xi1})+\sum_{j=1}^{2}(L_1F_{y1j}-L_2F_{y2j})$$

For the unsprung masses (i, j = 1, 2), we can write:

$$Mns_{ij}\,\ddot{z}_{ij}=-C_{sij}(\dot{z}_{ij}-z_{cij})-k_{sij}(z_{ij}-z_{cij})-k_{pij}(z_{ij}-z_{pij})-F_{aij}$$

where, F_{aij} represents the active force element to be controlled, F_{xij}, F_{yij}, F_{zij} represent the suspension forces under the wheels according to (Gx'), (Gy'), (Gz') axis of reference R'(G, x', y', z') related to the vehicle mass center C_{sij}, represents the damping of the secondary suspensions, k_{si} and k_{pi} are the stiffness of the secondary suspensions and the primary suspensions, respectively. $I_{x'x'}$, $I_{y'y'}$, $I_{z'z'}$ are the inertias of the rigid body according to the axes of the R'(G, x', y', z') reference related to the mass center of the vehicle, respectively. h_{1j} and h_{2j} are the vertical distances from the ground to the axis of rolling at the front wheel-axle unit and the rear wheel-axle unit, respectively and h represents the vertical distance from the ground to the center of gravity of the unsprung mass. L_1 and L_2 are the horizontal distances from the gravity center to the axis of rolling at the front wheel-axle unit and the rear wheel-axle unit, respectively and l_{ij} are the distances between the rolling axis and the vertical that which passes through the gravity center of each unsprung mass. Ms and Mnsij are the masses of the rigid body and the unsprung masses ij, respectively. θ, φ and ψ are the angle of roll, the angle of pitch and the angle of yaw of the rigid body, respectively. z_{pij} is the vertical road displacement and g is gravity.

The equations of the inertia center of the vehicle's movement which are obtained by the fundamental relation of dynamics are:

$$M.\,\ddot{x}_G-M.\dot{\psi}.\dot{y}_G=\sum_{i,j=1}^{2}F_{xij}$$

$$M.\,\ddot{y}_G+\dot{\psi}\,\dot{x}_G=\sum_{i,j=1}^{2}F_{yij}$$

$$M.\ddot{z}_G=\sum_{i,j=1}^{2}F_{zij}-M\times g$$

with

$$M=M_S+\sum_{i,j=1}^{2}Mns_{ij}$$

The velocities of the gravity center in the reference R' are expressed by:

$$\begin{cases}\dot{x}_G=V.\cos\delta\\ \dot{y}_G=V.\sin\delta\end{cases}$$

and accelerations after a linearization are:

$$\begin{cases} \ddot{x}_G = \dot{V}\,(1 - \frac{\delta^2}{2}) - V\,\dot{\delta}\,\delta - \dot{\psi}\,V\,\delta \\ \ddot{y}_G = \dot{V}\,\delta + V\,\dot{\delta} + \dot{\psi}\,V \end{cases}$$

where, δ represents the drift of the pneumatic and V represents the longitudinal velocity of the vehicle.

To study the vehicle's displacement in 3D, it is imperative to define a reference frame in which one will be able to consider the displacements in a straight line, but also the movements of yaw, pitch and roll. The selected references are:

R (w, x, y, z), the direct and fixed galilean reference frame which is related to the ground reference frame. R'(G, x', y', z'), the reference frame related to the mass center of the vehicle and obtained by the rotation of angle ψ around the axis (G, z'). R"(G, x", y", z"), the reference frame related to the mass center of the vehicle and obtained by the rotation of angle φ around the axis (G, y"). And R"'(G, x"', y"', z"'), the reference frame related to the mass center of the vehicle and obtained by the rotation of angle θ around the axis (G, x"').

The dynamic equations of the vehicle are established in the reference frame R"'. These equations are then expressed in the reference frame R, by using a matrix of passage P, linearized to the first order:

$$P = \begin{pmatrix} 1 & -\psi & \varphi \\ \psi & 1 & -\theta \\ -\varphi & \theta & 1 \end{pmatrix}$$

These equations are those of vehicle's body. We will give the wheel's equations.

Wheels: The model of tyres used is a Pacejka's model (Pacejka, 2006), which allows the representation of tyre behaviour by taking account of the longitudinal/transverse coupling. The angles which we will refer to are those which are necessary for Pacejka's model (Fig. 2) and used for simulation. (G, x_r, y_r, z_r) represent the reference frame of the wheel. The pneumatic drifts (δ_{pij}) are expressed in the following way:

$$\delta_{pij} = \arctan\left(\frac{\vec{V}_0 \cdot \vec{y}_r}{\vec{V}_0 \cdot \vec{x}_r}\right) - \alpha_{ij}$$

where, α_{ij} is the steering of the wheel's direction caused by the driver.

The vertical efforts below each wheel are expressed in the following way:

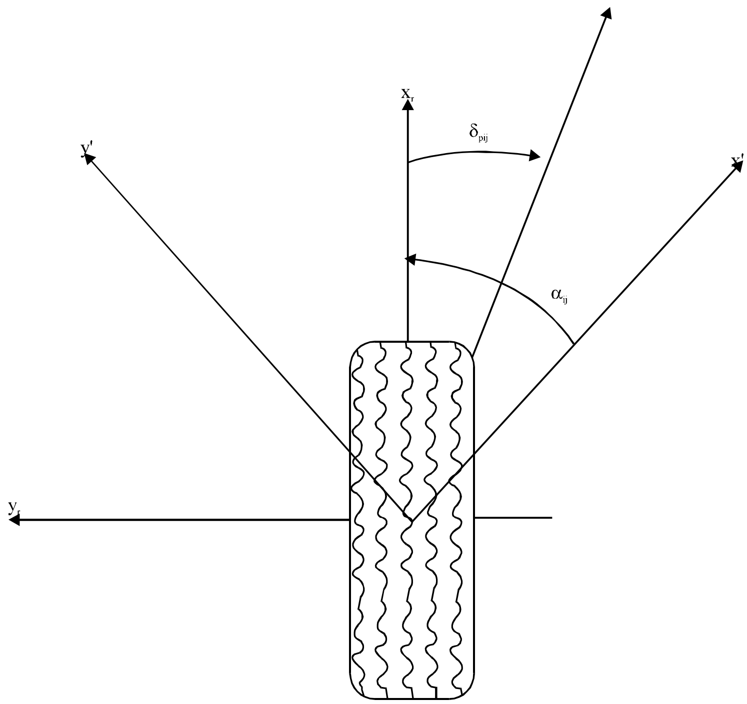

Fig. 2: The reper associated with the tire of study

$$F_{zij} = -k_{pij}(z_{ij} - z_{pij}) - C_{pij}(\dot{z}_{ij} - \dot{z}_{pij}) + F_{zsij}$$

where, F_{zsij} is the static effort due to the load below each wheel. The forces given in the reference frame of each wheel are those of Pacejka's model. The passage of the efforts in the wheel reference frame to the frame R' is expressed as follows:

$$F_{xrij} = F_{xij} \cos \alpha_{ij} - F_{yij} \sin \alpha_{ij}$$

$$F_{yrij} = F_{yij} \cos \alpha_{ij} - F_{xij} \sin \alpha_{ij}$$

The pneumatic slip is defined by the following relation:

$$\begin{cases} g = \dfrac{R\ w - V}{V} \times 100 \text{ if } R\ w < V \\ \\ \text{or} \\ \\ g = \dfrac{R\ w - V}{R\ w} \times 100 \text{ if } R\ w > V \end{cases}$$

All these equations enabled to develop software to simulate the road behaviour of a vehicle.

Description of the simulation software: The system is integrated in a simulation environment consisting of the complete model of the vehicle and is called VDS (Vehicle dynamics Software). This model was developed by the structures and systems dynamics team of the Ecole Centrale de Lyon during my thesis into 2001.

The synthesis parameters of the active suspensions were improved and integrated in the numerical model to the Institut National Polytechnique Houphouët Boigny Yamoussoukro into 2007. This model enables simulation of the dynamic behaviour of a vehicle in real time and thus allows experimentation on the frame using a control simulator, integrating the reactions of a real driver and the aerodynamics.

The model of tyre is also integrated in the tool for simulation and it takes account of the longitudinal/transverse coupling. The simulation tool was developed under MATLAB/Simulink.

Simulation: The purpose of the control of the active suspensions is to help the driver in extreme situations of driving which generally come to a loss of control of the vehicle and thus to an accident. The extreme situations concerned by a system of control of the trajectory for the light vehicles are above curve and underneath curve. And the system equipped with active suspension must maintain the vehicle on the trajectory which is imposed by the position of the steering angle and desired by the driver. The law of order synthesized in Ori *et al*. (2011) was integrated on a vehicle. Among the situations of simulated urgencies, we present the situation which consists in obtaining a circular trajectory like the round point ones and then we will appreciate the vehicle handling equipped with an active suspension compared to the one which is not.

Circular trajectory-rond-point: In this operation, the vehicle runs during 15 sec with 20 m sec^{-1} (approximately 72 km h^{-1}) in a turn on the left with a steering angle of 1.7 radian (approximately 97.4 degrees) then brakes one second after the perception of an obstacle. We observe the effects of control on the vehicle's trajectory during this operation.

RESULTS

Responding automatically to the driver's instruction, the vehicle is decelerated by pressing on the brakes to around -7.5 m sec^{-2}. This deceleration involves a loss of adherence. This loss of adherence degrades braking. That results in a loss of longitudinal acceleration (γ_x) and transverse acceleration (γ_y) at the end of 15 sec (Fig. 3a-b).

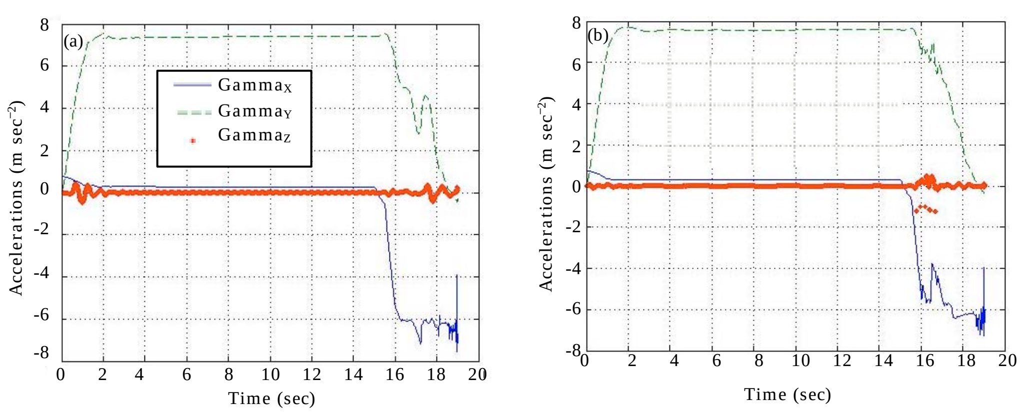

Fig. 3: Accelerations of the (a) non controlled vehicle and (b) controlled vehicle

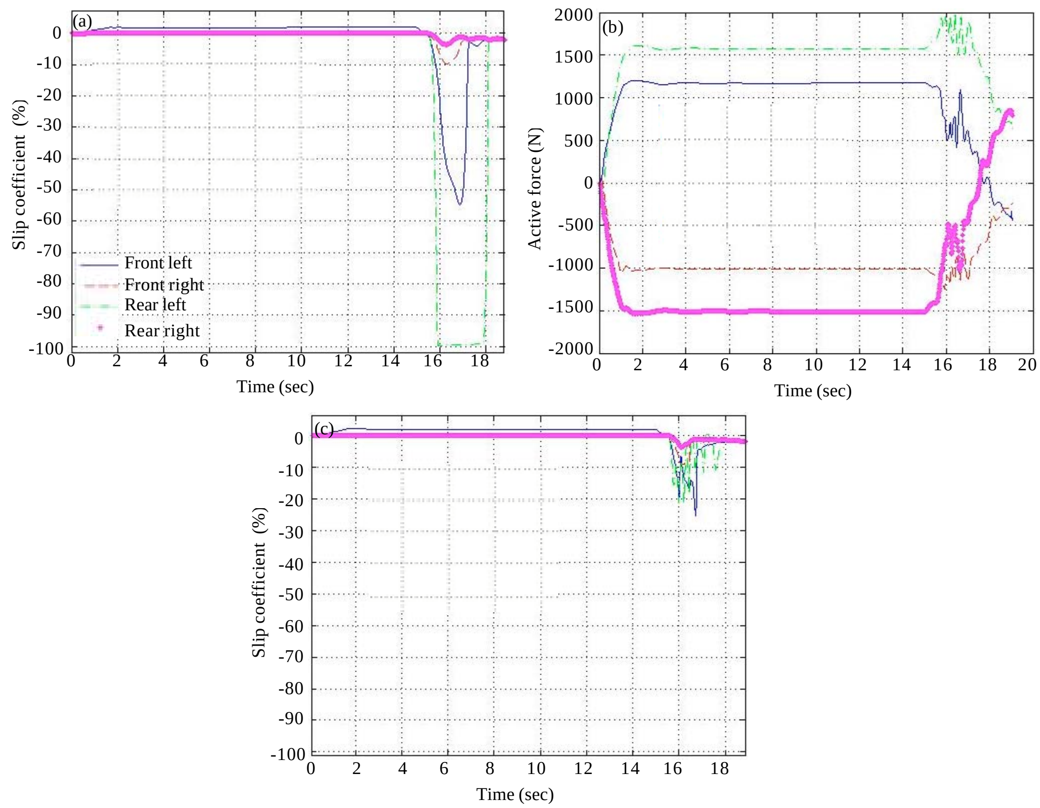

Fig. 4: Slip coefficient of (a) non controlled vehicle; (b) active force and (c) controlled vehicle

When the driver steers the wheel left at the beginning of the operation, a lateral load transfer occurs which tends to unload the left front and rear wheels in favour of the right front and rear wheels. And braking generates a transfer of longitudinal load from the rear wheels to the front ones after 15 sec of operation. The left rear wheel, unloaded from a large share of its load, is then on the verge of losing contact with the road. This wheel cannot therefore transmit any effort to the ground and it quickly locks during the braking phase. That results in a rate slip of -100% on the left rear wheel after deceleration (Fig. 4a).

The active force synthesized by the control law (Ori *et al*., 2011) and drawn in Fig. 4b, instantly balances the load of the suspended mass by adding an additional load when a wheel becomes unloaded and by releasing the wheel when it is saturated. Through its action, the active force allows the left front and rear wheels to return to a slip value (-15%) that makes easier the driver's maneuvering (Fig. 4c).

The displacements of controlled suspensions (Fig. 5a) are less significant than those of the non controlled suspensions (Fig. 5b). Which proves that control improves the comfort of the passengers.

In Fig. 6, the trajectories seen in the two previous cases are superposed. The non-controlled vehicle cannot respond to the driver's orders. It has a radius of curvature more significant than the driver's instruction, which can cause a skidding. Whereas the controlled vehicle perfectly follows the desired trajectory.

Figure 7 shows the total instant power consumed by the four wheels during the maneuver. It never exceeds 400 W, which is very reasonable. Energy consumption is a primary factor if the

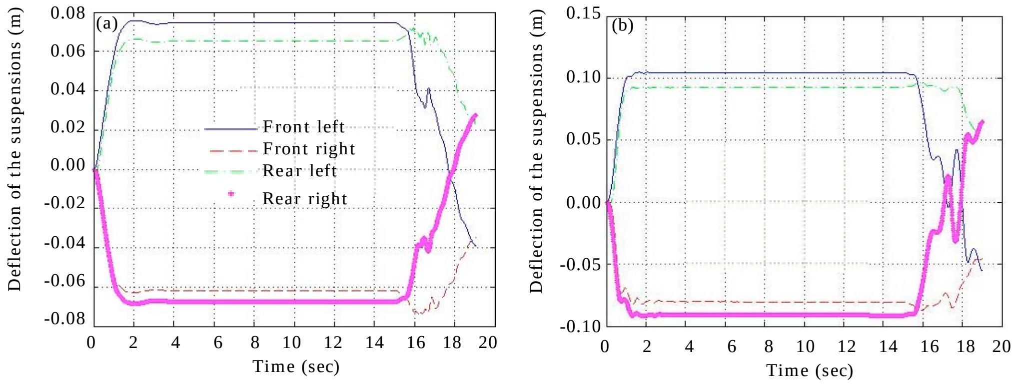

Fig. 5: Deflection of (a) controlled vehicle's suspension and (b) non controlled vehicle's suspension

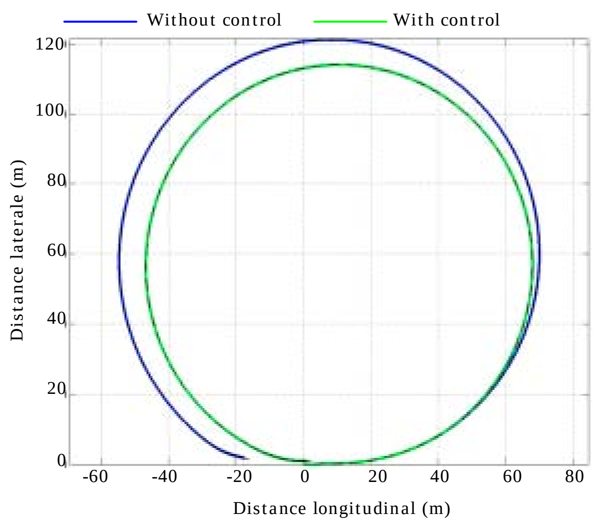

Fig. 6: Vehicle's trajectories

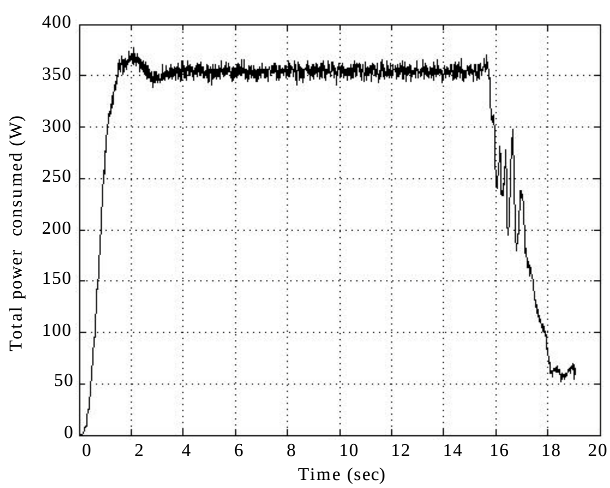

Fig. 7: Total power consumed

system has to be establish because current vehicles occasionally have high consumption levels. A system that would add an extra consumption of energy would have little chances of flourishing.

Many maneuver simulations have been conducted, notably while turning or driving in a straight line. The results of all these trials show that the controlled suspension system gives a better trajectory and a better respect of the angle order of the steering wheel, included on a low friction, which is a clear advantage.

DISCUSSION

Several theoretical, experimental or numerical works has already been proposed to improve vibratory comfort and also to ease the maneuverability of the vehicle. Among these works, one can quote those of Hiroki and Shingo (2008) which reduce the longitudinal vibrations while acting on the unsprung masses, those of Kreutz *et al.* (2009) improved the vehicle's handling while acting only on the rear wheels. The work suggested in this paper reduces the vibrations of the unsprung masses by action on the 4 wheels. This reduction of the vibrations of the unsprung masses improves the comfort of the passengers.

The Works of Azadi *et al.* (2010) used the Vehicle Dynamic Control system (VDC) to decrease the side movements during the critical maneuverability. System VDC contains several blocks like ABS (Anti-lock Braking System) for its brake. The system studied in this document does not include many blocks. It also reduces the braking distances and improves the vehicle's trajectories.

The work suggested by Baffet *et al.* (2008) gave an ideal turn according to the driver's instruction and bend using the observers of the type of Kalman's filters. The trajectories of the vehicle are perfect but the vibratory comfort of the case is not mentioned.

Experimental tests were carried out by Doumiati *et al.* (2009) to consider an ideal coupling between the transfer of load side and the normal forces for the handling and the safety of the passengers. This coupling is an excellent result for safety but it is controlled by different observers on a forecast of filtre/estimation. There is also the work of Cho *et al.* (2008) which describe a control of frame unified to improve the stability of the vehicle by using several blocks. The frame is controlled by the ESC (Electronic of stability Control) and maneuvrability by the AFS (Active Front Steering) which is a system of active direction.

The concepts which have more chances to succeed are the easiest to put in practice. The works of Doumiati *et al.* (2009) and Cho *et al.* (2008) used several blocks to control simultaneously to obtain a good stability and an excellent road behavior.

Rashid *et al.* (2006) gave a good comprehension of the characteristics to the damper of magnetic resonance. This damper is effective for isolating the case from the vibrations but doesn't mention its impact on the handling.

This study suggested in this document is based on a simple concept which consists in reducing the vertical acceleration of the unsprung mass. That is done while acting on the secondary suspensions and permits to reduce the braking distance (Ori *et al.*, 2011) then to improve the road behavior in turn without damaging the comfort of the passengers.

CONCLUSION

A new strategy of control of the dynamic behavior has been presented. The principle is to act on the suspensions with an aim of decreasing the curvature radius of the trajectory made possible by adherence.

The results of simulation show the real contribution of this strategy of control on the vehicle handling, on the sense of security tested by the driver, this without harming the comfort of the passengers. This device of suspensions can be tested then implemented on vehicles because it adapts to the extreme configurations of manoeuvrability.

REFERENCES

Andreasson, J. and T. Bunte, 2006. Global chassis control based on inverse vehicle dynamics models. Vehicle Syst. Dyn., 44: 321-328.

Azadi, S., M. Vaziri and M. Hoseini, 2010. Vehicle dynamic control of a passenger car applying flexible body model. Vehicle Syst. Dyn., 48: 587-617.

Baffet, G., A. Charara, D. Lechner and D. Thomas, 2008. Experimental evaluation of observers for tire-road forces, sideslip angle and wheel cornering stiffness. Vehicle Syst. Dyn., 46: 501-520.

Cho, W., J. Yoon, J. Kim, J. Hur and K. Yi, 2008. An investigation into unified chassis control scheme for optimised vehicle stability and manoeuvrability. Vehicle Syst. Dyn., 46: 87-105.

Chu, T.W. and R.P. Jones, 2008. Analysis and simulation of nonlinear handling characteristics of automotive vehicles with focus on lateral load transfer. Vehicle Syst. Dyn., 46: 17-31.

Demic, M., 1994. Optimization of vehicles elasto-damping elements characteristics from the aspect of ride comfort. Vehicle Syst. Dyn., 23: 351-377.

Doumiati, M., A. Victorino, A. Charara and D. Lechner, 2009. Lateral load transfer and normal forces estimation for vehicle safety: Experimental test. Vehicle Syst. Dyn., 47: 1511-1533.

Hiroki, K. and K. Shingo, 2008. Reduction of longitudinal vibration by side-view arrangement of suspension. Vehicle Syst. Dyn., 46: 161-173.

Kreutz, M., M. Horn and J. Zehetner, 2009. Improving vehicle dynamics by active rear wheel steering systems. Vehicle Syst. Dyn., 47: 1551-1564.

Ori, T.R., 2001. Suspensions actives et comportement dynamique des vehicules lors du freinage: Active suspensions and vehicles dynamic behaviour during braking phase. Ph.D. Thesis, Ecole Centrale de Lyon.

Ori, T.R., P. Gbaha, O. Asseu and A.L. Bot, 2011. Vehicle stopping distance by means of suspensions control. Asian J. Sci. Res., 4: 28-41.

Pacejka, H.B., 2006. Tyre and Vehicle Dynamics. 2nd Edn., Elsevier Science Publication, New York, ISBN-13: 980-0-7506-6918-4, pp: 156-215.

Rashid, M.M., M.A. Hussain and N.A. Rahim, 2006. Application of magneto-rheological damper for car suspension control. J. Applied Sci., 6: 933-938.

Shim, T. and C. Ghike, 2007. Understanding the limitations of different vehicle models for roll dynamics studies. Vehicle Syst. Dyn., 45: 191-216.

Venhovens, P.J.T.H. and K. Naab, 1999. Vehicle dynamics estimation using kalman filters. Vehicle Syst. Dyn., 32: 171-184.