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To cite this version:

Abbas Chokor, Reine Talj, Ali Charara, Moustapha Doumiati, Abdelhamid Rabhi. Rollover Prevention Using Active Suspension System. 20th IEEE International Conference on Intelligent Transportation Systems (ITSC 2017), Oct 2017, Yokohama, Japan. pp.1706-1711. hal-01679146

HAL Id: hal-01679146

https://hal.science/hal-01679146v1

Submitted on 9 Jan 2018

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Rollover Prevention Using Active Suspension System

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***Abstract*—This paper aims to turn the vehicle roll angle to a specified position by means of controlling the active suspensions system. The objective is to prevent vehicle rollover, and thus reassures driver's feeling on an on-road drive during hard cornering at high speeds. The proposed controller architecture is hierarchical, and contains two control levels. The upper controller is based on Lyapunov theory where a virtual control input representing an additive roll moment around the roll axis is calculated to converge the vehicle roll to its desired value. The low level controller transforms the generated roll moment to effective active suspension forces at each vehicle's corner. The results of a J-turn test and a Fish-hook test at 130 km/h initial speed show the accuracy of the proposed controller in preventing rollover without decelerating the vehicle.**

Keywords—Rollover prevention, driver self-confidence, active suspension control, Lyapunov control.

I. INTRODUCTION

Vehicle rollover is defined as 90° or more rotation of the vehicle around its longitudinal axis [1]. This phenomenon starts by two wheels' lift off (in the same vehicle side), hereby, rollover risk can be also evaluated around the axis joining the two other wheels stilled on the ground. In spite of the fact that rollovers constitute a small percentage of all accidents (3%), they commit fatal injuries, nearly 33% of all deaths from passenger vehicle crashes (National Highway Traffic Safety Administration NHTSA, 2011). For this reason, rollover has become an important safety issue for many researchers and automotive societies. According to several studies e.g. [1], there are two types of rollover: tripped and untripped rollovers. Tripped rollover occurs due to an external force on the vehicle, like wheels impact with a curb or a pot hole, an accident with another vehicle, or even a violent wind. Untripped rollover occurs due to an excess in the lateral acceleration or to the roll dynamics in a passive suspension integrated vehicle, e.g. going around a curved road with a sharp steering at high speed or undertaking a quick lane change. In spite of the fact that the majority of rollovers are tripped rollovers, many studies have been developed to prevent untripped rollover, which is back to its dependency on the measurable, estimable and predictable vehicle dynamics, unlike tripped rollover, which happens spontaneously, where studies are still not well investigated.

In literature, there exist several criteria to detect untripped rollover risk. These criteria depend essentially on the vehicle lateral acceleration, its induced roll motion (roll angle and roll rate) and its parameters. In [4], authors have defined a Rollover Index (RI) for rollover detection, authors have developed a differential braking based controller to control the vehicle yaw rate in order to diminish the lateral acceleration to its desired value calculated from a predefined RI threshold. In [5], in order to decrease the lateral acceleration, authors have proposed a simple braking based controller to diminish the vehicle speed. In [6], authors feed-back the vehicle roll angle and roll rate in a passive suspension integrated vehicle, to calculate the active steering command which prevents rollover according to the closed loop transfer function of the linearized vehicle model. In [2], authors have described rollover prevention in a high level controller by setting boundaries for the vehicle yaw rate. Whatever the used controller for rollover problem: simple braking, differential braking, or active steering. It has harming side effect on the vehicle trajectory (effect of yaw rate limitation and active steering): driving the vehicle outside the road, causing tires wear due to the frequent use of the braking system, and dropping the vehicle's speed.

The present work is an extension of the recently published paper [3]. The main originality of this work is the development of a new controller to turn (with a specified angle) the vehicle roll angle toward the inside wheels in a road corner by controlling the active forces in the active suspension systems integrated on the four vehicle corners. The specified roll angle position is calculated according to the lateral acceleration. It compensates the vehicle roll motion toward the outside, and adds the needed roll angle toward the inside. Consequently, it approaches the center of gravity toward the inner wheels which helps to avoid rollover. The specified roll angle acts as an incline support under the passengers making them more comfortable and gives the driver the intention of piloting the vehicle, instead of braking or releasing fuel pedal to manually adjust vehicle speed to diminish induced roll effect while cornering. The results show the accuracy of this technique in increasing lateral acceleration threshold that commits inside wheels lift-off, instead of diminishing the lateral acceleration without affecting vehicle speed and being very careful to the trajectory. For both objectives, the created roll angle plays the same role as the road cross bank angle, except that the effect of this last is on the whole vehicle while our concern is just on the suspended mass.

Section II.A exposes the untripped rollover phenomenon. In Section II.B, the vehicle model developed in the previous paper [3] is extended, to deal with the effect of changing the center of gravity position due to roll motion on the suspended vehicle weight. Section II.C develops a new controller with a specified

roll position reference to efficiently prevent rollover situations. Section III compares the new-featured proposed model to SCANeR Studio Simulator on a double lane change test at 100 km/h speed. Section IV validates by the J-turn manoeuver the proposed approach in preventing rollover, while guarantying: no changing in vehicle speed and minimum effect on the trajectory.

II. VEHICLE DYNAMICS AND CONTROL

A. Rollover problem formulation

For better understanding rollover problem, suppose in a first time that the vehicle is a rigid body (without suspension system), for the reason to show the effect of the lateral acceleration alone on the vertical load transfer. In a second time, the effect of the induced roll motion in a suspended vehicle (as a result of the lateral acceleration) in committing rollover is introduced. To carry out lateral acceleration effect, suppose that the vehicle is subjected to a quasi-static lateral acceleration which vary slowly comparing to the vehicle speed, this is done through a steady steering at high speed, e.g. highway turn (see Fig. 1). The lateral acceleration a_y applied on the vehicle center of gravity creates the d'Alembert's force (centrifugal force) $M.a_y$ which acts to turn over the vehicle in the outside direction of the turn with the torque $M.a_y.h$ acting on the axis formed by the outside wheels contact points with the ground. However, an opposite moment takes place to counterbalance this moment. It is performed by a new vertical couple of forces distanced by t_f forming a load transfer from inner to outer wheels:

$$F_{z_o} - F_{z_i} = M.a_y.h/t_f, \qquad (1)$$

where F_{z_o} and F_{z_i} are respectively the outer and inner vertical forces formed by adding each two tire vertical forces in the same vehicle side, $t_f = t_r$ are half vehicle track, g is the gravitational constant, h is the vehicle center of gravity height. Taking all force's moments maintaining equilibrium around the same axis (before starting rollover) gives:

$$M.a_y.h - \mathrm{M.g}.t_f + F_{z_i}.2.t_f = 0. \qquad (2)$$

If the vehicle speed V increases, or the radius of the turn R decreases (the curvature increases), the lateral acceleration increases (for any rigid body: $a_y = \mathrm{V}^2/\mathrm{R}$ in inertial frame). Increasing a_y happens while guarantying equilibrium in (2), which is done by a natural decreasing of the other single variable F_{z_i}, up to a certain amount of a_y where F_{z_i} becomes 0, which represents inner wheels lift off. Equation (2) becomes: $M.a_{y_lift_off}.h - M.g.t_f = 0$, hence:

$$a_{y_lift_off} = \frac{t_f}{h} g. \qquad (3)$$

t_f/h is called static stability factor (SSF). It is a constant value that depends on the vehicle geometry (1.2 for passengers cars, 1 for light trucks and 0.45 for heavy trucks [1]), that means heavy trucks rollover occurs at lower lateral accelerations.

In real situations, due to the suspension system, the vehicle is not yet a simple rigid body. For studying roll effect on rollover, suppose that the vehicle has one degree of freedom represented by the roll angle θ between the suspended and unsuspended mass (see Fig. 2), which deviates the suspended mass center of gravity by a positive angle θ (toward the outside) around the roll

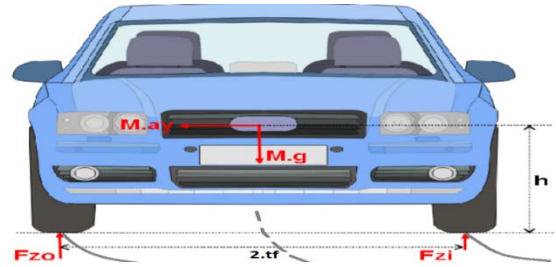

Fig. 1 Lateral acceleration on a road corner

axis. Hereby, the moment off all forces around the axis joining outer wheels becomes:

$$M_s.a_y.h - M_s.\mathrm{g}.(t_f - (h - hr)\sin\theta) + F_{z_i}.2.t_f = 0. \qquad (4)$$

By analogy with previous analysis, under the assumption of small angles $\sin\theta \approx \theta$, $a_{y_lift_off}$ becomes:

$$a_{y_lift_off} = \frac{t_f - (h - hr).\theta}{h} g, \qquad (5)$$

that means $a_{y_lift_off}$ is diminished when considering passive suspension system by approximately 5% [1], hereby, comparing to a rigid body vehicle, the vehicle occupied with a passive suspension system starts rollover at lower lateral accelerations. The transient response of θ is ignored in this analysis, while roll overshoot may occurs due to the roll dynamic developed in the next Section. This overshoot add to θ a small amount, which consequently, diminish $a_{y_lift_off}$ another 5%, and may cause vehicle rollover when moving at a lateral acceleration near to $a_{y_lift_off}$. This analysis also ignores the lateral acceleration error measured by the accelerometer subjected to a roll motion and other physical elements uncertainties causing rollover like center of gravity vertical shift, changes in tires and ground contact surfaces. In order to stay in a safe driving region, a safety factor of 0.7 of the total $a_{y_lift_off}$ expressed in (5) is proposed, this new value is denoted a_{y_safe}:

$$a_{y_safe} = 0.7 a_{y_lift_off} = 0.7 \frac{t_f - (h - hr).\theta}{h} g. \qquad (6)$$

B. Vehicle model

The full vehicle model adopted to validate the controller has been already developed in the previous paper [3]. In this Section, the model is enhanced to be more accurate with the complexity of roll motion effect on the vehicle. Hereby, an extension considering all the changes that have been done is presented, some expressions may be repeated to make the combination between both models as simple as possible (for more explanation see [3]).

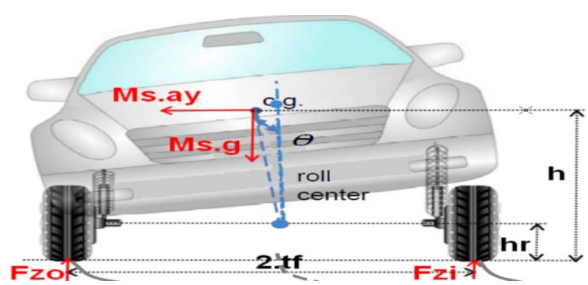

Fig. 2 Vehicle roll angle due to the lateral acceleration

Due to the lateral acceleration a_y, the vehicle suspended mass M_s is solicited to a roll motion θ, the roll center of roll rotation is the vertical projection of the suspended mass center of gravity ($h_\theta = h - h_r$) on the plane containing the four wheels centers (Fig. 2). The roll dynamics calculated by forces moments becomes:

$$\ddot{\theta} = \frac{1}{I_x + M_s h_\theta^2}\begin{pmatrix} \left(-F_{s_{fr}} + F_{s_{fl}}\right)t_f + \left(-F_{s_{rr}} + F_{s_{rl}}\right)t_r \\ +M_s(h_\theta \cos\theta + z).a_y + M_s(h_\theta \sin\theta + z).g \end{pmatrix}, \quad (7)$$

where $F_{s_{ij}}$ ($i = \{f, r\}$ for front and rear , and $j = \{r, l\}$ for right and left) are the four vehicle corners suspended forces, z is the suspended mass center of gravity vertical displacement, I_x is the moment of inertia of the suspended mass around its longitudinal axis (Fig. 3). The suspended forces are calculated from the quarter vehicle model [3]:

$$F_{s_{ij}} = -k_{s_{ij}}\left(z_{s_{ij}} - z_{us_{ij}}\right) - c_{s_{ij}}\left(\dot{z}_{s_{ij}} - \dot{z}_{us_{ij}}\right) + U_{ij}, \quad (8)$$

where $z_{s_{ij}}$ and $z_{us_{ij}}$ are the suspended and unsuspended mass vertical displacement, U_{ij} are the active suspension forces, $k_{s_{ij}}$ and $c_{s_{ij}}$ are the suspension coefficients at the wheel *ij*.

Suppose $F_{dz_{ij}}$ the vertical force developed by the suspension dynamic on the wheel ij, each $F_{dz_{ij}}$ is calculated according to the quarter-vehicle vertical model in [3]:

$$F_{dz_{ij}} = -k_{t_{ij}}\left(z_{us_{ij}} - z_{r_{ij}}\right) - c_{t_{ij}}\left(\dot{z}_{us_{ij}} - \dot{z}_{r_{ij}}\right). \quad (9)$$

Suppose $F_{Z_LT_{ij}}$ the vertical force developed on the tire *ij* by the load transfer resulting from the lateral acceleration, (the longitudinal acceleration effect is neglected) and m_{ij} is the quarter vehicle mass:

$$F_{Z_LT_{ij}} = \pm m_{ij}.\left(\frac{h}{tf}\right).ay. \quad (10)$$

The total force $F_{z_{ij}}$ on the wheel ij is:

$$F_{z_{ij}} = F_{dz_{ij}} + F_{Z_LT_{ij}} + m_{ij}.g. \quad (11)$$

C. Active suspension controller-new objectives

Rigidifying the suspensions in a semi-active suspended vehicle may reduce the passive vehicle roll rotation toward the outside of the corner, so it decreases θ to nearly zero. This procedure arises $a_{y_lift_off}$ in (5) again to its static stability value, consequently, arising a_{y_safe}. The contribution that adds the active suspension system is the ability to continue turning roll angle in the negative direction (to the inner side of the corner) which accomplish double objectives:

The first one is arising $a_{y_lift_off}$ in (5) more than its static stability value, that means a lateral acceleration near the static stability value will no more cause outer wheels lift off. Consequently, arising a_{y_safe} which means that the critical region (unsafe region) is shifted to a higher value of lateral acceleration and the vehicle is returned to a safe region.

Theoretically, shifting a_{y_safe} is done while maintaining the lateral acceleration at its value, that is to say there is no action in the goal of diminishing lateral acceleration to avoid rollover

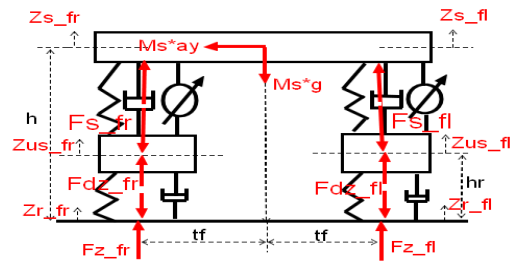

Fig. 3 Vertical vehicle free body diagram

as in braking or steering rollover avoidance controllers, where they influence vehicle speed and trajectory.

The second one is giving more comfort and sense of piloting to the driver when cornering. This is done by choosing a precise roll angle toward the inside of the corner. This last is supposed to be linear with the lateral acceleration, that means the mapping between the annoying lateral acceleration and the desired roll angle is constant in order to avoid violent and irregular rate changing. Thus, at zero lateral acceleration (straight road) the desired roll position is zero, and at the static safe lateral acceleration threshold 0.7SSF.g is equal to 10° (the maximum it can be due to the vehicle design constraints), then:

$$\theta_{des} = -\frac{10.\frac{\pi}{180}}{0.7\mathrm{SSF.g}}a_y. \quad (12)$$

The four $F_{s_{ij}}$ which appear explicitly in the roll acceleration (7) contain the four control inputs U_{ij} as described in (8). Let:

$$\mathcal{M}_\theta = \left(-U_{fr} + U_{fl}\right)t_f + \left(-U_{rr} + U_{rl}\right)t_r, \quad (13)$$

the active forces' moment considered as an intermediate control input taking into account all inputs of (7), so (7) becomes:

$$\ddot{\theta} = \frac{1}{I_x + M_s h_\theta^2}\begin{pmatrix} \mathcal{M}_\theta + \left(-F_{fr} + F_{fl}\right)t_f + \left(-F_{rr} + F_{rl}\right)t_r \\ +M_s(h_\theta \cos\theta + z).a_y + M_s(h_\theta \sin\theta + z).g \end{pmatrix}, \quad (14)$$

where F_{ij} are $F_{s_{ij}}$ without U_{ij}:

$$F_{ij} = -k_{s_{ij}}\left(z_{s_{ij}} - z_{us_{ij}}\right) - c_{s_{ij}}\left(\dot{z}_{s_{ij}} - \dot{z}_{us_{ij}}\right) \quad (15)$$

The dynamic of θ in (14) has a relative degree of two w.r.t the virtual control input $\mathcal{M}_\theta$, then let:

$$e_\theta = \theta - \theta_{des}, \quad (16)$$

$$s_\theta = \dot{e}_\theta + k_{1\theta}e_\theta + k_{2\theta}\int_0^t e_\theta d\tau, \quad (17)$$

$$V_\theta = \frac{1}{2}{s_\theta}^2, \quad (18)$$

$$\dot{s}_\theta = -\alpha_\theta s_\theta, \quad (19)$$

where: e_θ is the error between the actual roll angle and its desired value, $\dot{e}_\theta$ is e_θ derivative, s_θ is a function representing roll states, V_θ is a positive definite Lyapunov candidate function, α_θ is positive constant verifying $\dot{V}_\theta = s_\theta \dot{s}_\theta < 0$, $k_{1\theta}$, $k_{2\theta}$ and α_θ are positive constants representing controller gains. Appling now this approach: $\dot{s}_\theta = \ddot{e}_\theta + k_{1\theta}\dot{e}_\theta + k_{2\theta}e_\theta = -\alpha_\theta\left(\dot{e}_\theta + k_{1\theta}e_\theta + k_{2\theta}\int_0^t e_\theta d\tau\right)$, then:

$$\ddot{e}_\theta = -(\alpha_\theta + k_{1\theta})\dot{e}_\theta - (\alpha_\theta k_{1\theta} + k_{2\theta})e_\theta - \alpha_\theta k_{2\theta}\int_0^t e_\theta d\tau = \ddot{\theta} - \ddot{\theta}_{des}.$$

Substituting $\ddot{\theta}$ by (14), the high level control input is derived:

$$\begin{aligned}\mathcal{M}_\theta = &-(\alpha_\theta + k_{1\theta})(I_x + M_s h_\theta{}^2)(\dot{\theta} - \dot{\theta}_{des}) \\ &-(\alpha_\theta k_{1\theta} + k_{2\theta})(I_x + M_s h_\theta{}^2)(\theta - \theta_{des}) \\ &-\alpha_\theta k_{2\theta}(I_x + M_s h_\theta{}^2)\int_0^t (\theta - \theta_{des})d\tau \\ &-(-F_{fr} + F_{fl})t_f - (-F_{rr} + F_{rl})t_r \\ &-M_s(h_\theta \cos\theta + z)a_y - M_s(h_\theta \sin\theta + z).g \\ &+(I_x + M_s h_\theta{}^2)\ddot{\theta}_{des}.\end{aligned} \quad (20)$$

Finding this control input needs feedback from the Inertial Measurement Unit IMU for θ and $\dot{\theta}$, an accelerometer for a_y, a position sensor for z (can be neglected), knowledge on vehicle parameters, and instead of using an expensive force detection sensor, F_{ij} are estimated from (15) where its inside variables are already estimated in [3] using roll, pitch and vertical displacement sensors.

In order to create the desired $\mathcal{M}_\theta$ around the roll axis, the control inputs U_{ij} are distributed as follow (Fig. 4):

$$U_{fl} = 0.5\frac{\mathcal{M}_\theta}{t_f}.\frac{l_r}{l_f + l_r}, \quad (21)$$

$$U_{fr} = -0.5\frac{\mathcal{M}_\theta}{t_f}.\frac{l_r}{l_f + l_r}, \quad (22)$$

$$U_{rl} = 0.5\frac{\mathcal{M}_\theta}{t_r}.\frac{l_f}{l_f + l_r}, \quad (23)$$

$$U_{rr} = -0.5\frac{\mathcal{M}_\theta}{t_r}.\frac{l_f}{l_f + l_r}. \quad (24)$$

This choice also avoids any interference with vertical and pitch motions, where the summation of all forces and moment around the pitch axis are zeros.

The actuator full model is not the interest of this paper, but in order to have feasible control inputs, it is simplified to a first order filter 1/(1+T.s) which has a *T=0.1* second as response time and a maximum force capacity of 9800 N [11].

III. Vehicle Model Validation

In this Section, the proposed model is validated, on a double lane change test using the professional simulator 'SCANeR Studio' dedicated to vehicle dynamics simulations. The double lane change test is chosen to solicit the lateral and roll vehicle motions, that is to have validated states (a_y and θ) for well quantification of rollover risk. The vehicle parameters used for this test are already defined in [3]. The vehicle speed is maintained 100 km/h (Fig. 7) by using SCANeR Studio longitudinal controller to avoid speed drop. Thus, the total wheels torques (Fig. 5) generated by the SCANeR are equally feedforwarded to the front wheels. The same is done for front wheels steering angles shown in Fig. 6. The steady 2° difference between the left and the right becomes from vehicle geometry constrains. The lateral acceleration (Fig. 8), roll angle (Fig. 9) and vertical load transfer (Fig. 10) comparison with SCANeR show the accuracy in modelling these variable states which are the most interesting in rollover phenomenon.

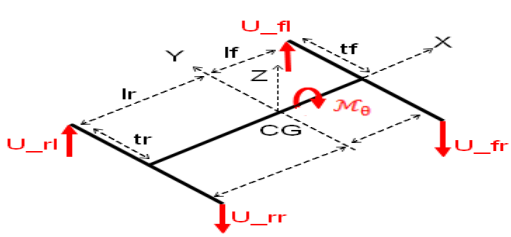

Fig. 4 Active forces distribution

IV. Controller Validation

The proposed roll controller by active suspensions has been validated in Matlab/Simulink, on a J-Turn test. The J-Turn test represents a turning manoeuvre with an aggressive steering, the test takes four seconds, then the driver has two seconds to take back the control as shown in Fig. 11.

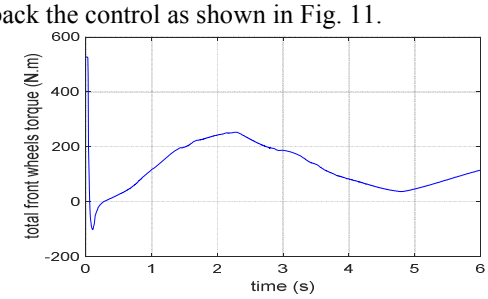

Fig. 5 Wheels Torques

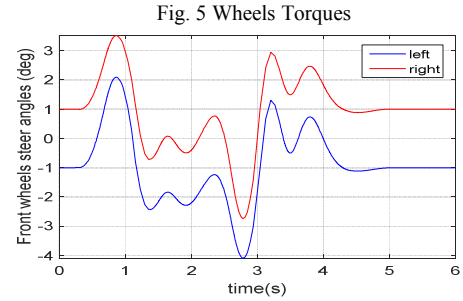

Fig. 6 Front wheels steering angles

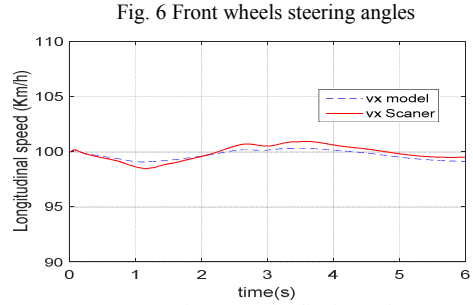

Fig. 7 Longitudinal speed

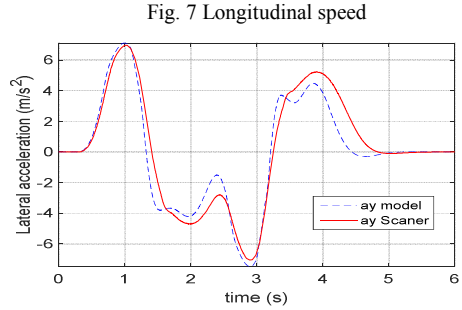

Fig. 8 Lateral acceleration

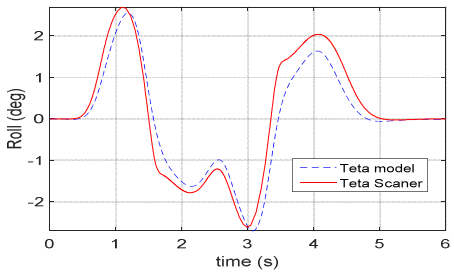

Fig. 9 Roll angle

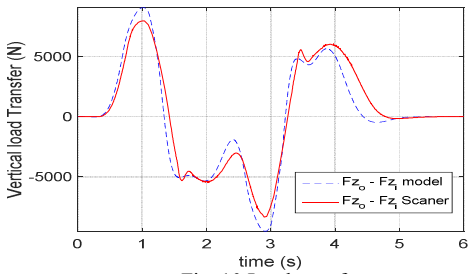

Fig. 10 Load transfer

Due to frictions and the aggressive steering, vehicle speed drops during the test. In order to remain it high, the vehicle initial speed is chosen to be 130 km/h as shown in Fig. 12. The same figure shows exactly the same results for both simulations (with and without roll control) making the roll control by active suspension system has no effect on vehicle speed. Fig. 13 shows the uncontrolled, controlled and desired roll angles, where the controlled one is on the opposite direction w.r.t the uncontrolled one (as in passive suspension) and follows the desired one calculated according to the lateral acceleration in (12). Due to the passive roll angle, Fig. 14 shows that a_{y_safe} decreases, making the vehicle subjected to rollover risk. Indeed the vehicle lateral acceleration becomes in the unsafe region (above a_{y_safe}). Turning the roll angle in the opposite direction solves the problem by remaking a_{y_safe} above the lateral acceleration. The small difference between both lateral acceleration comes from the change in vertical forces applied to wheels, which have a little effect on lateral tires forces. This change in vertical forces comes from the suspension active forces (control inputs) in Fig. 15 which act in both direction: on the suspended mass to change roll angle, and on the wheels. The created roll angle, which approach the suspended mass center of gravity to inner wheels, contribute in limiting vertical load transfer to outer wheels which has been validated in Fig. 16.

Fig. 17 shows the trajectories performed and the distance error between both situations. After six seconds, the test is supposed to be ended and the rollover risk is avoided. The controller effect on the trajectory is 1 meter which is acceptable due to the fact that the vehicle remains in the same lane. The steering input is the same for both controlled and uncontrolled tests. While in fact the driver is in the vehicle, and he is supposed to re-adapt his steering input to follow the trajectory. This is not considered in the simulations because we are not using a driver model.

The second test is the Fish-hook test (steering angle in Fig. 18) at the same initial speed 130 km/h. The main idea behind this test is to have a validated approach when the induced roll angle (without control) is in the opposite direction as shown in Fig. 19, due to a steering to the left then to the right.

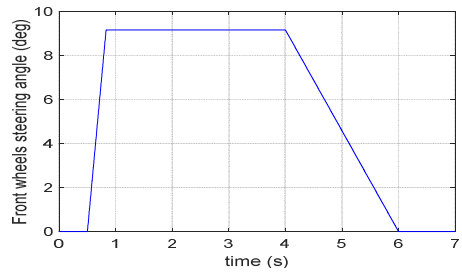

Fig. 11 Front wheels steering angle

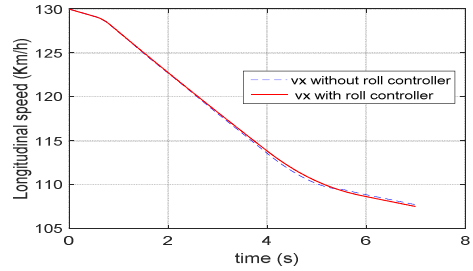

Fig. 12 Longitudinal speed

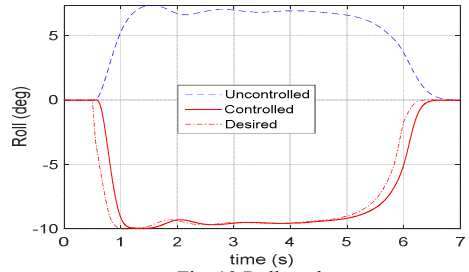

Fig. 13 Roll angle

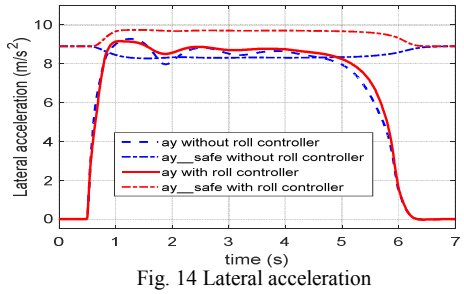

Fig. 14 Lateral acceleration

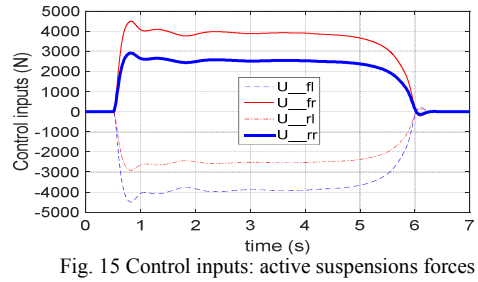

Fig. 15 Control inputs: active suspensions forces

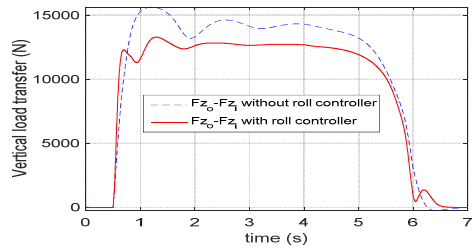

Fig. 16 Load transfer

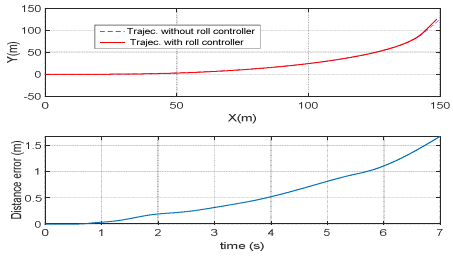

Fig. 17 Trajectory and error distance

When turning to the right (steer angle negative), the vehicle lateral acceleration becomes negative (in the opposite direction of lateral motion axis) as shown in Fig. 20, then the SSF in (3) becomes $-t_f/h$. Thus, a_{y_safe} becomes:

$$a_{y_safe} = 0.7a_{y_lift_off} = 0.7\frac{t_f-(h-hr).\theta}{h}g, \quad \text{if } a_y \geq 0 \qquad (25)$$

$$a_{y_safe} = 0.7a_{y_lift_off} = 0.7\frac{-t_f-(h-hr).\theta}{h}g. \quad \text{if } a_y < 0 \qquad (26)$$

The results show that the controlled roll angle follows its desired value and rollover is avoided in both directions.

V. Conclusion

In this paper, a roll controller using active suspensions is proposed, with a specific proposed desired roll angle. The results of the proposed roll controller show its accuracy in achieving its both objectives: driver's comfort and rollover avoidance. This approach limits the use of the other actuators which have side effects on vehicle speed and trajectory, in the wide range of vehicle speed from approximately 105 km/h to 130 km/h (Fig. 12), causing wheel lift off for the chosen steering. When moving in the extreme dangerous situation at very high lateral acceleration, we have to deal with the limitation of the proposed controller, due to the limited load transfer regulation by approaching the center of gravity toward the inside, thus, a limited a_{y_safe} raising, where other actuators should be used to diminish the lateral acceleration.

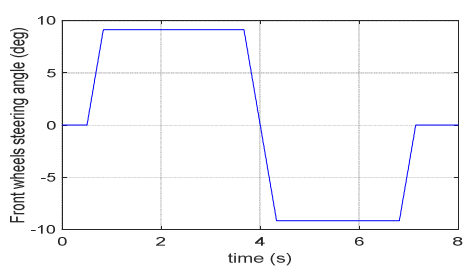

Fig. 18 Front wheels steering angle

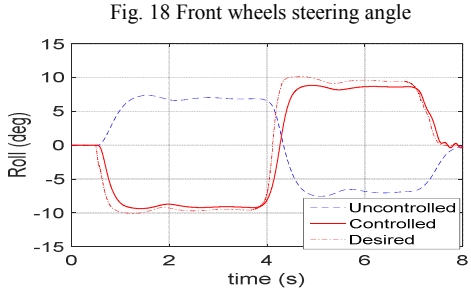

Fig. 19 Roll angle

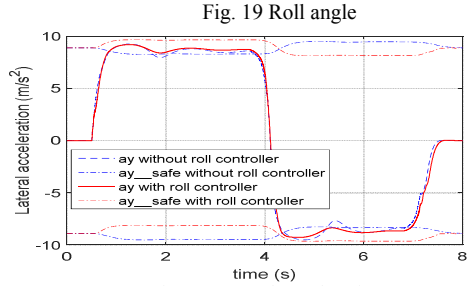

Fig. 20 Lateral acceleration

VI. Acknowledgment

The authors would like to thank the Hauts-de-France Region and the European Regional Development Fund (ERDF) 2014/2020 for the funding of this work. This work was also carried out in the framework of the Labex MS2T, (Reference ANR-11-IDEX-0004-02) and the Equipex ROBOTEX (Reference ANR-10-EQPX-44-01) which were funded by the French Government, through the program " Investments for the future" managed by the National Agency for Research.

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