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A rollover indicator based on the prediction of the load transfer in presence of sliding: application to an All Terrain Vehicle

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Abstract— **The lateral rollover of quad bikes represents a significant part of severe accidents in the field of agricultural work. The specifities of such vehicles (small wheelbase, track and weight, as well as high speed), together with the terrain configuration (off-road environment) prevent from describing rollover occurence as it is proposed for car-like vehicles. In particular, sliding effects significantly affects the evaluation of the rollover risk. This paper proposes a rollover risk indicator dedicated to off-road vehicles, taking into account the environment properties and more particularly the grip condition and its variation. It is based on the prediction of the lateral load transfer relying on vehicles models including sliding effects. This indicator can be run on-line when the vehicle is moving. It allows to anticipate a potential danger, and could then be used to design security systems. Performances of this indicator are demonstrated using the multibody dynamic simulation software Adams.**

I. INTRODUCTION

The market of light all terrain vehicles (ATVs) and especially quad bikes extends very quickly. These vehicles, initially designed for performing farm tasks, are now largely used as leisure activities. Unfortunately, the number of accidents is increasing with this market extension. For instance, 50 severe accidents per year have been reported in France only in the field of agricultural work (see [4]), while in the U.S.A, the CPSC (see [6]) has made a list of 470 ATV deaths in 2004. Most of these accidents are related to lateral rollovers.

Such critical situations generally occur when the pilot makes the vehicle turn too fast, especially on sloping surfaces or high grip irregular grounds. According to [18], quad bikes specificities (small weight relatively to the pilot, small wheelbase and track), together with high reachable speed emphasize the rollover risk with respect to on-road vehicles. These features provide a huge manageability to quad bikes, but these vehicles are then particularly subjected to rollover. Numerous security systems dedicated to rollover prevention have been developed for road vehicles. The most common systems are mechanical ones, such as anti-roll bars (see [9]) and electronical ones such as active suspensions, active anti-roll bars, steering and braking control (see for instance [1] and [17]).

However, active devices developed for car industry generally rely on vehicles dynamic models. Most of them do not integrate a tire model ([8] and [10]) or only use a linear tire model which can only account for tire pseudo-sliding area (such as in [16]). But contrary to urban vehicles, sliding effects are very significant in ATVs applications and can vary in real time (see III-D). A steady linear model then appears to be unsuitable for ATV rollover detection. Consequently, new approaches have to be designed in order to develop dynamic models from which stability devices for off-road vehicles could be developed.

This paper proposes a rollover indicator valid in presence of sliding to be used in automatic control devices for off-road vehicles stability. So as to develop this indicator, the following approach has been used: a first semi-analytical model based on vehicle roll and yaw frames is defined without accounting for sliding effects. In the sequel, this model is used to identify some parameters values. Next, a second semi-analytical model is introduced and takes into account sliding effects thanks to a wheel/ground contact modeling. Then, an algorithm is developed to calculate the lateral load transfer in presence of sliding. These two models are simple enough to be computed in real time. Finally, the rollover indicator relies on the prediction of the future lateral load transfer and allows to anticipate hazardous situations. This approach has been tested on a multibody model built with Adams software (broadly used in the car industry) and demonstrates the capabilities of such an indicator in predicting off-road vehicles rollover.

II. DYNAMIC MODELING IN ABSENCE OF SLIDING

The modeling objective is to describe the vehicle dynamics in order to compute the lateral load transfer. In this paper, the vehicle velocity, the steering angle and the ground inclination are the three inputs assumed to be available. The lateral load transfer constitutes the model output defined by the following expression:

$$LLT = \left(\frac{F_{n2} - F_{n1}}{F_{n2} + F_{n1}}\right) \tag{1}$$

where F_{n1} and F_{n2} are the normal forces applied on the left and right sides of the vehicle. A unitary load transfer value corresponds to the largest possible load transfer: if $|LLT|$ is equal to 1, then two wheels of the same side have lifted off and the vehicle starts to rollover. According to [7], if the lateral load transfer reaches the range [0.8, 0.9] then the quad bike is close to rollover. Since the expected indicator is intented to be used into stabilizing control laws, the lowest value 0.8 is here considered as the rollover critical threshold.

In order to estimate the lateral load transfer when pure rolling without sliding contact conditions are satisfied, two 2D models are now introduced.

A. Notations and modeling without sliding effects

In order to extract the normal forces applied on the vehicle, a simplified representation of the vehicle in its roll frame has been used as in [16] and it is depicted in Fig.1.

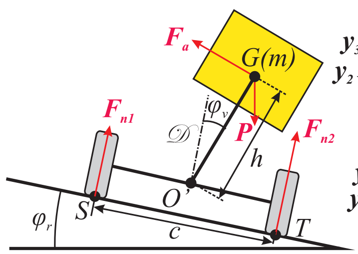


Fig. 1. Vehicle roll model and parameters.

The parameters used in the roll model without sliding are:

- O' is the vehicle roll center. The roll center location is assumed to be constant. This is realistic as long as the load transfer is inferior to 1,
- G is the center of gravity of the vehicle suspended mass m described as a parallelepiped,
- $P = mg$ is the gravity force of the suspended mass with g denoting the gravity acceleration,
- h is the distance between O' and G,
- c is the vehicle track,
- φ_r is the ground inclination,
- φ_v is the roll angle of the suspended mass,
- F_{n1} is the normal force on the vehicle left side,
- F_{n2} is the normal force on the vehicle right side,
- F_a is a restoring-force associated with the roll movement. This force is considered here to be parametrized by two parameters, k_r the stiffness coefficient and b_r the damping coefficient. The expression of this force is related to the roll movement by equation (2):

$$\overrightarrow{F_a} = \frac{1}{h}\left(k_r\varphi_v + b_r\dot{\varphi}_v\right)\overrightarrow{y_3} \tag{2}$$

However, the calculation of the normal forces F_{n1} and F_{n2} and therefore of LLT, requires the knowledge of some motion variables. They can be obtained from a second simplified representation of the vehicle in the yaw frame, known as Ackermann model (see [3]), depicted in Fig. 2.
The parameters used in this yaw model are:

- O is the instantaneous center of rotation,
- a is the front half-wheelbase,
- b is the rear half-wheelbase,
- L is the vehicle wheelbase,
- R is the curvature radius,
- δ is the steering angle,
- v is the vehicle linear velocity at the center of the rear axle.

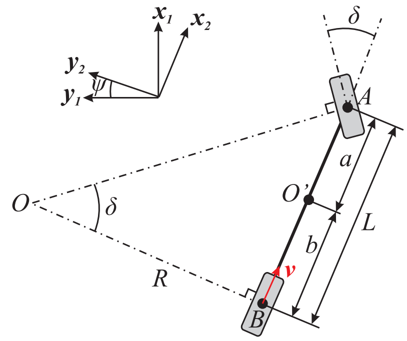


Fig. 2. Vehicle yaw model and parameters.

Dynamic modeling is then carried out relying on the two following hypotheses:

- As rolling without sliding contact conditions are assumed, the yaw rate $\dot{\psi}$ can be computed relying on the instantaneous center of rotation:

$$\dot{\psi} = \frac{v\cdot\tan(\delta)}{L} \tag{3}$$

- The suspended mass is assumed to be symmetrical with respect to the two planes $(z_3,\ y_3)$ and $(x_3,\ z_3)$. The inertial matrix is then assumed to be diagonal:

$$I_{G/R_3} = \begin{bmatrix} I_x & 0 & 0 \\ 0 & I_y & 0 \\ 0 & 0 & I_z \end{bmatrix} \tag{4}$$

B. Equations of the load transfer without sliding

The load transfer derives from the fundamental principle of the dynamic, which ensures that:

$$\begin{cases} m\overrightarrow{a_G}\cdot\overrightarrow{y_2} = \left(\overrightarrow{P}+\overrightarrow{F_a}\right)\cdot\overrightarrow{y_2} \\ m\overrightarrow{a_G}\cdot\overrightarrow{z_2} = \left(\overrightarrow{P}+\overrightarrow{F_a}+\overrightarrow{F_{n1}}+\overrightarrow{F_{n2}}\right)\cdot\overrightarrow{z_2} \\ \overrightarrow{\Delta_{G/R_3}}\cdot\overrightarrow{x_2} = \left(\overrightarrow{M_{G,F_{n1}}}+\overrightarrow{M_{G,F_{n2}}}\right)\cdot\overrightarrow{x_2} \end{cases} \tag{5}$$

where $\overrightarrow{a_G}$ is the acceleration at the center of gravity, $\overrightarrow{\Delta_{G/R_3}}$ is the dynamic momentum at the center of gravity expressed in R_3 (R_3 is $(x_3,\ y_3,\ z_3)$ frame shown on Fig.1). $\overrightarrow{M_{G,F_{n1}}}$, $\overrightarrow{M_{G,F_{n2}}}$ are the different momenta due to normal forces $\overrightarrow{F_{n1}}$ and $\overrightarrow{F_{n2}}$ at the center of gravity. From equations (5), variations of φ_v, F_{n1} and F_{n2} can be calculated. General equations, which take into account the ground inclination and its variations are provided in [2]. On a flat ground ($\varphi_r = 0$), these equations can be simplified as:

- $$\ddot{\varphi}_v = \frac{1}{h\cos(\varphi_v)}\left[h\dot{\varphi}_v^{\,2}\sin(\varphi_v) + h\dot{\psi}^2\sin(\varphi_v) + v\dot{\psi} + b\ddot{\psi} - \left(\frac{k_r\varphi_v + b_r\dot{\varphi}_v}{mh}\right)\cos(\varphi_v)\right] \tag{6}$$

- $$F_{n1}+F_{n2} = m\left[-h\ddot{\varphi}_v\sin(\varphi_v) - h\dot{\varphi}_v^{\,2}\cos(\varphi_v) + g - \left(\frac{k_r\varphi_v + b_r\dot{\varphi}_v}{mh}\right)\sin(\varphi_v)\right] \tag{7}$$

- $$F_{n1}-F_{n2} = \frac{2}{c}\left[I_x\ddot{\varphi}_v + (I_z - I_y)\left[\dot{\psi}^2\cos(\varphi_v)sin(\varphi_v)\right] - h\sin(\varphi_v)(F_{n1}+F_{n2})\right] \tag{8}$$

The normal forces F_{n1} and F_{n2}, and therefore the load transfer, can be deduced from equations (7) and (8). Roll angle φ_v and yaw rate $\dot{\psi}$ appearing in these expressions can be obtained from equations (3) and (6). Therefore equations (3), (6), (7) and (8) constitute the semi-analytical model without sliding effects, hereafter noted NSM (Non-Sliding Model). As the damping coefficient b_r only acts on transient load transfer behavior and does not affect the steady-state value, its estimation can be avoided. As a result, only the two parameters h and k_r have to be estimated. This is achieved off-line using a Newton-Raphson method (as detailed in [2]). The other parameters (mass, inertial products, etc) can be measured.

III. Dynamic model with sliding effects

The NSM model is suitable for calculating the load transfer on high grip grounds, but gives a bad estimation of the load transfer when sliding occurs. Indeed, the grip conditions deeply impact the load transfer, especially in the non-linear area of tire ground contact. Since quad bikes are precisely supposed to move on irregular slippery ground, another semi-analytical model accounting for non-linear tire ground contact is proposed hereafter in order to support vehicle rollover detection.

A. Notations and modeling with sliding effects

Sliding effects are reflected on the vehicle yaw rate because it alters the location of the instantaneous center of rotation. In order to describe such phenomena, a tire model has to be incorporated into the vehicle yaw model. The proposed model relies on side slip angles (such as in [13]) depicted on Fig. 3. More precisely, the new parameters introduced into the yaw model are:

- β is the global slip angle of the vehicle,
- α_r is the rear slip angle of the vehicle,
- α_f is the front slip angle of the vehicle,
- C is the tire stiffness,
- u is the vehicle velocity at the roll center,
- F_{t1} is the lateral force generated on the front tire,
- F_{t2} is the lateral force generated on the rear tire.

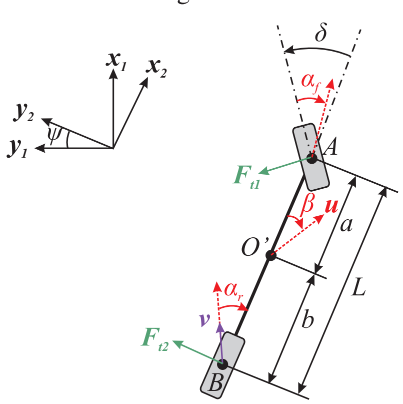


Fig. 3. Vehicle yaw model with sliding parameters.

As previously, longitudinal forces are neglected and parameters h and k_r are assumed to be known by off-line estimation using NSM model (see II-B).

B. Sliding model: equations and behavior

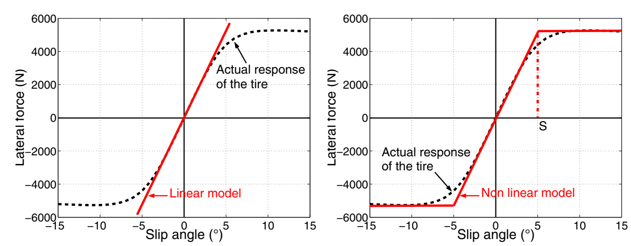
Variations of the parameters accounting for sliding effects have been derived in [12] and are recalled below:

$$\begin{cases} \ddot{\psi} = \frac{1}{I_z}\left(-aF_{t1} + bF_{t2}\right) \\ \dot{\beta} = -\frac{1}{um}\left(C\alpha_f + C\alpha_r\right) - \dot{\psi} \\ \alpha_r = \beta - \frac{b\dot{\psi}}{u} \\ \alpha_f = \beta + \frac{a\dot{\psi}}{u} - \delta \\ u = \frac{v\cos(\alpha_r)}{\cos(\beta)} \end{cases} \tag{9}$$

A linear tire model can only describe pseudo-sliding effects, whereas quad bikes are submitted to actual sliding. Therefore a non-linear model has to be considered (Fig. 4(a)). Famous Pacejka tire model [14] accurately describes such phenomena, but is hardly tractable since numerous and varying parameters need to be known. A simpler model, consisting in adding a non-linear part to the linear tire model is here proposed, as depicted on Fig. 4(b). This model takes into account the saturation of lateral forces, whose final expressions are then:

$$\begin{cases} F_{t1} = sgn(\alpha_f)\cdot\min\left(C\left|\alpha_f\right|, CS\right) \\ F_{t2} = sgn(\alpha_r)\cdot\min\left(C\left|\alpha_r\right|, CS\right) \end{cases} \tag{10}$$

where S is a saturating threshold (considered constant) of the lateral forces.



(a) Linear model. (b) Non-linear model used.

Fig. 4. Tire models.

C. Equations of load transfer

On a flat ground, the expressions of F_{n1} and F_{n2} are still given by (7) and (8). On the contrary, the variations of the yaw rate and of the roll angle are different: $\dot{\psi}$ has to be derived from (9) instead of (3), and the following expression can be obtained for the variation of φ_v:

- $$\ddot{\varphi}_v = \frac{1}{h\cos(\varphi_v)}\Big[h\dot{\varphi}_v^{\,2}\sin(\varphi_v) + h\dot{\psi}^2\sin(\varphi_v) + u\dot{\psi}\cos(\beta) + \dot{u}\sin(\beta) + u\dot{\beta}\cos(\beta) - \left(\frac{k_r\varphi_v + b_r\dot{\varphi}_v}{mh}\right)\cos(\varphi_v)\Big] \tag{11}$$

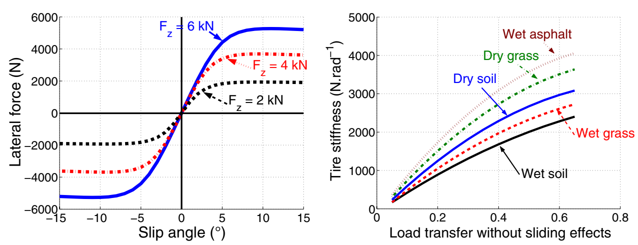
This second semi-analytical model constituted of (7), (8), (9), (10) and (11) is named below WSM (With Sliding Model). However, prior to compute the lateral load transfer from WSM, the tire stiffness C has to be estimated.

D. Estimation of tire stiffness

1) Tire stiffness dependence: Tire stiffness is not constant when the vehicle moves. Indeed, tire stiffness is a function of both tire load (see Fig. 5(a)) and grip conditions.

Contrary to the on-road case, tire load cannot be known, since F_{n1} and F_{n2} are precisely expected to be derived from model WSM. The proposed approach consists in making use of the NSM model in order to approach normal forces and then evaluate tire stiffness. More precisely, an off-line learning process has been carried out: for a given grip condition, numerous simulation trials with model NSM have been achieved, for different values of inputs v and δ. At each time, the load transfer has been computed. Independently, relying on a ground truth (actual quad bikes, or here an Adams model, see V-A), the tire stiffness has been estimated when the same inputs are applied. Therefore, a graph "tire stiffness versus load transfer in absence of sliding" can be drawn. This graph is named a ground class.

In order to describe the different soils that can be met, the same learning process is achieved for several grip conditions. This leads to a network of ground classes, as shown in Fig. 5(b).



(a) Tire stiffness dependence with respect to tire load.

(b) Ground classes.

Fig. 5. Tire stiffness dependence.

2) Tire stiffness evaluation: In order to select the ground class representative for the current grip condition, it is proposed below to rely on the yaw rate according to an iterative procedure. Indeed, the vehicle yaw rate is very sensitive to tire stiffness, as it can be seen from equation (9). Therefore the proposed approach consists in comparing the yaw rate computed from WSM model ($\dot{\psi}_{WSM}$) to a ground truth provided by vehicle sensors ($\dot{\psi}_{measured}$) (for instance a gyrometer or an INS). Then, the ground class selection is carried out by minimizing the difference between the two yaw rate values. This iterative method is represented on Fig. 6 and consists in the following steps:

1) The load transfer without sliding is calculated according to the vehicle current velocity and steering angle,
2) A value for the tire stiffness is obtained from our initial choice for the ground class, see Fig. 5(b),
3) This value is reported into model WSM in order to obtain the load transfer with sliding effects and the expected vehicle yaw rate $\dot{\psi}_{WSM}$,
4) This expected vehicle yaw rate $\dot{\psi}_{WSM}$ is compared to the yaw rate measurement $\dot{\psi}_{measured}$,
5) The ground class is then iteratively adapted until the most suitable class with respect to current grip condition is obtained,
6) Relying on this updated class, the load transfer with sliding is finally available.

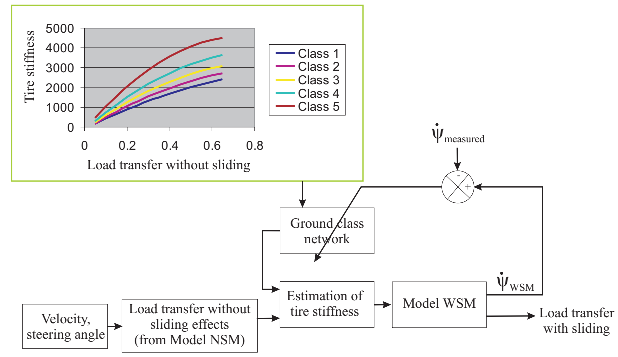


Fig. 6. Calculation algorithm of the load transfer with sliding.

IV. ROLLOVER INDICATOR

As mentioned above, the two main inputs of NSM and WSM models are velocity and the steering angle of the quad bike. On a sloping ground, φ_r should be considered as a third input. In the sequel, for the sake of simplicity, a flat ground is assumed (preliminary work on sloping ground can be found in [2]).

Reporting these two inputs into NSM and WSM models can provide the current values of the load transfer with and without sliding effects. Therefore imminent rollover accidents can be detected. However, in order to be able to apply corrective actions, it would be preferable to anticipate the lateral load transfer on an horizon of prediction (as is done in [5]). This can be achieved by relying on PFC (Predictive Function Control) formalism detailed in [15] and depicted on Fig. 7.

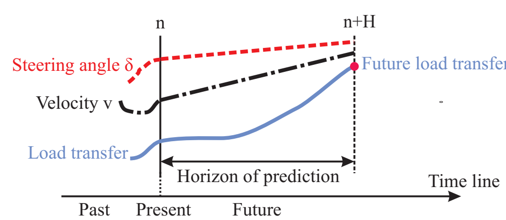


Fig. 7. General description of prediction principle.

First, the future speed $v(n+H)$ and the future steering angle $\delta(n+H)$ are evaluated as follows:

$$\begin{cases} v(n+H) & = & v(n)+H\cdot\dot{v}(n) \\ \delta(n+H) & = & \delta(n)+H\cdot\dot{\delta}(n) \end{cases} \tag{12}$$

H is the horizon of prediction, $(v(n),\ \delta(n))$ are the velocity and the steering angle at present time and $(\dot{v}(n),\ \dot{\delta}(n))$ are the acceleration and the steering rate at present time. Then, relying on the predicted values (12), the steady-state value of the load transfer can be computed, and constitutes the rollover indicator proposed in this paper. If this indicator exceeds the critical value 0.8, it can be anticipated that the ATV is close to rollover. The lookahead horizon must be chosen in such a way that some corrective actions could be performed in the meanwhile. Simulations show that a 2s horizon is relevant.

V. Validation with Adams multibody software

A. Development of the Adams model

The capabilities of the rollover indicator have been investigated with respect to a quad bike model developed in the multibody software Adams. This software is devoted to numerical modelisation. It allows to take into account numerous parameters and elements.

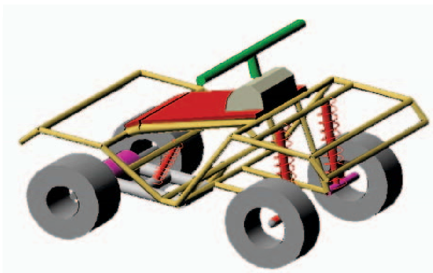


Fig. 8. Quad bike designed with Adams.

Fig. 8 shows the Adams model that has been built. This vehicle is equipped with front suspensions and a rear trailing arm. A differential is present on the vehicle rear axle. The main vehicle parameters are listed in Table I.

TABLE I
Vehicle parameters

Body mass (m)	250 kg
Wheelbase (L)	1250 mm
Track width (c)	950 mm
Center of gravity height from ground	700 mm
Front wheel radius	254 mm
Rear wheel radius	230 mm
Identified distance between O' and G (h)	730 mm
Identified roll stiffness (k_r)	2360 $N.rad^{-1}$

The different tests carried out with Adams model are a first step in the validation of NSM and WSM models. Actual experiments are currently under development.

B. Validation of NSM model

In order to validate NSM model, h and k_r have first to be identified. Therefore Adams model has been run for a given steering angle and for several vehicle velocities, in the case of a high grip ground, and the corresponding load transfer values have been recorded. Then, Newton-Raphson non-linear identification algorithm (see [11]) has been used to identify the values of k_r and h reported in Table I.
Finally, Adams model has been run on a high grip ground for a large set of constant velocity and steering angle values. Fig. 9 shows that such values of h and k_r lead to a satisfactory representativeness of the NSM model with respect to Adams model on high grip ground. Therefore the NSM model is validated.

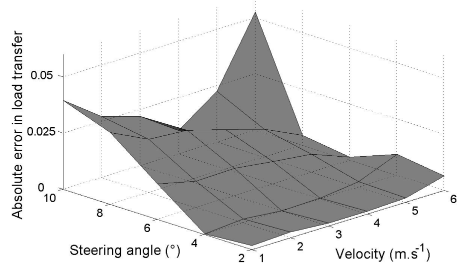


Fig. 9. Absolute errors between Adams and NSM values of load transfer.

C. Validation of WSM model

The calculation algorithm for load transfer with sliding effects is now investigated. Three ground classes have been calibrated relying on Adams model with three different grip conditions. Then, five tests have been carried out with arbitrary grip conditions (different from the three ground classes), velocities and steering angles. The load transfer evaluated from Adams model as well as from NSM and WSM models has been recorded. Table II gathers the different results. The values of v and δ were measured from Adams model.

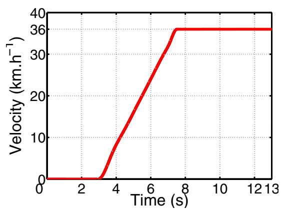
TABLE II
Numerical comparison of load transfer with sliding

Test	1	2	3	4	5
Selected class	low			mid	high
Velocity ($m.s^{-1}$)	5.7	4.6	6.3	3.9	6
Steering angle (°)	8	10	8	6	4
LLT_A: Adams LLT	0.36	0.27	0.44	0.12	0.22
LLT with NSM	0.42	0.34	0.53	0.14	0.22
LLT with WSM	0.37	0.28	0.46	0.13	0.22
Error in % LLT_A vs LLT_{NSM}	16.7	25.9	20.5	16.7	0
Error in % LLT_A vs LLT_{WSM}	2.8	3.7	4.5	8.3	0

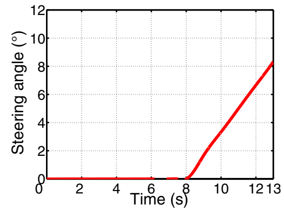
When grip conditions are high (test 5 in Table II), the load transfer computed from NSM or WSM models reflects perfectly the load transfer provided by Adams model. In the case of medium grip conditions (test 4 in Table II), the absolute errors are still very small whatever the model (even if the relative error is quite important with NSM model). On the contrary, in the case of low grip conditions (tests 1, 2 and 3 in Table II) where sliding effects are significant, it can be observed that only WSM model is able to provide a satisfactory load transfer estimation. The relative error provided by WSM model with respect to Adams remains inferior to 5% whereas it climbs up to 25% using NSM model. This shows clearly the relevancy of WSM model and demonstrates the benefit of taking sliding effects into account: since load transfer computed from NSM model is overevaluated with respect to its actual value, the direct application of on-road indicators to off-road vehicles leads to critical situations ($LLT > 0.8$) detection although the vehicle is still far from rollover. On the contrary the load transfer computed from WSM model reflects the vehicle correct situation. As a result, the proposed off-road indicator appears to be relevant whatever the ground conditions.

D. Rollover indicator validation

The off-road indicator based on WSM model is now evaluated. The reference velocity and steering angle imposed in Adams model are shown on Fig. 10. Due to such sharp inputs, the vehicle load transfer exceeds its critical value 0.8 at $t = 11.1s$ in the Adams run.



(a) Velocity.



(b) Steering angle.

Fig. 10. Velocity and steering angle imposed.

Fig. 11 presents the actual load transfer recorded from Adams model (blue line), the instantaneous load transfer estimated from WSM model (black line) and the predicted steady state load transfer (indicator state) with a prediction horizon $H = 2s$ (in red line). First, it can be observed that the instantaneous load transfer (obtained by imposing $H = 0s$) is correctly superposed with the actual one. This demonstrates once more the relevancy of WSM model. Furthermore, the indicator detects a rollover risk at $t = 8.8s$, whereas the actual load transfer reaches the critical value 0.8 only at $t = 11.1s$. Therefore, the rollover indicator lets enough time to activate stabilizing corrective actions. The proposed algorithm appears satisfactory and enables the anticipation of potential risks.

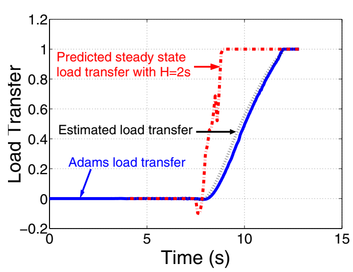


Fig. 11. Prediction results.

VI. Conclusion and Future Works

This paper proposes a rollover indicator dedicated to light ATV, based on the prediction of the lateral load transfer. Unlike indicators designed for on-road vehicles, it integrates off-road environments specificities, which deeply act on the rollover risk. The construction of this new indicator relies on two dynamical models. The first one, based on pure rolling assumption, is dedicated to parameters identification and tire stiffness estimation. The second model permits to introduce the influence of sliding effects into load transfer estimation. It is characterized by a new yaw representation of the vehicle and is relevant for off-road applications, as the grip conditions are continuously estimated thanks to an on-line selection of ground contact parameters previously identified. Contrary to NSM, WSM model enables accurate load transfer estimation in presence of sliding. Finally, predictive control techniques, relying on WSM model, enable to design a relevant rollover indicator whatever the level of grip conditions, as demonstrated in this paper.

Nevertheless, the estimation of tire stiffness, relying on ground classes, may lead to some inaccuracies when grip conditions are far from any ground class. Therefore, new solutions for the estimation of grip conditions, based on the tire/ground parameters observation, are under development. The indicator proposed in this paper is the first step in designing active security systems. Future work is now focused on the design of control laws, which could prevent vehicles from rollover within the time range offered by the lookahead horizon. The last point to be developed is to account for the ATV driver's behavior. Currently a parametrized driver has been implemented into Adams model. Driver's behavior has now to be captured in semi-analytical models by adapting on-line some parameters.

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