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), together with the terrain configuration (off-road environment) prevent from describing rollover occurence as it is proposed for car-like vehicles moving on asphalted roads. This paper proposes a rollover risk indicator dedicated to off-road vehicles such as quad bikes, which integrates the environment properties via grip estimation. It is based on the prediction of the lateral load transfer relying 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 appeared initially in the U.S.A. but are now worldwide marketed. Initially designed for performing agricultural work, they are now largely used for the leisures. Unfortunately, the number of accidents is increasing with this market extension. For instance, in France, in the agricultural area, a French insurance company [1] has reported around fifty ATVs severe accidents every year, since 2003. Most of these accidents are related to rollovers and especially lateral rollovers.

Most of the time, lateral rollovers are due to the centrifugal force together with the inclination of the ground, which both create lateral load transfer on the vehicle tires. According to [2] the high lateral rollover risk with quad bikes follows from their small geometrical dimensions (wheelbase and track) and their small weight. This risk is then increased since quad bikes have the particularity to reach high speeds (sometimes higher than 100 $km.h^{-1}$) on irregular grounds. These features provide a huge maniablity to quad bikes, but they are particularly subjected to rollover.

Rollovers have been tackled but essentially for road vehicles. Numerous security systems have been developped and some of them have been specially devised for dynamic stability. The most common systems are mechanical ones, such as anti-roll bars (see [3]). These systems are added to a major proportion of cars, trucks, tractors and sometimes quad bikes. Another solutions are constituted with automatic systems as active suspensions and some other devices able to modify the steering angle imposed by the driver in order to avoid rollover in critical situations (see for instance [4] and [5]).

However active devices developed for car industry generally rely on dynamic models to represent vehicles. Most of these models do not integrate tire model ([6] and [7]) or only use a linear tire model which can only describe the pseudo-sliding area of the tire (such as in [8]). But contrary to urban vehicles, sliding effects are very significant in ATVs applications. Consequently dynamic road vehicles models could hardly be used to describe quad bikes and to develop stability devices for these vehicles.

Therefore the main goal of this paper is to develop a rollover indicator valid in presence of sliding and able, in future work, to drive automatic control device. 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 sliding effects. This model will be instrumental in providing on-line some parameters. Next, a second semi-analytical model which takes into account sliding effects, thanks to a wheel/ground contact modelisation, is introduced. Then, an algorithm is developped to calculate the lateral load transfer in presence of sliding. These two models are sufficiently simple to be computed in real time and then to be used inside an active security control law. Finally, the rollover indicator is introduced. It relies on the prediction of the future lateral load transfer in order to anticipate rollover situations. This approach has been tested on a multibody model built with the software Adams (broadly used in the car industry) and demonstrate the capability of such indicator to predict rollover situations.

II. Dynamic modeling in absence of sliding

The objective of modelling is to describe the dynamics of the vehicle and the variation of the lateral load transfer. In this paper the vehicle velocity, the steering angle and the ground inclination are the three inputs of the models. The lateral load transfer (or commonly load transfer) is 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 $|LLT|$ exceeds 1, the two wheels lift off and the vehicle could rollover. According to [9], if the lateral load transfer reaches the range [0.8, 0.9] then the quad bike is close to rollover. Since rollover indicator is here intented to be used into stabilizing control laws, the lowest value 0.8 will be considered as the rollover critical threshold.

In order to derive the variation of 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

First, 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 [8] and [10]. It is depicted in Fig.1.

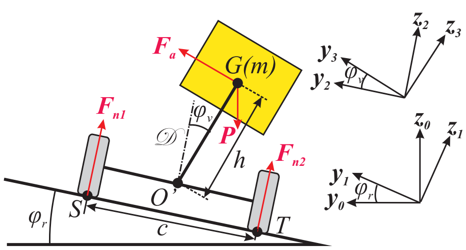


Fig. 1. Vehicle roll model and parameters.

The parameters used in the roll model without sliding are:

- O' is the roll center of the vehicle.
  The roll center location is assumed to be constant. It 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 the roll center and the center of gravity,
- c is the track of the vehicle,
- φ_r is the inclination of the ground,
- φ_v is the roll angle of the suspended mass,
- F_{n1} is the normal force on the left side of the vehicle,
- F_{n2} is the normal force on the right side of the vehicle,
- F_a is a restoring-force associated with the roll movement. This force is considered here 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 in order to derive the normal forces F_{n1} and F_{n2} and therefore LLT, some motion variables of the vehicle have also to be known. These variables can be obtained from a second simplified representation of the vehicle: this representation in the yaw frame known as bicycle or Ackermann model (see [11]) is depicted in Fig. 2.

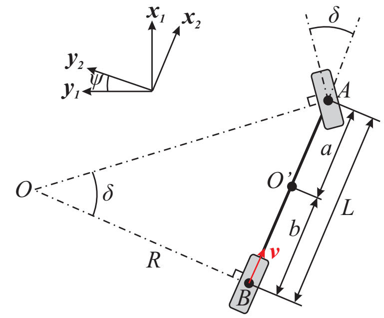


Fig. 2. Vehicle yaw model and parameters.

The parameters used in this part of the 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 wheelbase of the vehicle,
- R is the curvature radius,
- δ is the steering angle,
- v is the vehicle linear velocity at the center of the rear axle.

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

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

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

- 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 diagonal:

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

B. Equations of the load transfer without sliding

Load transfer can then be evaluated 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). $\overrightarrow{M_{G,F_{n1}}}$, $\overrightarrow{M_{G,F_{n2}}}$ the different momenta due to normal efforts $\overrightarrow{F_{n1}}$ and $\overrightarrow{F_{n2}}$ at the center of gravity. From equations (5), variations of φ_v, F_{n1} and F_{n2} can be derived. General equations, which take into account the ground inclination are provided in [12]. On a flat ground ($\varphi_r = 0$), general equations can be simplified as the following ones:

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

- $$F_{n1}+F_{n2}=m\left[-h\ddot{\varphi}_v\sin(\varphi_v)-h\dot{\varphi}_v{}^2\cos(\varphi_v)+g-\left(\frac{k_r\varphi_v+b_r\dot{\varphi}_v}{mh}\right)\sin(\varphi_v)\right] \quad (7)$$
- $$F_{n1}-F_{n2}=\frac{2}{c}\left[I_x\ddot{\varphi}_v+(I_z-I_y)\left[\dot{\psi}^2\cos(\varphi_v)sin(\varphi_v)\right]-h\sin(\varphi_v)(F_{n1}+F_{n2})\right] \quad (8)$$

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 $\dot{\psi}$ which appears in these expressions can be obtained from equations (3), (6). Therefore equations (3), (6), (7) and (8) constitute the semi-analytical model without sliding effects, hereafter noted NSM (Non-Sliding Model). The damping coefficient b_r does not act on the final value of the lateral load transfer, so only two remaining parameters (h and k_r) must be estimated. As detailed in [12], the Newton Raphson method has been implemented. The other parameters (mass, inertial products, etc) can be measured.

III. Dynamic model with sliding effects

The NSM model is significant for calculating the load transfer on high grip ground but gives a bad estimation of the load transfer when sliding occurs. However quad bikes are precisely supposed to move on irregular slippery ground. Therefore another semi-analytical model taking into account for sliding is now built.

A. Notations and modeling with sliding effects

Sliding effects are reflected on the yaw rate of the vehicle because an instantaneous center of rotation does no longer exist. In order to describe such phenomena, a tire model has to be incorporated into the yaw model of the vehicle. The main difficulty is then to choose the tire model and its parameters. The tire model proposed below is derived from a linear model described in [13] and depicted in Fig. 3. 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,
- u is the velocity of the vehicle 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.

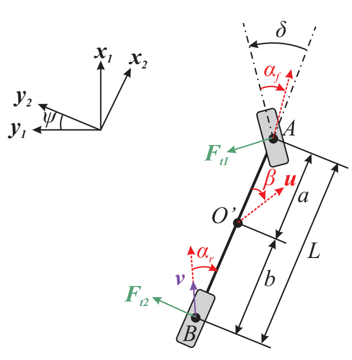


Fig. 3. Yaw vehicle model with sliding parameters.

As previously, longitudinal forces are neglected.

B. Sliding model: equations and behavior

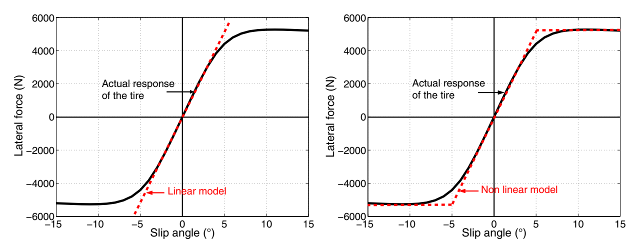
Variations of the parameters accounting for sliding effects have been derived in [14] and are recalled below:

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

A linear tire model can only describe pseudo-sliding effects. However in practical situations, quad bikes are submitted to actual sliding. Therefore a non-linear model has to be considered (Fig. 4(a)). Famous Pacejka tire model [15] accurately describes such phenomena, but cannot be used because of the difficulty in evaluating the numerous parameters. Therefore a simpler model, consisting in adding a non-linear part to the linear tire model, is here proposed as depicted on Fig. 4(b). Then, lateral forces are given by the following expression:

$$\left\{\begin{array}{lcl}F_{t1}&=&sgn(\alpha_f)\cdot\min\left(C\left|\alpha_f\right|,CS\right)\\ F_{t2}&=&sgn(\alpha_r)\cdot\min\left(C\left|\alpha_r\right|,CS\right)\end{array}\right. \quad (10)$$

where C is the tire stiffness and S a saturating threshold (considered constant) of the lateral forces.



(a) Tire linear model. (b) Tire non-linear model.

Fig. 4. Tire models.

C. Equations of load transfer

On a flat ground, the expressions of F_{n1} and F_{n2} are the same as (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 on a flat ground, the following expression can be obtained for the variation of φ_v (from the fundamental principle of the dynamic):

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

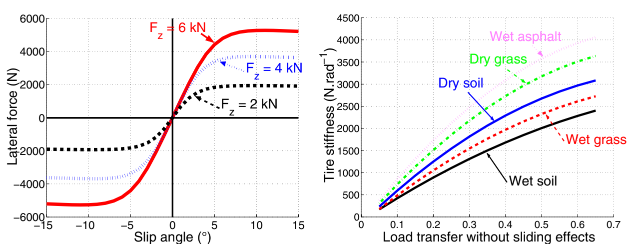
This second semi-analytical model constituted of (7), (8), (9), (10) and (11) is named below WSM (With Sliding Model). However, prior to derive the lateral load transfer from WSM, the tire stiffness C has to be estimated.

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.

Unfortunately, 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 model NSM 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 run, for different values of inputs v and δ, and the load transfer has each time 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.

The same learning process has been 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.

The on-line estimation of tire stiffness in presence of sliding will then consist first in selecting the most suitable ground class. Then, the load transfer calculated on-line via model NSM can be used to obtain the current value of tire stiffness.

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 yaw rate of the vehicle is very sensitive to the tire stiffness as it can be seen from equation (9). Therefore the proposed approach consists in comparing the yaw rate computed from the model WSM ($\dot{\psi}_{WSM}$) to a ground truth provided by vehicle sensors ($\dot{\psi}_{measured}$) (for instance a gyrometer, an INS, etc), and then to select the ground class in order that the difference between the two yaw rate values is minimum. The iterative method is represented on Fig. 6 and consists in the following steps:

1). First, the load transfer without sliding is calculated according to the current velocity and steering angle of the vehicle,

2). From the initial ground class, a value for the tire stiffness is chosen,

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 to the most suitable class with respect to current grip condition,

6). Relying on this updated class, the load transfer with sliding is finally obtained.

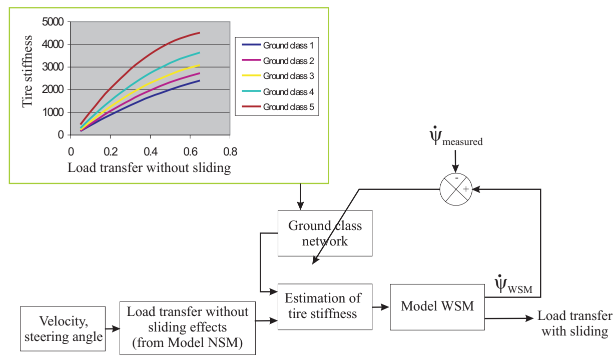


Fig. 6. Calculation algorithm of the load transfer with sliding.

IV. ROLLOVER INDICATOR

As it has already been pointed out, the velocity and the steering angle of the quad bike are two main inputs of NSM and WSM models. 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 [12]).

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 could then be detected. However, in order to be able to apply corrective actions, it would be preferable to anticipate the lateral load transfer on a horizon of prediction. This can be done relying on PFC (Predictive Function Control) formalism detailed in [16] and depicted on Fig. 7.

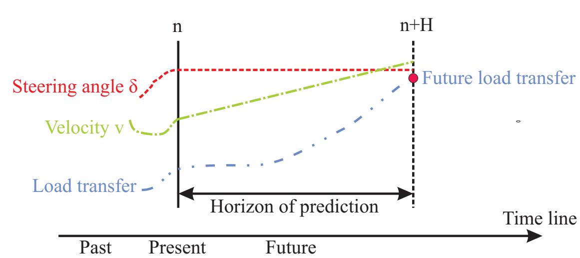


Fig. 7. General description of prediction principle.

According to Fig. 7, the rollover indicator is designed relying on the future variation of the lateral load transfer, calculated from the future speed $v(n+H)$ and the future steering angle $\delta(n+H)$:

$$\left\{ \begin{array}{lcl} v(n+H) & = & v(n) + H \cdot \dot{v}(n) \\ \delta(n+H) & = & \delta(n) + H \cdot \dot{\delta}(n) \end{array} \right. \qquad (12)$$

where H is the horizon of prediction, $v(n)$, $\delta(n)$ the velocity and the steering angle at present time and $\dot{v}(n)$, $\dot{\delta}(n)$ the acceleration and the steering rate at present time. Relying on PFC formalism, at current time n, the expected load transfer at time $n+H$ can be estimated and if it exceeds the critical value 0.8, it can be anticipated that the ATV is close to rollover. The horizon of prediction must be chosen in such a way that some corrective actions could then be performed.

V. Validation with Adams multibody software

A. Development of the Adams model

The capabilities of the rollover indicator has been investigated with respect to a quad bike model developed in the multibody dynamic simulation software package Adams. This software is consecrated to numerical modelisation. It allows to take into account numerous parameters and elements. This is the reason why simulation time could be very long (more than 30 mn in our case) and therefore why such a model is not suitable for real-time applications.

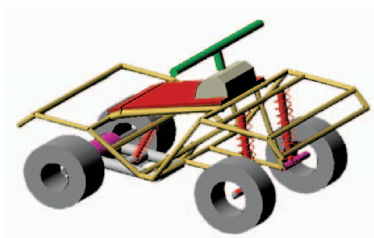


Fig. 8. Quad bike designed with Adams.

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 placed on the rear axle of the vehicle. Finally, 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

The different tests carried out with Adams model are a first step in the validation of NSM and WSM models. Of course, actual experiments would have to be conducted in future work.

B. Validation of the NSM model

In order to validate the NSM model, h and k_r have first to be identified. It has been carried out with Newton-Raphson non-linear identification algorithm, see [17]. More precisely, the 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 algorithm has been used to compute the values of k_r and h ensuring that the load transfer values provided by the NSM model match the ones provided by Adams model. The identification has been achieved with $\delta = 4°$ and v ranging from 1 and 6 $m.s^{-1}$ and has provided $h = 0.73m$ and $k_r = 2360N.rad^{-1}$.

Then it must be checked that these values of h and k_r are satisfactory whatever the value of inputs δ and v. Therefore, the Adams model has been run on a high grip ground for a large set of constant velocity and steering angle values. After stabilization of the lateral load transfer, absolute errors between load transfer values provided by NSM model and by the Adams model are calculated and displayed on Fig. 9. It can be observed that errors are very small so that the NSM model is validated.

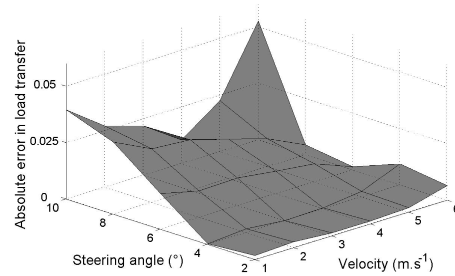


Fig. 9. Absolute errors between Adams and NSM values of load transfer.

C. Validation of the WSM model

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

TABLE II

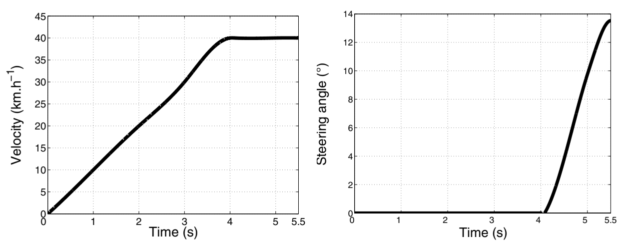
Numerical comparison of load transfer with sliding

Test	1	2	3	4	5
Selected ground 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, whereas WSM model already provides a slightly more satisfactory estimation. 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 the WSM model is able to provide a satisfactory load transfer estimation. The relative error with respect to Adams load transfer remains inferior to 5% when the relative error between load transfer provided by NSM and Adams model climbs up to 25%. This shows clearly the relevancy of WSM model. Moreover this demonstrates the benefit of taking sliding effects into account: since load transfer computed from NSM model is overevaluated with respect to its actual value, critical situations ($LLT > 0.8$) would be detected when the actual vehicle is still far from rollover. On the contrary the load transfer computed from the WSM model reflects the exact situation of the actual vehicle. Therefore it is a relevant indicator in view of developing a rollover indicator.

D. Rollover indicator validation

Rollover indicator is now evaluated. The reference velocity and steering angle imposed in the Adams model are shown on Fig. 10. Due to these sharp inputs, the vehicle rollovers at $t = 5.2s$ on the Adams run.



(a) Velocity. (b) Steering angle.

Fig. 10. Velocity and steering angle imposed.

The Fig. 11 presents the actual load transfer recorded from Adams model (blue line), the instantaneous load transfer computed from WSM model (black line) and the predicted load transfer computed also from WSM model with a prediction horizon $H = 2s$ (in red line) (the instantaneous load transfer has been obtained by simply imposing $H = 0s$). First it can be observed that the instantaneous load transfer is perfectly superposed on the actual one, which demonstrates once more the relevancy of the WSM model. Furthermore, the predicted load transfer with $H = 2s$ announces a rollover risk at $t = 4.2s$ whereas actual rollover occurs at $t = 5.2s$. Therefore, the rollover indicator lets enough time to activate stabilizing corrective actions. This rollover indicator appears satisfactory and enables the anticipation of potential risks.

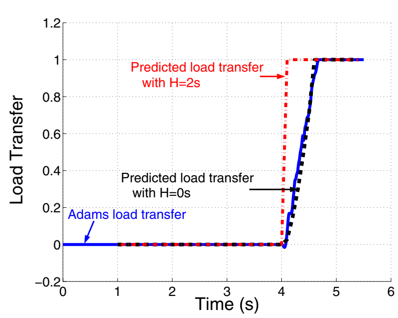


Fig. 11. Prediction results.

VI. Conclusion and Future Works

This paper proposes a rollover indicator dedicated to light ATV, based on the prediction of the lateral load transfer. A first semi-analytical model (named NSM) has been developed assuming rolling without sliding contact conditions. This first model is instrumental in the estimation of some dynamic parameters. Moreover the NSM model is accurate as long as vehicles move on high grip ground. On the contrary the load transfer is overevaluated when sliding occurs. Therefore a second semi-analytical model (named WSM) has been developed to take into account for sliding effects. It is characterized by a new yaw representation of the vehicle and is based on an on-line estimation of grip conditions. Contrary to the NSM, the 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, as demonstrated in this paper.

The next step consists now in designing control laws, which can prevent vehicles from rollover within the time range offered by the indicator. The second point to be developed is to take into account for the behavior of the ATV driver. Currently a parametrized driver has been implemented into the Adams model. Driver's behavior has now to be captured in the semi-analytical models by adapting on-line some parameters.

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