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DYNAMIC MODELING OF ALL TERRAIN VEHICLES DESIGNED FOR DYNAMIC STABILITY ANALYSIS

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Abstract. *The growing popularity of light All Terrain Vehicles (ATVs) as quad bikes is unfortunately accompanied by an increase of accidents and especially lateral overturning. As a result, their faculty to roll over encourage the development of on-board devices so as to prevent ATVs from hazardous situations. However the development of active devices requires a relevant knowledge of ATV lateral dynamic bahavior. So as to reach that point, this paper proposes two dynamic multibody models of ATV corresponding to the majority of quad bikes encountered in the agricultural area. Then, according to both a relevant tire/ground contact modeling and a pilot multibody model, the influence of mechanical components (suspension mechanisms, anti-roll bars, transmission) and parameters variations (sliding, pilot inclination) has been investigated throw a lateral stability criterion represented here by the lateral load transfer of the vehicle. Then, relying on this dynamic lateral stability analysis, it appears that the level of sliding is the main parameter affecting the evaluation of the lateral load transfer and therefore the dynamic stability of quad bikes.*

Therefore, since quad bikes are expected to move on natural and high slippery ground, this paper proposes a rollover risk indicator dedicated to off-road vehicles taking into account the environment properties. It is based on the prediction of the future lateral load transfer computed by a vehicle model including sliding parameters estimation, as the grip conditions are continuously estimated thanks to an on-line selection of ground contact parameters previously identified. Performances of this indicator are demonstrated using the multibody dynamic simulation software Adams and the two multibody models of quad bikes.

1 INTRODUCTION

The market of all terrain vehicles (ATVs) and especially quad bikes has increased very quickly and particularly in the agricultural branch. Indeed, the number of ATVs has increased by 40% in the U.S.A. in 2001 [6] and by 38% in France in 2005 [5].

Unfortunately, the number of accidents increases dramatically. For instance, in the U.S.A., the CPSC (Consumer Product safety Commission) has reported 7 188 deaths between 1982 and 2005 and has made a list of 136 700 injuries only for the year 2005 [6]. In the meantime, a french insurance company has reported an average of 50 serious accidents only in the agricultural branch since 2003 [5]. Laslty, the safekids New Zealand injury prevention service [19] has announced that:"ATV related mortality and morbidity respectively constitutes 6% of all motor vehicles related mortality and morbidity in New Zealand" and added: "Most of the injuries associated with ATVs occur when the driver loses control on hard uneven surfaces and the vehicle rolls over".

Their design, dedicated to huge driveability, is especially unstable on natural and irregular surfaces. Indeed, according to [18] and [2], the fact that ATVs can reach high speed on irregular ground, together with geometrical and mass characteristics (small wheelbase, small track, high center of mass and small weight relatively to the pilot), encourage severe accidents and especially lateral rollover.

Developments proposed in this paper are focused on the design of on-board devices so as to prevent ATVs from rollover accidents. To reach that point, one will be able to rely on the numerous devices dedicated to the dynamic stability of on-road vehicles: mechanical ones such as anti-roll bars [8] and electronical ones such as active suspensions, steering and braking control ([1] and [16]).

The development of active devices dedicated to rollover prevention of off-road vehicles requires a relevant knowledge of rollover dynamics. Then, the first step presented here consists in analysing the behavior of quad bikes when the vehicle is close to rollover. Therefore, multibody modeling is used to understand the dynamics of the vehicle in a position to overturn.

It is in this spirit that the following approach has been used: first, two multibody models of all terrain vehicles have been designed with the multibody dynamic software Adams. These two models have been built in order to represent the majority of quad bike morphologies that can be encountered in the agricultural area. Moreover the multibody models allow to supply realistic tendencies of the vehicle behavior. Then, a stability criterion, which consists in the LLT (Lateral Load Transfer) measurement is presented. It will permit to evaluate the propensity of the vehicle to rollover.

Next, an analytical model sufficiently simple to be computed in real time has been achieved. This model is based on vehicle roll and yaw frames and it takes into account sliding effects thanks to wheel/ground contact modeling. Furthermore, this model allows to compute a rollover risk indicator based on the prediction of the future lateral load transfer and allows to anticipate hazardous situations.

Finally, this indicator has been tested using both multibody models and demonstrates its capabilities in predicting off-road vehicle rollover. Different tests have also been carried out with the multibody models in order to study the influence of various parameters (mechanical components, sliding, pilot...) on the lateral dynamic stability of quad bikes.

2 ADAMS MULTIBODY MODELING

2.1 QUAD BIKES MODELS

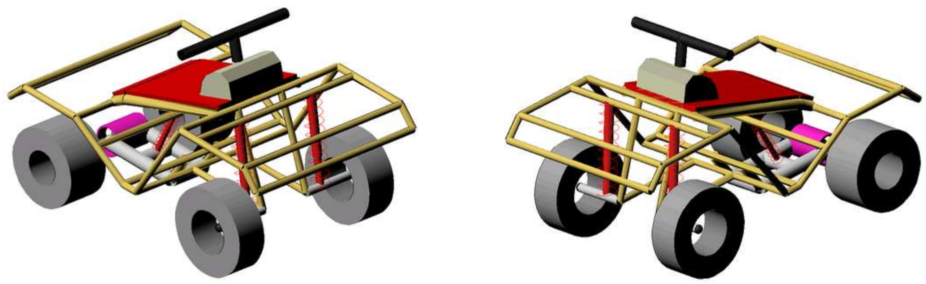
The Adams software has been used in order to model two different ATVs representative of the two main morphologies of ATV. This software (broadly used in the car and flight industry) is devoted to dynamic simulation and numerical modeling so as to study and to improve complex mechanical assemblies without real prototype. As a consequence, the quad bike models have been designed in order to study the dynamic stability of ATVs on natural and irregular ground and designed to represent a large part of quad bikes encountered in the agricultural branch. The main hypotheses linked with the modeling are the following:

- The ATV frames are modelled by non-deformable bodies (cylinders assembled via boolean operations).
- Tire/ground contact is modeled with impact normal force between two solids (here the tire, modeled by a rigid cylinder, and the ground, represented by a box) and with Coulomb friction model since no experimental data are available to estimate the parameters of a more complex Pacejka model (such as [13]) and particularly for quad bike tires. Moreover, since the impact normal force model is parametrized by a damping coefficient, this parameter allows to take into account the tire distortion at the wheel/ground contact point.
- The suspended mass is assumed to be represented by a parallelepiped. Therefore, this mass is symmetrical with respect to the roll and pitch planes of the vehicle. The inertial matrix is then diagonal at the center of gravity of the quad bikes:

$$I_G = \begin{bmatrix} I_x & 0 & 0 \\ 0 & I_y & 0 \\ 0 & 0 & I_z \end{bmatrix} \quad (1)$$

- All the parameters of the quad bikes: wheelbase, track, position of the center of gravity and inertial properties are assumed to be variable and can be modified easily so as to perform a sensitivity analysis.

Then, relying on theses hypotheses, the first model has been developed. This model is a Semi-Independent Suspension Model (hereafter noted SISM and depicted on Fig. 1) because it is composed of a rear trailing arm and front independent suspensions. Moreover, SISM includes a differential placed at the rear axle of the vehicle.



(a) Front isometric view of SISM. (b) Rear isometric view of SISM.

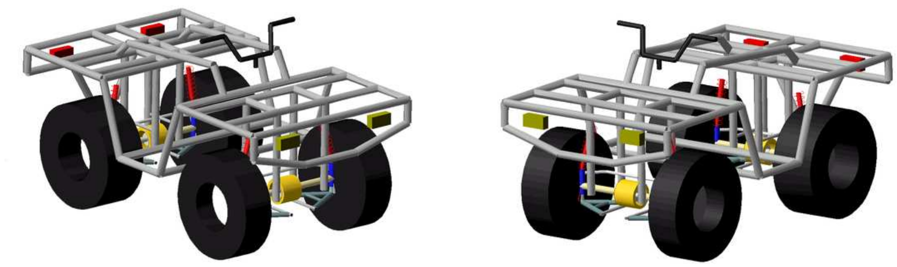
Figure 1: Isometric views of Semi-Independent Suspension Model (SISM).

The main SISM parameters are listed in Table 1 and represent the average parameters of a sporting quad bike.

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

Table 1: SISM parameters

The second model is composed of McPherson suspensions at the rear and the front of the vehicle. Therefore this model is called the Independent Suspension Model (noted ISM and depicted on Fig. 2) and is equipped with rear and front differentials. Furthermore, it is constituted of rear and front anti-roll bars.



(a) First isometric view of ISM. (b) Second isometric view of ISM.

Figure 2: Isometric views of Independent Suspension Model (ISM).

Such as in the case of the SISM, the main parameters of the ISM are listed in Table 2 and represent the average parameters of a utilitirian ATV. Then, one can check that the tire radius of SISM and ISM are different since the two quad bikes are used for different activities. Moreover, so as to compare the two morphologies of ISM and SISM, the same tire radius have been used.

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	330 mm
Rear wheel radius	330 mm

Table 2: ISM parameters

2.2 TIRE/GROUND CONTACT MODELING

Quad bikes are precisely supposed to move on irregular and slippery ground. As a consequence, tire/ground contact modeling is very important in order to have a good representation of sliding parameters such as sideslip angles and longitudinal sliding. Therefore, since the Pacejka model can not be used (because no response curves of quad bike tires are available so as to identify the Magic Formula parameters), the tire/ground contact has been modeled, on one hand, by an Impact-function-based-contact for normal forces and, in another hand, by a Coulomb-friction model for lateral and longitudinal forces of the tire/ground interface. In fact, normal forces generated by the impact force are modeled like in a nonlinear spring damper and according to Coulomb model, lateral and longitudinal forces depend on friction coefficient and on slip velocity of the contact point between tire and ground.

Then, according to Pacejka model [12], sliding is represented by two main parameters attached to each tire: the sideslip angle (hereafter noted α) and the longitudinal sliding coefficient (g). Therefore, in order to show the relevancy of the Coulomb-friction model, several tests have been carried out and sideslip parameters have been computed according to the following expression of α and g:

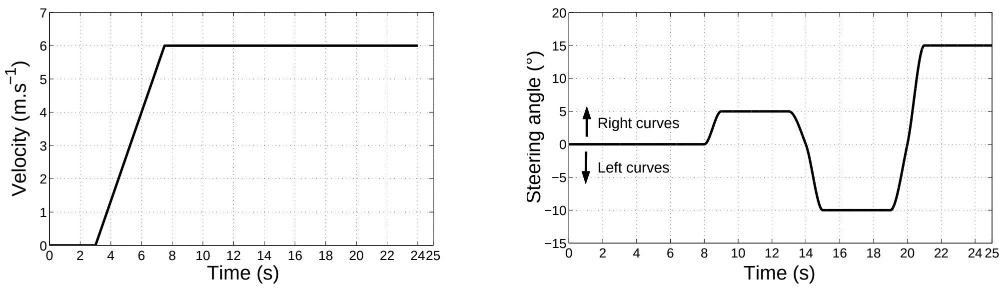
- Sideslip angle (α_i) corresponding to each wheel ($i = 1, 2, 3, 4$), is representative of the difference between the theoretical orientation of the speed vector at the tire centre and the actual one. Then, in function of lateral velocity of the tire center (noted V_{yi}) and the longitudinal velocity of the tire center (V_{xi}) in the vehicle frame, the expression of sideslip angle is:

$$\alpha_i = atan\left(\frac{V_{yi}}{V_{xi}}\right) \tag{2}$$

- The longitudinal sliding coefficient g_i, where i is also used to denote each wheel of the vehicle, is representative of the difference between theoretical longitudinal speed of the tire, computed with the angular speed of the wheel (ω_i) and the tire radius (R), and the real longitudinal velocity of the tire center (V_{xi}). Then, the expression of g_i is:

$$g_i = \left(\frac{V_x - R\omega}{Max(|V_x|, |R\omega|)}\right) \tag{3}$$

As a result, so as to show the relevancy of Coulomb friction model, one simulated test has been done relying on the reference velocity and on the steering angle imposed in the ISM (shown in Fig. 3), the evolution of sideslip angle and longitudinal sliding parameters, computed by expressions (2) and (3) , are presented on Fig. 4.



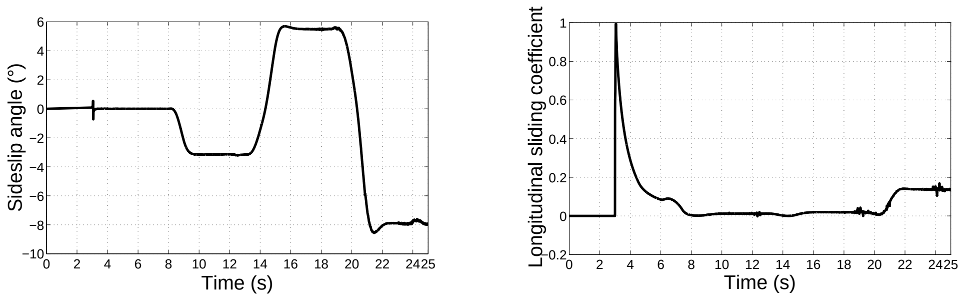
(a) Velocity.

(b) Steering angle.

Figure 3: Velocity and steering angle imposed to the ISM.

Fig. 4(a) presents the sideslip angle of the rear left tire of the ISM computed with expression 2. First, it can be logically observed that when the absolute value of the steering angle grows up, the absolute value of the sideslip angle also increases. Finally the sign of the sideslip angle changes as the curvature sign of the quad bike changes. As a consequence, Coulomb-friction model allows to obtain a good representation of the sideslip angle behavior in comparaison with the famous Pacejka model. Then, Fig. 4(b) presents the longitudinal sliding coefficient evolution of the rear left tire of the ISM computed with expressions 3. In this case at time $t = 3s$, when the vehicle starts to move, the longitudinal sliding coefficient is maximal and equal to the value of 1. This demonstrates the relevancy of Coulomb-friction model in the starting phase where sliding effects are very important. Secondly, one can checked that during the second curve the sliding coefficient is almost equal to the sliding coefficient of the first bend whereas steering angle has increased between the first and second curve. Indeed, since ISM is equipped with a rear diferential mechanism, then in the first curve to the right, the rear left wheel is located at the exterior of the bend contrary to the second curve where the wheel is situated at the interior of the bend. Therefore in the first curve the wheel has an angular velocity more important than in the second one. In conclusion, the rear ISM differential affects the evaluation of the sliding coefficient.

Finally, Coulomb-friction model seems accurate for the computation of sliding coefficients and more generally in sliding parameters according to Pacejka parameters expressions and behavior and permits to take into account sliding effects in Adams simulations.



(a) Sideslip angle at the rear left tire of the ISM.

(b) Longitudinal sliding coefficient at the rear left tire of the ISM.

Figure 4: Sliding parameters at the rear left tire of the ISM.

2.3 PILOT MODELING

According to [2], since the pilot represents 32% of the ATV average weight, the driver largely acts upon the dynamical behavior of the quad bike thanks to his mass and inertial properties compared with the quad bike average characteristics. Then, so as to study the influence of the pilot on the dynamic stability of ATVs, a dynamic modeling of a pilot has been developed. All the driver limbs are assumed to be modeled by ellipsoids. The geometrical characteristics of the pilot come from anthropometrical data [11] representative of a middle size male with a stature of 1.78m and weight of 78kg.

Figure 5: Sitting position of the pilot on the ISM.

Fig. 5 presents the sitting position of the pilot on the ISM. Then, the pilot model is composed of 12 revolute joints and 2 spherical joints. These features provide 18 degrees of freedoms. Motion of the pilot are controlled with respect to steering angle.

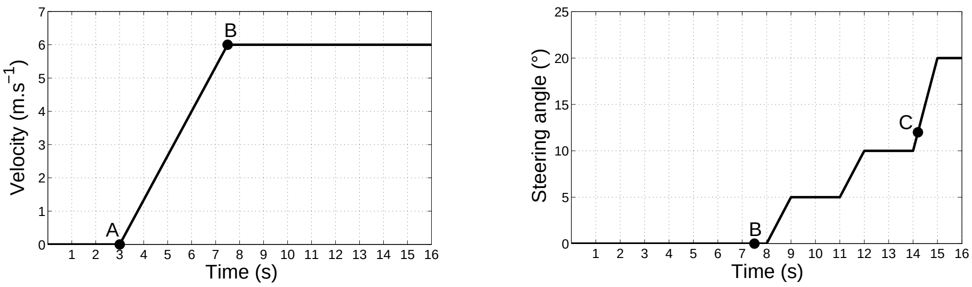
3 STABILITY CRITERION

In order to study the dynamic stability of quad bikes, a stability indicator has been chosen. As pointed out in [16], there are many stability criteria. In this paper risk comparison has been achieved with respect to the Lateral Load Transfer (hereafter noted LLT) expressed by the following expression:

$$LLT = \left(\frac{F_{n2} - F_{n1}}{F_{n2} + F_{n1}} \right) \tag{4}$$

where F_{n1} and F_{n2} are the normal forces applied on the left and right sides of the vehicle. The LLT is at the origin of the rollover phenomenon and therefore it is the most suitable stability criterion in order to point out a rollover risk and to reflect the dynamical behavior of the vehicle. Indeed, it is considered that rollover begins to occur when two wheels of the same side of the vehicle have lifted off. Relying on expression (4) this hazardous situation is effective if $|LLT|$ reaches the largest possible value of 1.

Relying on LLT expression, a simulation test has been carried out in order to show the dynamical behavior of a quad bike. The velocity and steering angle inputs are depicted on the following figure:

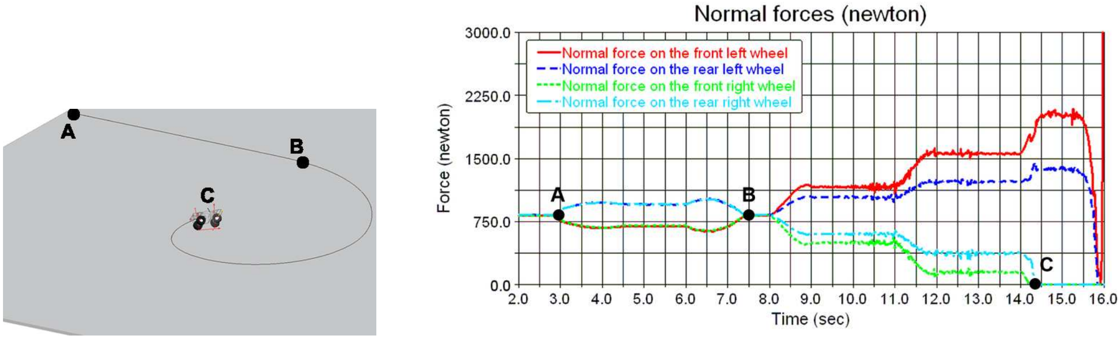


(a) Velocity.

(b) Steering angle.

Figure 6: Velocity and steering angle imposed to the ISM for lateral load transfer analysis.

Therefore, the vehicle path and the normal forces recorded from ISM are presented on Fig. 7 where the vehicle moves on a high grip ground.



(a) Vehicle path.

(b) Normal forces.

Figure 7: Vehicle path and normal forces evolution.

As it can be seen on Fig. 6, the vehicle starts to move at time $t = 3s$ (point A) and reaches a constant velocity at $t = 7.5s$ (point B). During this starting phase, the quad bike acceleration is not equal to zero. This is characterized by a pitch movement of the vehicle. Therefore between $t = 3s$ and $t = 7.5s$, on Fig. 7(b), the rear normal forces applied to the ATV are superior to the front ATV normal forces. The ISM starts to rollover at time $t = 14.2s$ (point C) when the two wheels situated on the right side of the vehicle have lifted off.

Then, on Fig. 7(b), at $t = 14.2s$ the normal forces applied to the right wheels are equal to the value of 0. Finally, as it is depicted on Fig. 8, the lateral load transfer gradually increases as the steering angle grows up at times $t = 9s$, $t = 12s$ and $t = 14s$, the LLT reaches the critical value of 1 at time $t = 14.2s$.

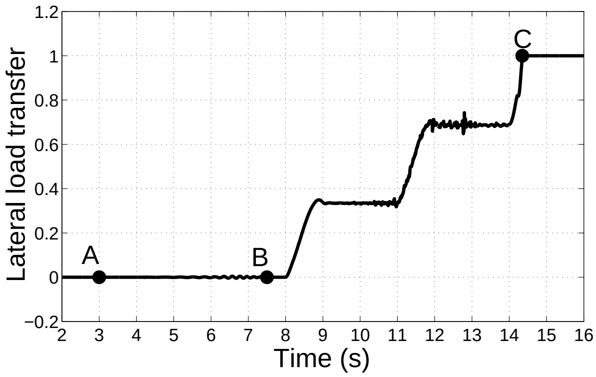


Figure 8: Lateral load transfer.

4 SEMI-ANALYTICAL MODELING

The objective of the vehicle semi-analytical modeling is to compute on-line the lateral load transfer imposed to the vehicle. In order to reach this aim, two semi-analytical models have been designed where the velocity, the steering angle and the ground inclination are the three model inputs, assumed to be available, while the LLT is the model output. The first model called Non Sliding Model (noted NSM) has been developed under rolling without sliding contact condition. This model is constituted with a yaw frame and a roll frame, depicted in [4] and constitutes the first step dedicated to the vehicle dynamic parameters identification. As a result, the roll stiffness and the distance between the vehicle roll center and the vehicle center of gravity can be identified thanks to the Newton-Raphson nonlinear identification algorithm [9]. The other parameters (inertial products, wheelbase...) are supposed to be measured.

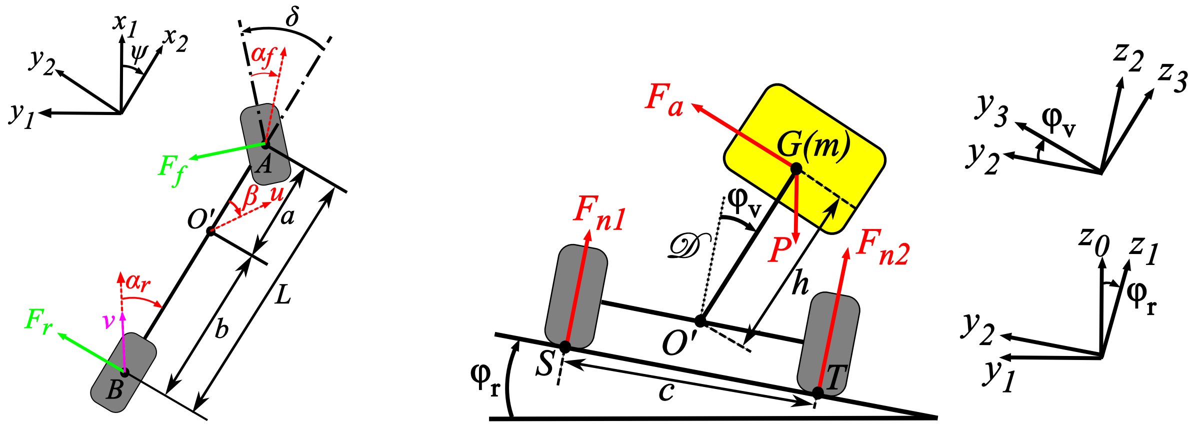
After the validation of NSM with Adams models and since ATVs are precisely supposed to move on natural and slippery ground, a new semi-analytical model has been developed and called With Sliding Model (noted WSM). It will be presented in the following part.

4.1 VEHICLE MODELING

As it is the case for NSM, WSM leans on two frames. The first frame (depicted on Fig. 9(a)) is the yaw frame represented by an Ackermann model where sliding parameters have been added as in [10]. The parameters introduced in this modeling are:

- O' is the vehicle roll center. The roll center location is assumed to be constant,
- a and b are respectively the front and rear vehicle half-wheelbases,

- L is the vehicle wheelbase,
- δ is the steering angle,
- v is the vehicle linear velocity at the center of the rear axle,
- ψ is the vehicle yaw angle,
- β is the global slip angle of the vehicle,
- α_r and α_f are respectively the rear and front vehicle slip angles,
- u is the vehicle velocity at the roll center,
- F_f and F_r are respectively the lateral forces generated on the front and rear tires.



(a) Yaw frame of the WSM.

(b) Roll frame of the WSM.

Figure 9: With Sliding Model (WSM).

Then, in order to extract normal forces applied on the vehicle, and relying on the vehicle motion variables stemming from yaw frame, a second simplified representation of the vehicle in its roll frame has been used as in [15] and depicted on Fig. 9(b).

The second set of parameters introduced by the roll frame are:

- 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} and F_{n2} are respectively the normal forces on the vehicle left and right sides,

- 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 roll stiffness coefficient and b_r the roll damping coefficient. The expression of this force is related to the roll movement by equation (5):

$$\overrightarrow{F_a} = \frac{1}{h}\left(k_r \varphi_v + b_r \dot{\varphi}_v\right) \overrightarrow{y_3} \tag{5}$$

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

- As it is the case for Adams modeling, 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:

$$I_{G/R_3} = \begin{bmatrix} I_x & 0 & 0 \\ 0 & I_y & 0 \\ 0 & 0 & I_z \end{bmatrix} \tag{6}$$

- Longitudinal forces are neglected,

- Parameters h and k_r are assumed to be known by off-line estimation using NSM.

4.2 LOAD TRANSFER WITH SLIDING EFFECTS

4.2.1 TIRE MODELING

New sliding parameters have been introduced in the yaw frame of the vehicle, then a tire model has to be chosen. The famous Pacejka model [13] is precisely supposed to be the most suitable for reflecting the sliding phenomenon but this model is difficult to use, in the case of natural environment, since numerous parameters, which can be variable, need to be known. Therefore, a nonlinear simpler model is used and is depicted on Fig. 10. This model, described in [4], relies on a linear tire model and takes into account the saturation of the lateral forces when rear or front slip angle are supposed to exceed a slip angle threshold noted S on Fig. 10.

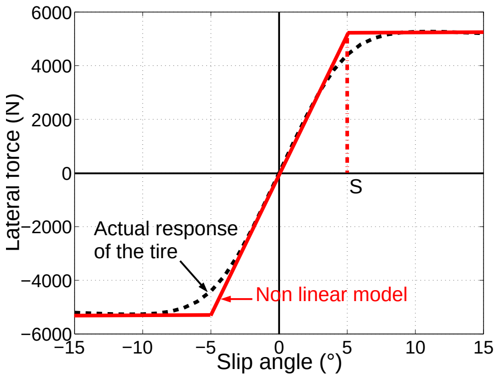


Figure 10: Nonlinear tire model used.

According to this tire modeling and with the tire cornering stiffness parameter (C), lateral forces expressions can be defined as:

$$\left\{ \begin{array}{lcl} F_f & = & sgn(\alpha_f) \cdot \min\left(C\left|\alpha_f\right|, CS\right) \\ F_r & = & sgn(\alpha_r) \cdot \min\left(C\left|\alpha_r\right|, CS\right) \end{array} \right. \tag{7}$$

4.2.2 EQUATIONS OF LLT

In function of the nonlinear tire model equations expressed in [4] and the roll frame representation, variations of φ_v, F_{n1} and F_{n2} can be computed on an irregular and slippery ground. General equations, derived from the fundamental principle of the dynamic, are provided in [3] and on a flat ground ($\varphi_r = 0$), these equations are:

$$\ddot{\varphi}_v = \frac{1}{h\cos(\varphi_v)}\left[h\dot{\varphi}_v{}^2\sin(\varphi_v) + h\dot{\psi}^2\sin(\varphi_v) + u\dot{\psi}\cos(\beta) + \dot{u}\sin(\beta) + u\dot{\beta}\cos(\beta) - \left(\frac{k_r\varphi_v + b_r\dot{\varphi}_v}{mh}\right)\cos(\varphi_v)\right] \quad (8)$$

$$F_{n1} + F_{n2} = m\left[-h\ddot{\varphi}_v\sin(\varphi_v) - h\dot{\varphi}_v{}^2\cos(\varphi_v) + g - \left(\frac{k_r\varphi_v + b_r\dot{\varphi}_v}{mh}\right)\sin(\varphi_v)\right] \quad (9)$$

$$F_{n1} - F_{n2} = \frac{2}{c}\left[I_x\ddot{\varphi}_v + (I_z - I_y)\left[\dot{\psi}^2\cos(\varphi_v)sin(\varphi_v)\right] - h\sin(\varphi_v)(F_{n1} + F_{n2})\right] \quad (10)$$

As a consequence, from equations (9) and (10), normal forces can be deduced if the vehicle yaw rate is well computed. Indeed, according to tire modeling equations expressed in [4], tire cornering stiffness has to be known in order to derive the yaw rate value $\dot{\psi}$, which appears in (8) and (10), and the value of LLT. Moreover, the tire cornering stiffness is a function of both the load transfer and the grip condition, as a result, since ATV are expected to move on natural ground, the tire cornering stiffness is not constant when the vehicle moves. Therefore, an identification algorithm dedicated to tire cornering stiffness estimation has been developed.

4.3 TIRE STIFFNESS ESTIMATION

Tire stiffness is not constant when the vehicle moves. Indeed tire stiffness is a function of both load transfer [17] and grip conditions [13].

Then, the tire stiffness estimation algorithm consists in using NSM in order to approach lateral load transfer and then estimate tire stiffness. More precisely, tire stiffness estimation is based on an off-line learning process. This learning process works as follow: for one grip condition, in order to take into account normal force dependence, it consists in making numerous trials for different values of velocity and steering angle and computing the lateral load transfer provided by NSM. At the same time and relying on LLT computation with WSM, the tire stiffness is estimated in comparison with the lateral load transfer with sliding effects measured on a ground truth (here Adams models). Finally, the same learning process is repeated for several grip conditions. This results in a ground classes network as depicted on Fig. 11.

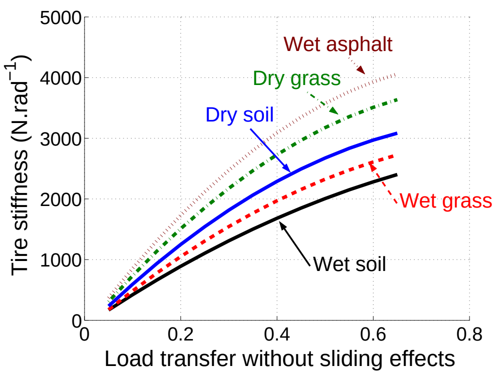


Figure 11: Ground classes network.

4.4 LATERAL LOAD TRANSFER COMPUTATION ALGORITHM

As it is mentioned in [4], lateral load transfer with sliding effects is computed by using an iterative method in order to select the ground class representative for the current grip condition. The computation algorithm is represented on Fig. 12. The first step of this method consists in the calculation of the lateral load transfer relying on NSM, on the current velocity and on the steering angle of the quad bike. Secondly, a value of the tire stiffness is derived from intial ground class (in practice a ground class representative of the higher grip condition) and permits to compute a value of LLT with sliding effects and a value of the vehicle yaw rate $\dot{\psi}_{WSM}$. This value $\dot{\psi}_{WSM}$ is compared with the vehicle yaw rate measured by a gyrometer and noted $\dot{\psi}_{measured}$. Then, relying on this comparison, the ground class is updated to the most suitable with respect to the current grip condition. Finally, the LLT with sliding effects is computed relying on the updated tire cornering stiffness.

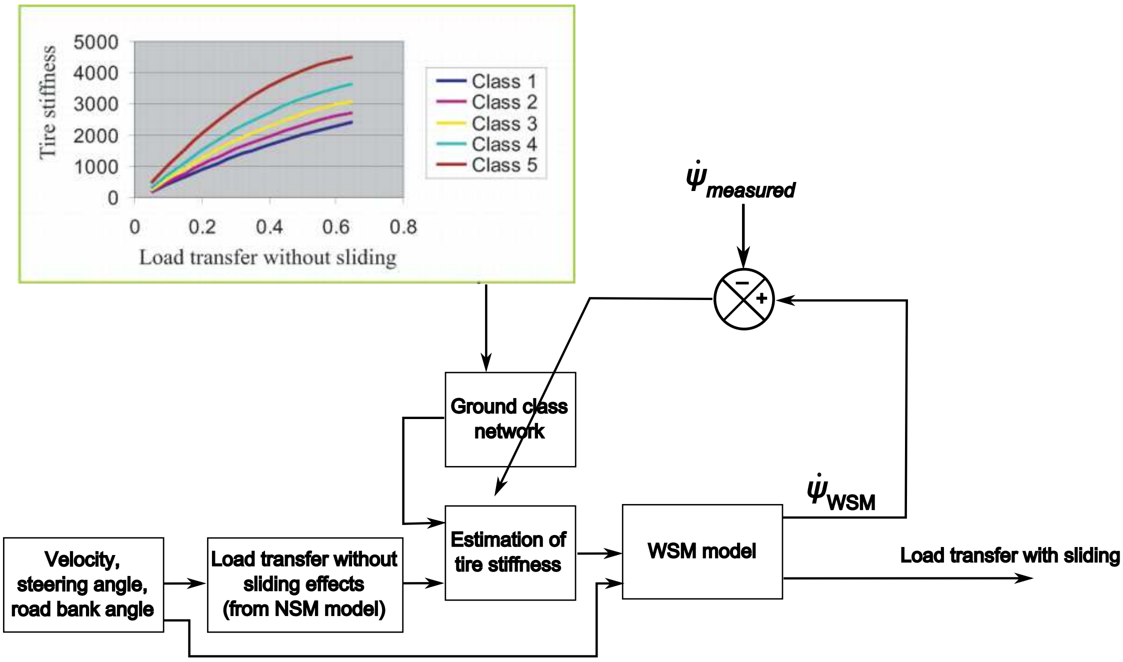


Figure 12: Lateral load transfer computation algorithm.

4.5 ROLLOVER RISK INDICATOR

As it has been seen on Fig.12 The velocity, the steering angle and the road bank angle are the three inputs of WSM.

However for the sake of simplicity, a flat ground is considered. Then, the actual value of the lateral load transfer with sliding effects can be computed relying on the velocity and the measured steering angle. As a consequence, as chosen currently (see [7]), we generally supposed that when the LLT exceeds the value of 0.8 the vehicle is very close to rollover and imminent hazardous situation can be detected. Nevertheless, since the aim of our work is focused on the design of on-board devices so as to prevent ATVs from rollover accidents, PFC (Predicitve Function Control) formalism detailed in [14] is used to anticipate rollover situations on a lookahead horizon, hereafter noted H. On this horizon of prediction, the future steady state value of LLT is derived according to the following method detailed on Fig. 13.

This method consists in evaluating the future value of the velocity $v(t+H)$ and of the steering angle $\delta(t+H)$ relying respectively on their actual values $v(t)$, $\delta(t)$ and their first derivative at present time $\dot{v}(t)$, $\dot{\delta}(t)$ as it is proposed in the following expression:

$$\begin{cases} v(t+H) &= v(t) + H \cdot \dot{v}(t) \\ \delta(t+H) &= \delta(t) + H \cdot \dot{\delta}(t) \end{cases} \tag{11}$$

Then, the future steady state value of the LLT is computed relying on the future inputs of WSM. After a transient phase, the future final value of LLT constitutes the rollover risk indicator proposed in this paper. Indeed, if this indicator exceeds the value of 0.8, then rollover situation can be anticipated.

Finally, simulations show that a lookahead horizon of 2s is accurate to perform some corrective actions.

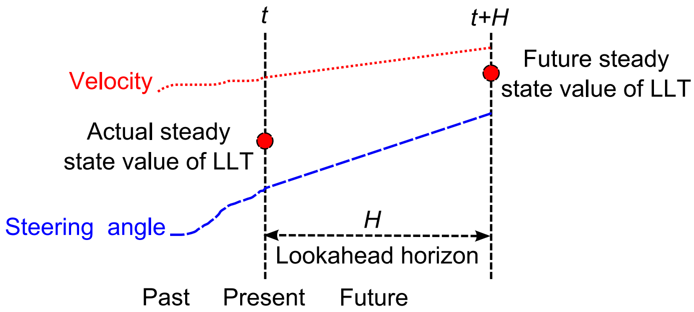


Figure 13: Predicitve principle for rollover anticipation.

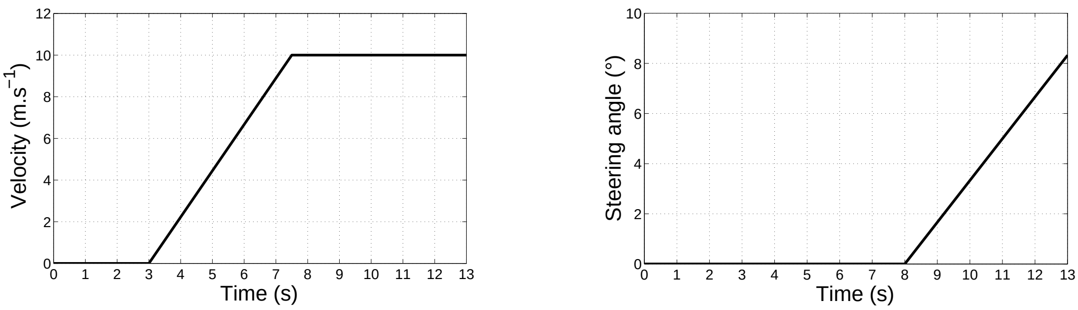
As a result, both WSM and the rollover risk predictive principle has been presented. Therefore, the aim of the next part is to validate this principle when the quad bike is in a hazardous situation on a flat slippery ground.

5 RESULTS

In this section, the first part is dedicated to the validation of the rollover risk indicator on a critical situation. In a second part, the influence of some parameters variation (sliding, pilot inclination) as well as the impact of some mechanical components (suspension morphologies, transmission, anti-roll bars) on the rollover risk is investigated.

5.1 VALIDATION OF ROLLOVER RISK INDICATOR

The proposed indicator has been evaluated on a flat slippery ground with ISM. The reference velocity and steering angle imposed are depicted on Fig. 14. Relying on these two curves (mainly because of the steering angle ramp), the ISM lateral load transfer exceeds the value of 0.8 at $t = 11.1s$.



(a) Velocity.

(b) Steering angle.

Figure 14: Reference velocity and steering angle imposed in ISM.

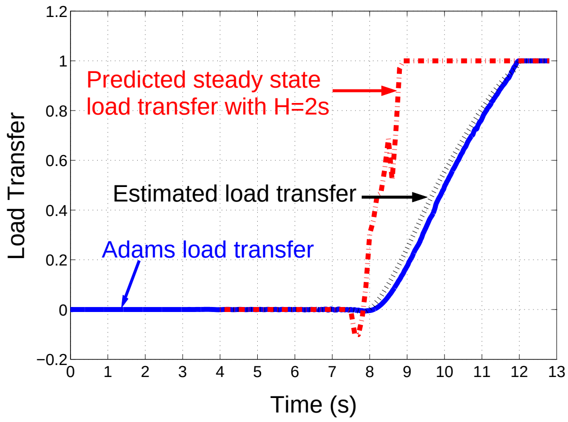


Figure 15: Rollover risk indicator results.

Fig. 15 shows the lateral load transfer recorded from ISM (blue solid line), the instantaneous lateral load transfer computed with WSM ($H = 0s$ in black dotted line) and the future steady state value of the LLT calculated with a lookahead horizon of 2s (red dashed dotted line). Then, one can checked that WSM is relevant to estimate the actual value of the LLT measured on Adams model. Moreover, the rollover indicator risk detects a hazardous situation at time $t = 8.8s$ contrary to the instantaneous one which exceeds the critical value of 0.8 at $t = 11.1s$. As a consequence, the rollover risk indicator lets enough time to avoid the critical rollover situation. This avoidance is expected to be done, in future work, with stabilizing control law acting on both the velocity and the steering angle.

5.2 DYNAMIC STABILITY ANALYSIS

The objective of this part is to study the dynamic lateral stability of quad bikes in function of several phenomena and parameters. For each analysis, the different surfaces represent the steady state value of the lateral load transfer provided by the Adams models when several curves are taken with different constant velocities and steering angle values.

5.2.1 MORPHOLOGY ANALYSIS

The influence of the quad bike morphology has been investigated with respect to the two Adams models developed. Indeed, ISM and SISM represent two kinds of quad bikes since their suspension mechanisms are different. Fig. 16 shows how the LLT rises when the velocity ranging from 1 and 6 $m.s^{-1}$ and the steering angle is bounded by 2 and 10°. The different tests have been carried out on a high grip ground where both ISM and SISM are equipped with a two-wheel drive transmission. The vehicle parameters used for the simulations are listed in Table 1 and Table 2, however so as to perform the simulation, all the design parameters of SISM and ISM are equal (tire radius, position of the center of gravity, inertial products...).

On Fig. 16, one can check that when the LLT increases, the difference between the two models (ISM and SISM) also rises until this difference reaches his maximum when the value of the velocity and the value of the steering angle are also maximal. Therefore, one can check that ISM type is more stable than SISM. Indeed, in this case and relying on the stability criterion, maximal LLT value is of 0.67 for SISM, while it reaches only the value of 0.59 for ISM. Finally, it can be noticed that in agreement with this study, the actual market of quad bikes is turned to the development of utilitarian quad bikes composed of full independent suspensions leading to a better lateral dynamic stability.

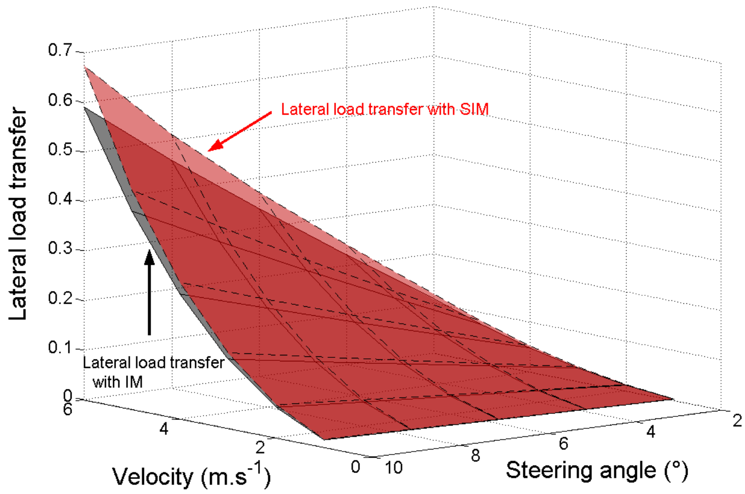
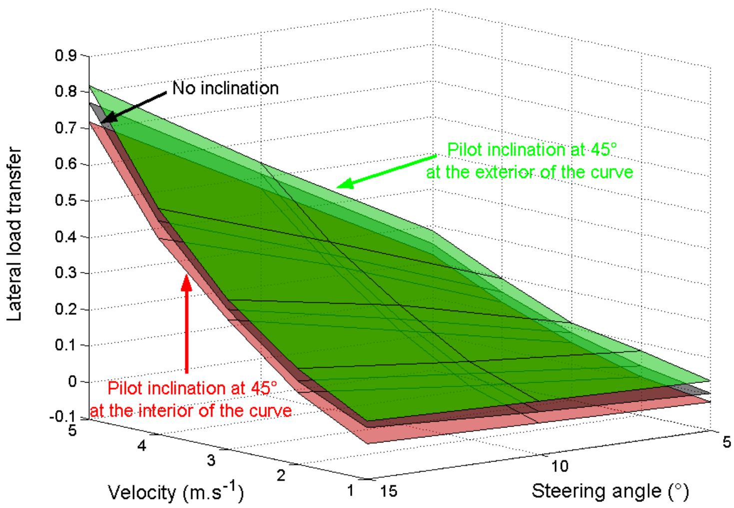


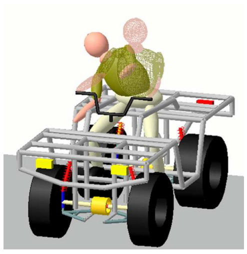
Figure 16: Influence of suspension morphology on LLT.

5.2.2 INFLUENCE OF THE PILOT ON DYNAMIC STABILITY

In this particular case, the dynamic stability of quad bikes has been examined in regard of the position of the pilot on quad bikes. So as to make this study, the dynamic modeling of the pilot has been implemented on the ISM and, in order to take into account the influence of the driver behavior, his pelvis/torso motion has been controlled with respect to the steering angle. In fact four pelvis/torso motions, depicted on Fig. 17(b) and Fig. 17(c), have been imposed on a high grip ground for several velocities and steering angle values. The first and second motions have consisted of respectively an inclination of the torso to the interior and the exterior of the bends, while the third and fourth situations correspond respectively to the pilot pelvis/torso motion at the rear and the front of the vehicle.



(a) Influence of the lateral motion on the rollover risk.



(b) Pilot lateral motion.

(c) Pilot longitudinal motion.

Figure 17: Influence of the lateral motion of the pilot on the rollover risk.

As a result, relying on ISM simulations, it can be first noticed that pelvis/torso motion in longitudinal way does not impact significantly the stability criterion. However, contrary to the longitudinal way, pilot motion in the lateral way (first and second pelvis/torso motions) has an influence on the LLT as it is depicted on Fig. 17(a). Indeed, ATV pilots have an influence on the dynamic lateral stability because the pilot motion modifies the position of the center of gravity of the system $\{pilot + ATV\}$ and the pilot weight represents more than 30% of the ATV average weight. Then, on Fig. 17(a), one can see this influence since only the pelvis/torso motion of the driver makes the quad bike more stable if the pilot bends to the interior of the curve or less stable if the pilot torso is bending to the exterior of the curve. As a consequence, in future work semi-analyticals models will be able to take into account the ATV driver behavior.

5.2.3 ANTI-ROLL BAR INFLUENCE ON LLT

Fig. 18 compares the impact of a rear anti-roll bar on LLT (stemming from ISM with and without rear anti-roll bar and no pilot) in function of the vehicle velocity and steering angle when the vehicle moves on a high grip ground. On this figure, one can checked that the rollover risk is reduced when the ATV is equipped with a rear anti-roll bar. For example at a speed of 7 $m.s^{-1}$ and a steering angle of 10°, in the presence of an anti-roll bar, LLT is reduced by 12% in relation to the LLT provided by ISM without rear anti-roll bar. However, the addition of a front anti roll bar does not noticeably modify the LLT provided by the ISM equipped with a rear anti-roll bar.

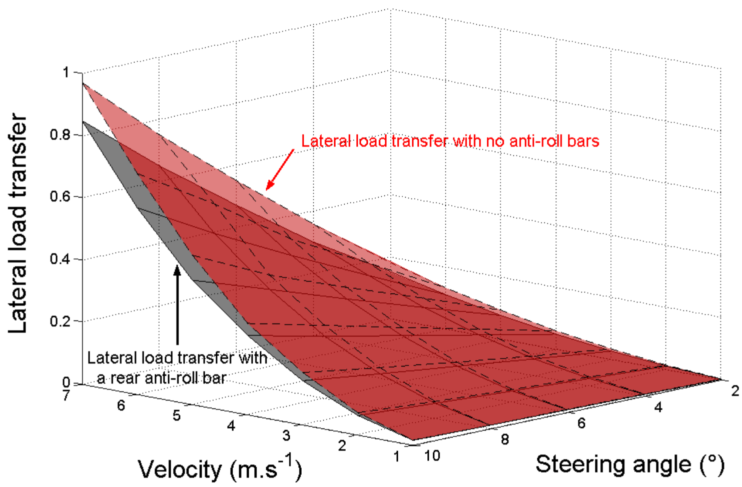


Figure 18: LLT evolution in ISM with and without anti-roll bar.

5.2.4 TRANSMISSION INFLUENCE ON LLT

As quad bikes are precisely assumed to move on high grip ground since they are on-road approved, differential mechanism are obligatory on ATVs and manufacturers have generally placed differential mechanisms on quad bikes so as to have two-wheel or four-wheel drive vehicles. Therefore, influence of transmission has been investigated in regard of the LLT and dynamic stability. As it is the case for previous studies, the steady state values of the lateral load transfer have been recorded for several velocities and steering angle inputs when the ISM is moving on a high grip ground with two-wheel or four-wheel drive transmission. Results are presented on Fig. 19.

On this figure, it is clear that four-wheel drive quad bikes are more stable than two-wheel drive ones. As a conclusion, it appears that differential mechanisms improve path tracking precision as well as lateral dynamic stability.

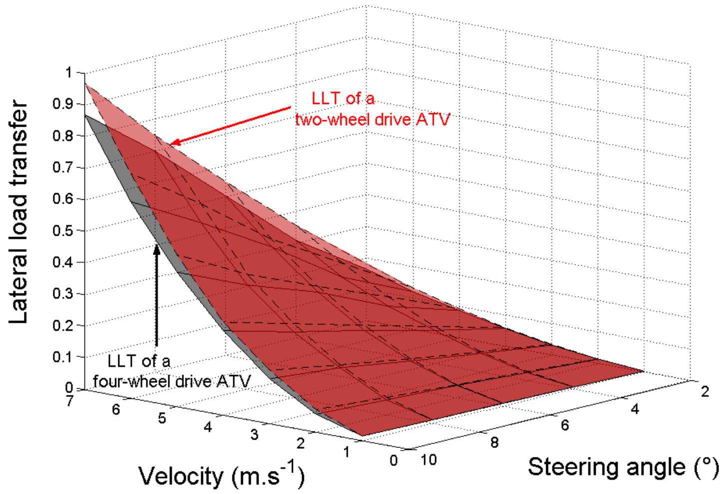


Figure 19: Influence of transmission on dynamic stability.

5.2.5 EFFECTS OF SLIDING ON DYNAMIC STABILITY

The last phenomenon studied in this paper, according to dynamic stability analysis, is the effect of sliding on the LLT. So as to make this study, steering angle and vehicle velocity have been imposed on ISM when both the vehicle moves on a flat high grip ground and on a flat slippery ground. Then, actual velocity, steering angle and lateral load transfer with sliding of the vehicle have been recorded and compared with the lateral load transfer without sliding provided by ISM when the same inputs as the velocity and steering angle measured are applied. Simulation results are presented on Fig. 20.

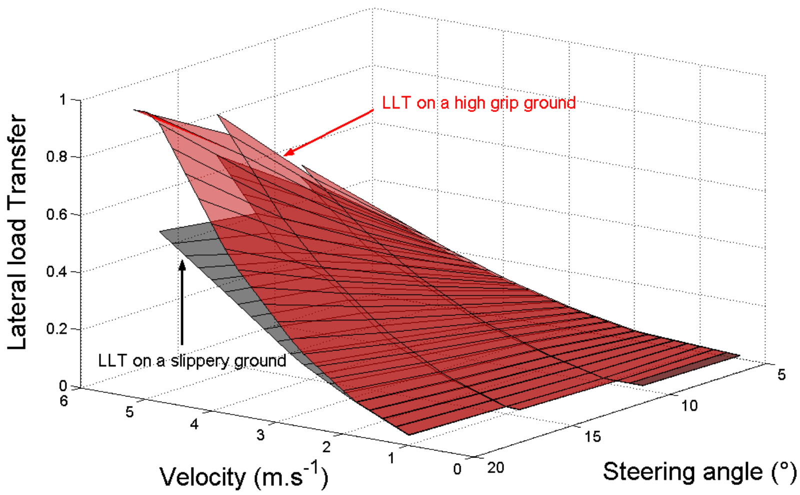


Figure 20: Effects of sliding on dynamic stability.

As the yaw rate is very sensitive to sliding effects, the lateral load transfer with sliding is very affected by this phenomenon. Indeed, on Fig. 20, the LLT with sliding effects is slighter than the LLT measured when the vehicle is moving on a high grip ground with the same inputs.

This demonstrates once more the incapacity of car-like vehicle models, developed with a rolling without sliding assumption, to describe the rollover risk of ATVs. Finally, this also demonstrates the significance of developing adaptive models such as WSM which take into account sliding effects in order to predict hazardous situations.

5.3 PARAMETER INFLUENCE SYNTHESIS

So as to conclude on the influence of parameters variations and mechanical components (except for morphology study) on the lateral dynamic stability, the stability improvement of each parameter and mechanical component has been computed relying on a nominal maxima of LLT corresponding here to the value of 0.7. The different result are expressed in Table 3.

Nominal max of LLT	0.7
Pilot stability improvement (at the exterior of the curve)	+ 9%
Pilot stability improvement (at the interior of the curve)	- 9%
Anti-roll bar stability improvement	+ 11%
Transmission stability improvement	+ 12%
Sliding stability improvement (for the sliding condition of the study)	+ 15%

Table 3: Stability improvement of each parameter and mechanical component.

According to Table 3, pilot inclination, transmission and anti-roll bars have the same contribution in the stability improvement, indeed they reduce the lateral load transfer by an average of 10% on a high grip ground. However, relying on the grip ground condition imposed during the simulation performed with ISM, it is clear that the level of sliding is the most important parameter which affects the dynamic stability of quad bikes.

Moreover, it can be noticed that the influence of mechanical parameters such as anti-roll bars, transmission, pilot inclination or suspension mechanisms does not affect the evaluation of the tire cornering stiffness, indeed these parameters have just an influence on the two quad bike parameters identify by NSM: the roll stiffness and the distance between the center of gravity and the roll center of the vehicle. Therefore, sliding (more generally grip condition) is the main parameter which affects the tire cornering stiffness evaluation and therefore the computation of the lateral load transfer on a natural and slippery ground.

6 CONCLUSION

This paper proposes an analysis of the dynamic stability of light ATVs such as quad bikes. So as to perform this study, two quad bikes have been modeled and developed with the interaction motion simulation sofware Adams. These two models are composed of various mechanical components such as differentials, anti-roll bars and suspension mechanisms used for their developments. Indeed, the first model (SISM) relies on a rear trailing arm and independent front suspensions whereas the second one (ISM) is composed of four McPherson suspensions, then both ISM and SISM represent the majority of quad bikes used in the agricultural area.

Then, contrary to on-road vehicles, quad bikes are moving on natural ground and so as to study the influence of sliding, a tire/ground contact model has been used. This model is based on an impact-force model for tire normal forces and a Coulomb-friction model for lateral and longitudinal forces exerted on the tire. According to Pacejka sliding parameters, simulations tests demonstrate the relevancy of this contact modeling to represent the sliding effects.

Moreover, it is generally assumed that the pilot has an influence on the dynamic stability of quad bikes. So as to check this statement, a pilot model composed of 18 degrees of freedom has been implemented on ISM and SISM.

In order to evaluate the stability of ATVs, a stability criterion has been chosen. This indicator is the lateral load transfer of the vehicle and is very suitable to reflect the ATV stability.

Relying on this criterion, it appears that ISM is more stable than SISM. Moreover, according to several pelvis/torso motions of the pilot, the study ends in the fact that only the pelvis/torso lateral motion has an influence on the lateral dynamic stability of the vehicle. Moreover, the vehicle is more stable if the pilot leans to the interior of the curve contrary to the exterior of the curve where logically the vehicle is less stable. Then, other tests first show how anti-roll bars increase the dynamic stability and how four-wheel drive ATVs are more stable than two-wheel drive ATVs.

Then, complex simulations tests show how the sliding modifies the dynamic stability of ATVs. Indeed, it clearly appears that quad bikes are more stable on a slippery ground contrary to a high grip ground, but this stability gain is carried out to the detriment of the path followed by the vehicle widely affected by the sliding phenomenon.

Finally, according to a comparison between parameters and mechanical components likely to affect the dynamic stability of quad bikes, it appears that anti-roll bars, transmission and pilot movement have the same influence on the dynamic stability of ATV, nevertheless, since the LLT is largely reduced on a high slippery ground, sliding appears to be the parameters affecting the most the stability criterion.

Therefore, since the aim of our work is the development of stabilizing corrective actions dedicated to off-road vehicle, new vehicle modeling has been developed. This model is characterized by a roll and a new yaw representations of the vehicle. WSM enables accurate load transfer estimation in presence of sliding, as the grip conditions are continuously estimated thanks to an on-line selection of ground contact parameters previously identified. Then, an off-road rollover risk indicator has been developed relying on WSM model and predictive control formalism. Finally, this rollover indicator has been successfully investigated in a critical situation simulated on a flat slippery ground with Adams model.

Future work will focus on performing real experiments on a quad bike in order to fit Adams models to actual vehicles by achieving parameter calibration. Secondly, as the updating of ground classes may lead to some inaccuracies when grip condition are far from any ground class, new solutions, based on a backstepping adaptive observer, are under development. The last point to be developed is to account for pilot in WSM by adpating on-line some parameters.

More generally, the development related to analytical models should open the way to the design of stabilization devices for rollover risk reduction in ATVs.

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