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Modeling and Simulation of the Motorcycle's Lowside Fall

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Abstract

The deployment of active safety systems enhancing the motorcycle stability and supporting riders in defusing critical and dangerous driving situations is a topic of major concern in the two-wheel research community. In the design and development of safety control systems, setting up an adequate model of the controlled system is a key issue since it should be able to describe adequately the motion of the vehicle in critical situations such as precarious adherence, cornering brake and acceleration, or dangerous falls. In literature, these situations are typically investigated by means of black box approaches, namely by using multibody numerical simulators in which the equations governing the vehicle dynamics are unknown. In this paper, instead, the authors propose an analytical model as alternative to black box approach for the simulation of critical and complex motorcycle's dynamics leading to falls. The model has been presented in author's earlier works, it has a minimum degree of complexity, considers the rear wheel traction/braking and takes into account the interactions between longitudinal and lateral friction forces acting on the tyres. This analytical model has allowed to investigate the lowside phenomenon and the simulation results will be presented.

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Keywords: Motorcycle stability; Active safety systems; Motorcycle modeling; Vehicle dynamics; Lowside fall

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1. Introduction

According to recent statistics [1], in Europe, motorcycle and mopeds fatalities represent about 20% of all road deaths although they constitute only 10% of all motor registered vehicles. Most motorcycle's accidents are due to falls caused by front or rear tyre slippages. In these cases, the loss of control occurs while cornering in extreme adherence conditions. When these conditions are generated by excessive acceleration or braking in curve, they may give rise to dangerous highside or lowside phenomena putting at serious risk the safety of the driver (Figure 1).

Fig. 1. The lowside fall. (Source: youtube.com)

Looking at the statistics, the development of active safety systems in support of passive systems is recommended to prevent the causes of accident. For a motorcycle, except for the Anti-lock Braking System (ABS), the deployment of active safety systems is lagging behind the one for cars. The main reason stands in the natural instability of the two-wheeled vehicle which makes the analysis of its dynamics much complex and thus it makes the developing of stability control systems more demanding [2].

The historical course that led to the understanding of the dynamics of a two-wheeled vehicle began in the late 1800's [3,4]. Over the years, analytical models have been introduced spanning from the simplest to the most complicated ones. The first category is well represented by the works of Timoschenko-Young [5] and Getz [6,7]. These models are characterized by a single mass center, simplified steer mechanism and they have no trail, nor inertias. Given their simplicity, these kinds of models are interesting from a control perspective but inadequate to properly capture the vehicle's dynamics in cornering for large roll angles or in acceleration/braking situations.

Instead, more complicated and detailed analytical models describe the dynamics of two-wheeled vehicles by considering multiple rigid bodies and multiple set of allowed motion freedoms [15]. However, these mathematical models are still characterized by unrealistic assumptions such as: decoupling of dynamics, the vertical dynamic (suspension motions) is neglected as well as the longitudinal dynamic and the motorcycle is considered at constant speed. Moreover, the tyre forces are linearized and no tyres slips are involved. Despite all these assumptions, due to the multibody structure, these models are still too complex for the design of model-based control systems.

To overcome the difficulties in deriving a detailed analytical model, currently, multibody software tools represent the state of the art for a complete modeling of motorcycle's dynamics [8,9], including complex conditions such as the falls [10,11]. They offer a considerable number of rigid bodies, high degrees of freedom (dofs) and accurate results. For example, a model with 6 bodies and 11 dofs is presented in [8]. On the other hand, while these tools are suitable for simulation purposes, they are not for the synthesis of control systems due to their black box nature, i.e. the equations governing the simulations are not provided.

In the present paper, the authors propose an alternative approach to multibody software for the simulation of critical motorcycle falls, which so far have never been analyzed in an analytical way. For this purpose, the authors consider a scalable analytical model with a minimum level of complexity that may be used for a model-based synthesis of a stability controller. The model consists of 2 rigid bodies, 7 dofs and 2 fully actuated inputs, the steering torque and the rear traction [12,13]. Differently from the analytical models mentioned above, the authors' model removes the assumptions of constant velocity in straight running and in cornering, it introduces the effect of the rear traction, the wheel slippages and the interactions between the tyre-road friction forces are described by means of the Magic Formula. All these features allow to simulate most of all motorcycle conditions and this paper focus on the results obtained by simulating the lowside fall. The paper is organized as follows: a brief description of the model is outlined in section 2. Section 3 reassumes the tyre model; section 4 provides some evaluations about the sensors needed to measure the major motorcycle dynamic parameters. Section 5 introduces the lowside fall, the simulations results are presented in section 6 and section 7 concludes the paper.

2. Motorcycle model outline

The motorcycle mathematical model employed in the simulations is shown in Figure 2. The vehicle consists of two rigid bodies with masses lumped in G_r and G_f. They are joined at the steering axis with the front body being free

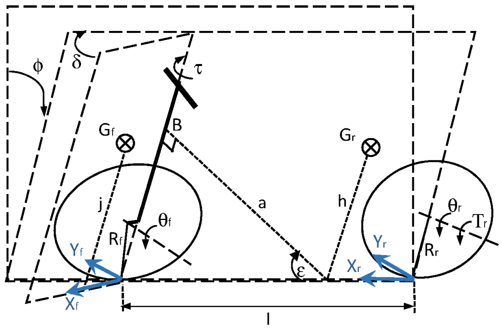

Fig. 2. Geometric parameters of the proposed model.

to rotate relative to the rear one. The rear body includes the seat, the rear forks, the rear wheel with radius R_r and the rider rigidly attached. The front body consists of the front wheel with radius R_f, forks, trail and handlebars while the steering mechanism is constrained by a linear steering damper. The vehicle is also characterized by the wheelbase l, the steering head angle □ and 7 dofs: the longitudinal and lateral velocity of the motorcycle $\dot{x}$ and $\dot{y}$ respectively, the yaw angle ψ, the roll angle ϕ, the rotation δ around the steer axis and the wheel's angular velocities $\dot{\theta}_r$ and $\dot{\theta}_f$. Two external actuations, the steering torque τ and the engine torque T_r are considered.

The vertical dynamic relating to the suspensions flexibilities is neglected since the lowside fall phenomenon can be generated whether the vertical dynamic is considered or not. Using the energy method of Lagrange, the equations of motion have been derived in [12,13] and given their length are not here reported. The model has been tested in different scenarios which have highlighted the dynamic behaviour of the vehicle in acceleration, deceleration and grip conditions. In order to capture the lowside fall in a bend, compared to previous works, this analytical model has been further enhanced considering a realistic and appropriate tyre model (Magic Formula) which fully capture the nonlinear behavior during slippage [15]. The longitudinal and lateral friction forces X_r, X_f, Y_r and Y_f acting on the wheels are described in the next section.

3. The tyres forces

A brief introduction of the friction forces and their interactions are provided in this section. In a model-based design of safety control system, it is essential for the model to properly describe the forces that arise between the tyre and the ground since these are necessary to change the motorcycle's speed, its direction and strongly affect the stability of the vehicle. In literature, an extensive research has been done in tyre modeling. Substantially these models are divided into two large families: on one side, there are the physical models based on complex formulations and mathematical equations that describe in detail the tyre structure and the interactions with the ground. On the other side, there are the empirical models based on transcendental formulae having no particular physical meaning, but they well fit experimental curves and measured data. In the middle, there are those models that describe the tyre forces and find the right compromise between accuracy and analytical complexity [14]. Since the author's model does not consider the vertical dynamics, the friction forces between the tyres and the road have the following form [15]:

$$X_i = F_z \mu_{xi} \quad Y_i = F_z \mu_{yi} \tag{1}$$

where the subscripts $i = \{r, f\}$ indicate the rear or the front tyre, x, y the direction along the longitudinal and lateral axes of the vehicle respectively, μ_x, μ_y are the friction coefficients and F_z is the vertical load, here assumed constant for both the wheels. To consider the tyre slippages, the friction coefficients $\mu_{x,y}$ have been modeled with the Magic Formula, recalled in the next section.

3.1. The Magic Formula

Introduced initially in 1987, the Magic Formula is the empirical reference model for real-time applications since it is reasonably accurate and fast to be computed.

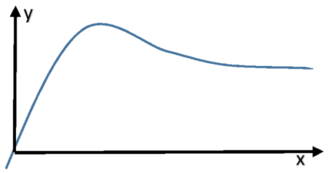

Fig. 3. The Magic Formula.

Based on this model, the general expression of the coefficients $\mu_{x,y}$ is given by:

$$\mu_{x,y} = D_{x,y}\, sin\left[C_{x,y} \tan^{-1}\left\{B_{x,y}k - E_{x,y}(B_{x,y}k - \tan^{-1}(B_{x,y}k))\right\}\right] \quad (2)$$

where the input k is the relative slip (longitudinal λ or lateral α) and the parameters $B_{x,y}, C_{x,y}, D_{x,y}, E_{x,y}$ represent the coefficients defining the curve shape, as shown in Figure 3. According the SAE standard [16], the longitudinal slip is given by:

$$\lambda_i = -\frac{V_x - V_R}{V_x}, \quad (3)$$

in which V_x and V_R are the forward and the rolling velocity of the motorcycle wheel respectively. In the same way, the lateral slip is defined as:

$$\alpha_i = tan^{-1}\left(\frac{V_y}{V_x}\right) \quad (4)$$

where V_x and V_y are the longitudinal and lateral velocity of the wheel respectively. Equation (2) refers to a car tyre, whose roll angle (camber angle) remains relatively small. On the contrary, motorcycle can lean up to 45°-50°. For this reason, it is preferable to define an equivalent lateral slip as:

$$\alpha_{eqi} = \alpha_i + \frac{k_\phi}{k_\alpha}\phi_i, \quad (5)$$

where k_α, k_ϕ are the cornering and the camber stiffness respectively and ϕ is the wheel camber angle. Replacing expressions (5) and (3) in (2), the longitudinal and lateral forces (1) become:

$$\begin{aligned} X_i' &= F_z\, D_x\, sin[C_x \tan^{-1}\{B_x\lambda_i - E_x(B_x\lambda_i - \tan^{-1}(B_x\lambda_i))\}] \\ Y_i' &= F_z\, D_y\, sin\left[C_y \tan^{-1}\left\{B_y\alpha_{eqi} - E_y\left(B_y\alpha_{eqi} - \tan^{-1}\left(B_y\alpha_{eqi}\right)\right)\right\}\right]. \end{aligned} \quad (6)$$

Finally, to better describe the behaviour of the friction forces in cornering condition, scaling factors can be introduced as:

$$\begin{cases} \sigma_{xi} = \frac{\lambda_i}{1+\lambda_i} \\ \sigma_{yi} = \frac{tan\alpha_{eqi}}{1+\lambda_i} \end{cases} \quad (7)$$

whose magnitude is given by:

$$\sigma_i = \sqrt{\sigma_{xi}^2 + \sigma_{yi}^2}. \tag{8}$$

The main disadvantage of equations (6) is that they do not describe the friction forces in transient state, however, a realistic situation implies these forces arise after a lagging time. To consider this delay, the following relaxation equations must be introduced:

$$\begin{aligned} \frac{\zeta_x}{V_x}\dot{X}_i + X_i &= \frac{\sigma_x}{\sigma} X_i' \\ \frac{\zeta_y}{V_x}\dot{Y}_i + Y_i &= \frac{\sigma_y}{\sigma} Y_i' \end{aligned} \tag{9}$$

where ζ_x, ζ_y are the tyre relaxation lengths, X_i', Y_i' have been computed in (6), σ_x, σ_y and σ have been calculated in (7) and (8). The expressions of X_i, Y_i determined in (9) are the friction forces of the model depicted in Figure 2.

4. Stability control implementation issue

In order to analyze the usefulness of the proposed model in the implementation of a stability control system, some considerations have been done in the follows.

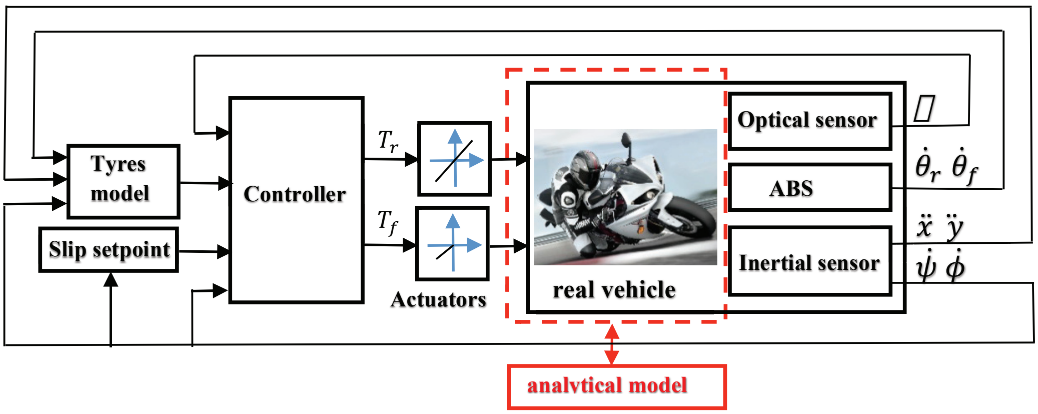

Fig. 4. The sensors deployed in safety control system.

To implement a stability control system, a selection of the most suitable control and controlled variables is required. In Figure 4 the implementation of a safety control system with sensors deployment is showed. Referring to Figure 4, the inputs that can be manipulated are the torques at the wheels (the traction and braking torque at the rear wheel T_r and the braking torque at the front wheel T_f) and the steering angle δ (by the rider). To enhance the stability of a motorcycle the yaw rate $\dot{\psi}$ and the roll rate $\dot{\phi}$ of the vehicle are the main variables that have to be measured, generally this can be achieved via MEMS gyroscopes. In critical vehicle's conditions the wheel's loss of adherence play an important role and then a measure of the wheel's velocity $\dot{\theta}_r$ and $\dot{\theta}_f$ is required. When the vehicle lateral slip changes, a lateral tyre force is created with a time lag, in this case a lateral accelerometer which measures the lateral acceleration of the vehicle $\ddot{y}$ makes us the information on the lateral forces acting on the tyre. Moreover, the rider's intention can be obtained by means of longitudinal accelerometer which measures the longitudinal acceleration $\ddot{x}$ of the vehicle. Summarizing, the following measurements are usually needed for the implementation of a stability controller:

- yaw rate $\dot{\psi}$, roll rate $\dot{\phi}$, longitudinal $\ddot{x}$ and lateral $\ddot{y}$ accelerations measured by an inertial sensor, they are usually provided by the Inertial Measurement Unit (IMU) composed by three 1-axis gyroscope and 3-axis MEMS accelerometer;
- steering angle δ measured by an optical sensor;

- rotational velocity for each tyre $\dot{\theta}_f$ and $\dot{\theta}_r$, normally provided by ABS systems, if this system is not available the vehicle has to be equipped with two encoders for the measuring of the rotational wheel's speed.

Finally, by taking into account experimental studies which have shown that the closed loop bandwidth at -3dB observed in a real motorcycle dynamics is in the range of 3-4 Hz [17], one may argue that this is the range of restitution of transient accelerations in emergency braking, sudden acceleration. As a matter of fact, a sampling cycle of more than 100 times per second is required for the sensor to measure the status of the motorcycle's driving dynamics. However, the problem of the estimation of the attitude of a motorcycle with inertial signals is not trivial mainly because of the very noisy environment in which the data are collected and the errors that affects the signals provided by the MEMS sensors.

5. Brief introduction to the lowside fall

In this section, a quick insight into a typical lowside dynamic caused by excessive rear brake is given. For a detailed explanation of the phenomenon refer to [11]. Let's consider a motorcycle entering a curve with excessive velocity. When the vehicle is in dynamic equilibrium, the lateral friction force acting on the tyres provides the values required to balance the moment generated by the vertical load.

Fig. 5. (a) the initial stage of lowside; (b) the lateral fall. (Source: youtube.com)

As soon as the rider applies hard on the rear brake to slow the vehicle down in curve and keep the trajectory (Figure 5a), a longitudinal slip might arise. Typically, the presence of longitudinal slippage in curve also affects the vehicle lateral behaviour in a way that the equilibrium is reached with a greater lateral slip angle than the one needed in normal conditions. In other words, the lateral force provided to the tyres is reduced in presence of longitudinal force and longitudinal slip hence in this condition the vehicle tends to increase the lateral slippage. If the rider does not release the brake the rear wheel may lock up and the rear slippage continues till the roll angle is so large that any attempt of the rider to regain attitude is useless. The bike is now out of control and it ends up to fall laterally and slide out along with the rider (Figure 5b).

6. Lowside simulation and results

In this subsection, we focus on the scenario representing the lowside fall caused by too much rear braking.

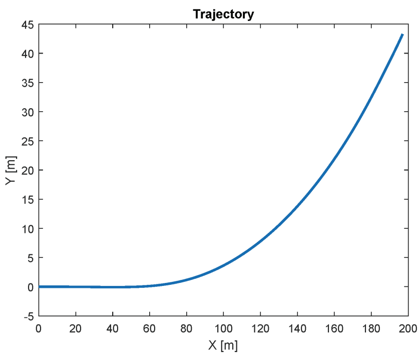

Fig. 6. The trajectory run during the simulation.

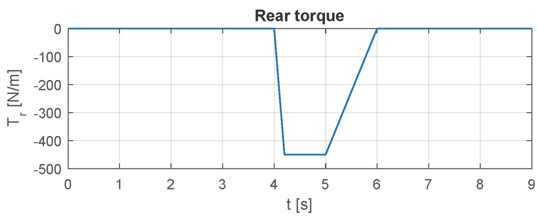

Fig. 7. The brake applied to rear wheel.

Figure 6 shows the trajectory travelled by the motorcycle and in Figure 7 the input torque simulating the rear brake applied by a virtual rider is shown.

Starting from about t = 2 seconds, the vehicle engages a curve at 45 m/s (164 km/h) with roll angle of about 40 degrees. During the early stage of the cornering and before braking, the vehicle is in dynamic balancing, the wheels are in complete adherence condition hence the rear lateral forces Y_r and Y_f grows (Figure 8) as the roll angle ϕ increases (Figure 9). For a description of the balancing condition for a vehicle in curve, the reader can refer to [15]. In the time window 4-5 seconds, the rider applies hard on the rear brake to slow the vehicle down and the wheel's speeds $\dot{\theta}_r$ and $\dot{\theta}_f$ begin to decrease although with different trends (Figure 10) as expected.

Precisely, in the same time window, the rear braking torque starts growing rapidly hence the longitudinal friction force X_r (the braking friction force) acting on the rear wheel increases quickly as well (Figure 8) accordingly to the Magic Formula wherein the increasing longitudinal slip λ is involved. The rear wheel start losing adherence hence the angular velocity of the rear wheel $\dot{\theta}_r$ decreases much faster than the front wheel $\dot{\theta}_f$ as expected (Figure 10). The front wheel is always in pure rolling so it does not experience any slippage and the front lateral force Y_f increases accordingly during the cornering (Figure 8-9). Figure 8 shows the lateral force Y_r acting on the rear wheel. At about 4.2 seconds this force starts decreasing while X_r keeps on growing. That means the rear wheel is sliding laterally because the force Y_r does not suffice to maintain the vehicle in balancing. This effect never stops and the roll angle ϕ quickly increases as well as the yaw angle ψ. The vehicle goes fast out of control and the roll angle ϕ reaches quickly 90° at 4.6 seconds. The vehicle capsizes and slides off of road dragging down the rider in its fall (Figure 5b).

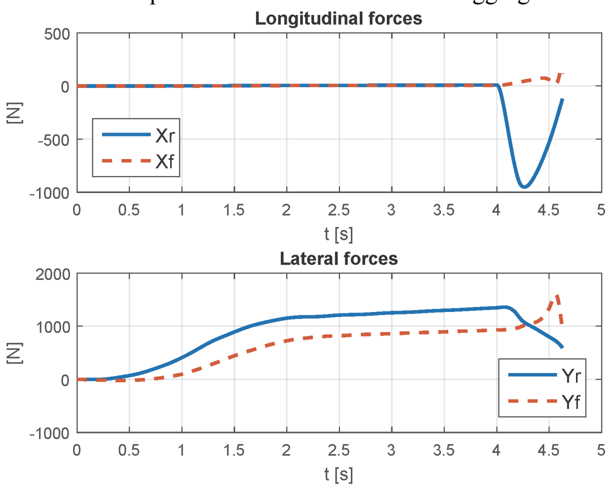

Fig. 8. The friction forces.

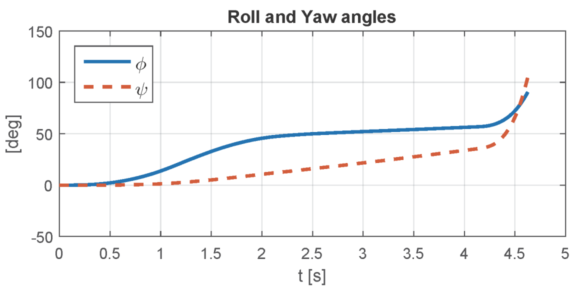

Fig. 9. The roll and yaw angles.

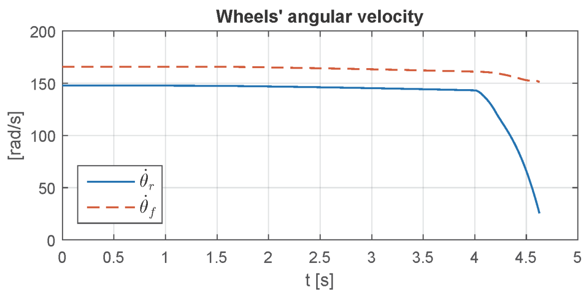

Fig. 10. The rear and front angular velocities.

7. Conclusion

In this paper, the lowside fall has been simulated by means an analytical model. General topics related to active safety systems for motorcycles and the features of the models that are required for the analysis of critical vehicle's dynamics have been introduced. A motorcycle's analytical model set up by the authors in their earlier works has been outlined and a brief description of the tyre model, along with the expressions of the friction forces acting on the tyres, have been presented. This model has been used to simulate the lowside fall excited by high turning speed and excessive rear braking and the results of the simulations have been discussed. The model is able to capture the major characteristics of the lowside dynamic of a two-wheeled vehicle. Further works will be addressed to investigate the motorcycle dynamic in different critical conditions, improve the model by implementing other features like the vertical motion and a possible use of the model for stability control purposes will be discussed.

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