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## Two wheels electric vehicle modelling : Parameters sensitivity analysis.

Y. Bello, T. Azib, *Member, IEEE*, C. Larouci, *Senior Member, IEEE*, , M. Boukhnifer, *Senior Member, IEEE*, N. Rizoug, *Member, IEEE*, D.A. Patino, *Member, IEEE*, F. Ruiz, *Senior Member, IEEE*.

**Abstract**—Nowadays, electric vehicles represent one of the most significant chances to reduce the pollution production rate. Unfortunately, in electric motors, the efficiency decreases by the relationship between speed proposed by the driver and the torque required by the vehicle. Those parameters can be estimated in order to make an efficiency optimization based on present and future road/weather conditions. Regrettably, this kind of control (optimal control) requires a model with low compilation time. Since bicycle motorcycle has nonlinearities, in this article, a state reduced linear dynamic model able to reproduce the behavior of a TWEV by more than one minute will be proposed. The model is oriented to an optimal controller in energetic field, for this reason, the most significant states are longitudinal speed to be used with the input torque to calculate the efficiency of the electric motor and the Yaw angle to creates constraints over the trajectory that has to be covered. After the model is proposed, its accuracy is tested by a comparison with a numerical iteration software. Finally, a sensitivity test is made in order to determine the behavior of the error according to the friction coefficients of the rear and front pneumatics.

Electric Vehicle, Dynamic Model, sensitivity test, Autonomy increasing.

### I. INTRODUCTION

Even though the electric vehicles (EVs) have increased its presence in the word, the percentage of EVs in contrast to the gas vehicles still being relatively low. As an example, they represent 5% in the Latin-American market [1]. The situation for two wheels electric vehicles like bicycles or motorcycles TWEV is not very different. This situation can be caused mainly for three reasons:

- A lack of an appropriated infrastructure to charge this kind of vehicles.
- The prices of electric vehicles.
- The performances offered by this technology (autonomy, life cycle, etc...).

Two of three reasons are related to the trust of the rider to complete the desired distance. For this reason, this article focuses its effort to build a tool helping to improve the capacity for estimating the autonomy available in a TWEV using relevant information which makes efficiency vary. The objective to build this model is to make an optimization over the driving profile based on the information of the trip required by the user and the information of the vehicle

properties. To achieve that goal, the first step is to reproduce the mechanical behavior of the vehicle. This paper presents the process to find the mechanical model divided into two parts:

- A first part, the theoretical modelization of the longitudinal and lateral movement.
- A second part related with the sensibility test in order to reduce the linear model error based on data from environmental physics simulator software.

The second part of the process is required to ensure the limitations of a linear model are enough small to be considered a representation of the non-linear model of the pneumatic dynamic. Since the linear model is designed for a close loop optimization control, prediction horizon and maximum error acceptable have to be chosen by the researcher. In this case, the prediction horizon will be 60 seconds due to the range of speed of an urban speed profile and maximum acceptable error will be 10% due to the error grow rate. However, both values can be optimized in the future according to optimal controller performance. The mechanical model formulation is important because even when the same bicycle model is used on cars, in TWEV there are new aspects like roll angle or the shape of the pneumatic which need to be considered by its relevance in their driving profile. Also, papers like [2] [3] mention the trajectory conditions, weather and driving profile as the main factors involved in the autonomy uncertainty.

The second step will be reproduced the electrical behavior, the losses of motor, inverter, and battery have to be modeled. In this paper, just the first step is discussed. This prediction will ensure that the state of charge(SoC) present in the battery is enough to complete the trajectory required. Indeed, if energy prediction gives the trust to the user to use the EV even with lower levels of SoC, the user has the perception of an autonomy increasing, because the charging will be done just when it is needed.

According to the National Highway Transportation Safety Administration (NHTSA) of the United States of North America, the main reason for fatal crash accidents or rollover accidents in that country are related to lane changing [4]. This is the reason why the Yaw angle is taken into account by the dynamic model.

There are three kinds of lateral systems [5]: Lane departure warning, lane keeping system and yaw stability control system. The first one alerts to the driver to lane departures. Good examples of this system are AutoVue, AssistWare or SafeTrac. The second one is able to detect undesired lane departure and control the steering angle of the vehicle

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to keep the line. It is a steering control in parallel to the driver steering control. Examples of this technologies are lane-keeping-system (LKS) by Nissan or lane-departure-avoidance (LDA) by the same company [4][6]. The last system is in charge to prevent spinning and drifting events. The system is able to estimate road/weather conditions to find a minimum risk curvature possible for speed and friction available. In this case, the system is able to change steering and speed reference value in order to improve the safety of the driver.

The paper will be developed in the following order: The dynamics required will be defined in Section II, the equations which describe the dynamics chosen will be explained in Sections III and IV. The verification of the model proposed is made with BIKESIM software as an environmental physics simulator software in Section V and a model sensitivity test to determine the behavior of the model in face of the friction coefficient variation is presented in VI. After the sensitivity test, in Section VII the new friction coefficient obtained will be used to repeat the model verification with BIKESIM and the results will be discussed. Finally, conclusions and future work is presented in Section VIII

## II. DYNAMICS REQUIRED.

Although motorcycles are composed of a variety of mechanical parts, from a kinematic point of view, a motorcycle can be defined as simply a multi-body platform composed of four rigid bodies:

- The rear assembly.
- The front assembly.
- The front wheel.
- The rear wheel.

This motorcycle definition is valid under the following assumptions:

- The suspension is enough rigid, in consequence, the  $\theta$  dynamic along the Y-axis (Figure 1) caused by load exchange between the front and rear assembly can be ignored.
- A perfect coupling between driver and motorcycle bodies. The momentum caused by the rider body is not considered.
- An homogenous pressure distribution of pneumatic. This assumption lets to conclude that the longitudinal and lateral forces are distributed homogeneously along the pneumatic too.

The four rigid bodies are connected by 3 revolute joints, as each revolute joint remove 5 degrees of freedom (DOF), 3 revolute joints inhibit 15 DOF of the spatial freedom of the platform. Taking into account the three kinds of wheel motions: roll, spin and forward, each wheel point of contact remove 3 DOF. Since each body has 6 DOF, from the 24 DOF of the 4 main bodies, 21 DOF are removed by the movement restrictions ( $15+(2*3)$ ). It is a total of 3 spatial DOF [4]. Those are forward, roll and steering motion of the motorcycle. This hypothesis is valid if slippage is ignored. In real conditions, longitudinal and lateral slippage cause

the combination of DOF mentioned before becomes in the 6 DOF shown in Figures 1 and 2. Those DOF are:

- Forward motion, it is caused when there is slippage on front and rear wheel by braking events or in the rear one by acceleration events. This dynamic will be referenced as longitudinal dynamic and it is parallel to X wheels axis.
- Lateral motion. This dynamic will be referenced as Lateral dynamic and it is parallel to Y wheels axis.
- Roll motion. It is the moment generated around X axis formed by the two points of contact of both wheels. This is an unstable dynamic relevant for trajectory tracking. As the model is oriented to energetic behavior this dynamic is ignored but the roll angle is used as a parametric input.
- Steering motion. It is the moment generated around the vertical axis of the handlebar. This dynamic wont be considered due to the energetic model orientations. Instead, the steering angle effect will be considered as a parametric input too.
- Yaw motion. It is the moment generated around the Z vehicle axis. Even when the dynamic model is energetic control oriented, this dynamic is useful to make security constraints over the longitudinal feasible profile in future works. For this reason, it will be contemplated.
- Theta motion. it is the dynamic caused by the load exchange between rear and front assembly. This dynamic is ignored.

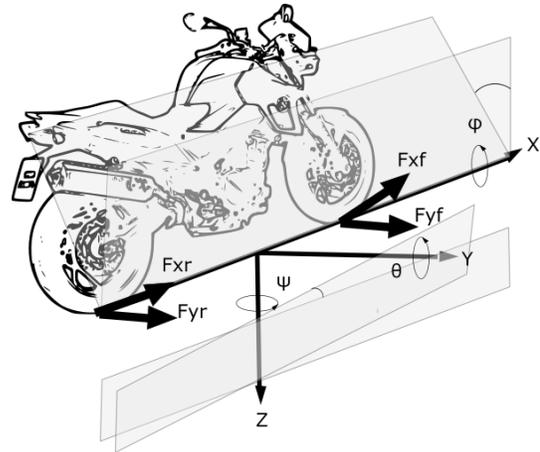


Fig. 1: Degrees of freedom of motorcycle.

In conclusion, the dynamic model required is composed by following states, inputs, and parametric inputs:

$$\begin{aligned} x &= [\dot{x} \quad \dot{y} \quad \psi] \\ u &= [T_r \quad T_f] \\ u_p &= [\phi \quad \delta] \end{aligned} \quad (1)$$

Where  $\dot{x}$ ,  $\dot{y}$ ,  $\psi$ ,  $T_r$ ,  $T_f$  are longitudinal/lateral speeds, the angle between local X axis and global X axis, and rear/front wheel torques.

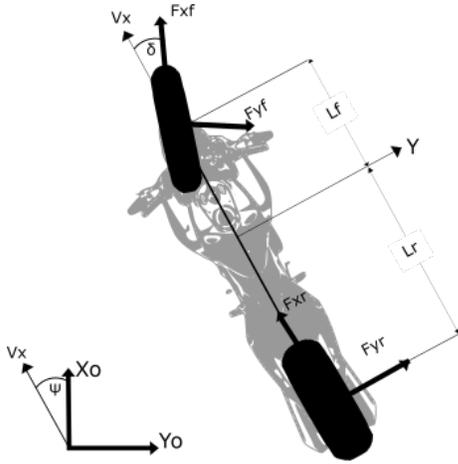


Fig. 2: Top view of degrees of freedom of motorcycle.

### III. LONGITUDINAL MODEL.

This paper considers an urban speed profile, for this reason, slippage over a longitudinal axis can be omitted [7]. As a result of this hypothesis, the angular speed of each wheel can be omitted as a new state and the longitudinal movement wont depend on the relationship of longitudinal speed and its equivalent angular speed[6]. It will be the result of the summation of forces along the X-axis. In

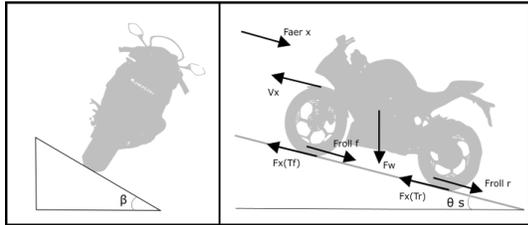


Fig. 3: Forces involved in longitudinal dynamics.

Figure 3, the road considerations and the forces involved are reported. Applying Newtons second law for motion along the X axis and contemplating two components involved in total longitudinal acceleration (longitudinal acceleration due to motion along the X-axis  $\ddot{x}$  and the centripetal acceleration  $\dot{y}\dot{\psi}$ ), the Longitudinal dynamic is described as:

$$M(\ddot{x} - \dot{y}\dot{\psi}) = \frac{T_f}{R_{wf}} + F_{roll} + F_{aer,x} + F_w \sin(\theta_s) \sin(\beta) \quad (2)$$

Where  $R_{wf}$ ,  $F_{roll}$ ,  $F_{aer,x}$ ,  $F_w$ ,  $\theta_s$ ,  $\beta$  are the effective radius of rear wheel, roll resistance, aerodynamic force associated to X-axis, weight, slope angle and bank angle. The negative forces related to energy losses will be discussed deeply in the next sub-section.

#### A. Forces involved in energy losses.

According to [5] [8] the most relevant opposite forces to the longitudinal movement in a vehicle are:

- Aerodynamic force.
- Rolling resistance force.
- Slope of the road.

a) *Aerodynamic force*:: It can be expressed as:

$$F_{aereo} = \frac{-1}{2} \rho C_d A_f (V_{wind} * \cos(\alpha_{air}))^2 \quad (3)$$

Where  $\rho$ ,  $C_d$ ,  $A_f$ ,  $\dot{x}$ ,  $V_{wind}$ ,  $\alpha_{air}$  are the air density, drag coefficient, front area, vehicle speed, wind speed and the angle between air and vehicle direction. In average conditions, aerodynamic force can cause between 20% and 40% of energy expenses in a motorcycle depending on the motorcycle chassis, the speed of air and the angle between vehicle speed and airspeed [4]. The aerodynamic force has to be separated by its influence in X-axis and Y-axis. Those magnitudes will depend on the angle between wind direction and vehicle speed vector direction.

b) *Rolling Resistance Force*: : It can be expressed as:

$$F_{roll} = -(\mu_0 + \mu_1 \dot{x}^2) F_z \cos(\theta_s) \quad (4)$$

Where  $\mu_0$ ,  $\mu_1$  are two friction coefficients of the road,  $F_z$  is the normal force of the vehicle and  $\theta_s$  is the slope of the road. The friction coefficients of the road depend on the state of the road. For example, a new asphalt is [0.01, 0.008] and the frozen asphalt is [0.001, 0.00082]. This variation of coefficient can cause an error of 15% of the autonomy estimation [4]. The motorcycle is a multi-body platform then, there is a load exchange between the front frame and rear frame caused by acceleration and brakes profile. However, a peak normal force is considered to calculate the rolling resistance force due to the frequencies of the normal force transference is faster than energetic dynamics.

c) *Slope of the road*: : When the road presents a slope by a medium or long distance, the load exchange cannot be ignored. In this case, the load exchange has not a dynamic but it creates a weight component in X and Y axis which affects the forward and lateral movement dynamic. The equation which describes this effect is:

$$F_w = mg \quad (5)$$

Where  $m$ ,  $g$  the mass and gravity.

Papers like [2] have modeled the impact of the external parameters to reduce the energy uncertainty. The result of works like this one is an electric vehicle assistant able to improve the usability of an electric vehicle. Even when the electric vehicle assistant (EVA) is able to reduce the energy uncertainty about a 34%, those results do not use mechanic and electric limitations of motor, driver, and battery. For this reason, it is expected to improve those results using those data.

In conclusion, the external parameters cause an autonomy uncertainty able to be considered a research problem. The last sections have developed most of the tools to deal with it, but at this moment, this information is not added to the power chain efficiency information to makes the electric motor works in the most effective energy region.

### IV. LATERAL MODEL.

Applying Newtons second law for motion along the Y-axis and contemplating the two components involved in total

lateral acceleration (lateral acceleration due to motion along the Y-axis  $\ddot{y}$  and the centripetal acceleration  $\dot{x}\dot{\psi}$ ), the Lateral dynamic is described as:

$$M(\ddot{y} + \dot{x}\dot{\psi}) = F_{y_f} + F_{y_r} + F_{aer_y} - F_w \cos(\theta_s) \sin(\beta) \quad (6)$$

$$I_{zz}\ddot{\psi} = M_z = L_f F_{y_f} - L_r F_{y_r} \quad (7)$$

Where,  $F_{y_f}$ ,  $F_{y_r}$ ,  $F_{aer_y}$ ,  $F_w$ ,  $M_z$ ,  $I_{zz}$  are the rear and front lateral force, aerodynamic force associated to Y axis, lateral effect of weight, momentum along Z axis and inertial moment around Z axis. The behavior of lateral forces will be discussed in next sub-section.

#### A. Lateral Forces.

The lateral forces are known as slip forces because its behavior is strongly related with slip angle. The slip angle is the angle between the vehicle gravity center speed and the X-axis of the vehicle. The expression of slip angle is [9]:

$$\alpha_f = -\arctan^{-1}\left(\frac{\dot{y} + L_f \dot{\psi}}{\dot{x}}\right) + \delta \cos(\epsilon) \quad (8)$$

$$\alpha_r = -\arctan^{-1}\left(\frac{\dot{y} - L_r \dot{\psi}}{\dot{x}}\right) \quad (9)$$

Lateral forces in motorcycle are considerably affected by camber angle too. Camber angle is the angle between the ground surface and the XY plane of each wheel. Their mathematical expression is:

$$\gamma_f = \sin(\epsilon) \cos(\phi) \sin(\delta) + \sin(\phi) \cos(\delta) \quad (10)$$

$$\gamma_r = \phi \quad (11)$$

Lateral forces are:

$$F_{y_f} = C_{f1} \alpha_f + C_{f2} \gamma_f \quad (12)$$

$$F_{y_r} = C_{r1} \alpha_r + C_{r2} \gamma_r \quad (13)$$

If the Equations 8 and 9 are used to make lateral forces calculus with small angle approximation, Equations 12 and 13 becomes in:

$$F_{y_f} = C_{f1} \left( \frac{-\dot{y} - L_f \dot{\psi}}{\dot{x}} + \delta \cos(\epsilon) \right) + C_{f2} \gamma_f \quad (14)$$

$$F_{y_r} = C_{r1} \left( -\frac{\dot{y} + L_r \dot{\psi}}{\dot{x}} \right) + C_{r2} \gamma_r \quad (15)$$

The relevant equations for lateral dynamics " $F_{y_f} + F_{y_r}$ " and " $L_f F_{y_f} + L_r F_{y_r}$ " can be reorganized to separate the effect of  $\dot{\psi}$  rate, lateral and longitudinal speed ratio and steering angle.

$$\begin{aligned} F_{y_f} + F_{y_r} &= \left( -\frac{L_f C_{f1}}{\dot{x}} + \frac{L_r C_{r1}}{\dot{x}} \right) \dot{\psi} - (C_{f1} + C_{r1}) \frac{\dot{y}}{\dot{x}} \\ &+ C_{r1} \delta \cos(\epsilon) + (C_{f2} \gamma_f + C_{r2} \gamma_r) \\ L_f F_{y_f} - L_r F_{y_r} &= \left( -\frac{L_f^2 C_{f1}}{\dot{x}} - \frac{L_r^2 C_{r1}}{\dot{x}} \right) \dot{\psi} - (L_f C_{f1} \\ &- L_r C_{r1}) \frac{\dot{y}}{\dot{x}} + L_r C_{r1} \delta \cos(\epsilon) + (L_f C_{f2} \gamma_f + L_r C_{r2} \gamma_r) \end{aligned} \quad (16)$$

In a short way to write it:

$$F_{y_f} + F_{y_r} = C_r \dot{\psi} + C_\beta \frac{\dot{y}}{\dot{x}} + C_\delta \delta \cos(\epsilon) + (C_{f2} \gamma_f + C_{r2} \gamma_r) \quad (17)$$

$$L_f F_{y_f} + L_r F_{y_r} = D_r \dot{\psi} + D_\beta \frac{\dot{y}}{\dot{x}} + D_\delta \delta \cos(\epsilon) + (D_{gf} \gamma_f + D_{gr} \gamma_r) \quad (18)$$

Finally, lateral dynamics are:

$$\begin{aligned} M(\ddot{y} - \dot{x}\dot{\psi}) &= C_r \dot{\psi} + C_\beta \frac{\dot{y}}{\dot{x}} + C_\delta \delta \cos(\epsilon) \\ &+ (C_{f2} \gamma_f + C_{r2} \gamma_r) + F_{aer_y} - F_w \cos(\theta_s) \sin(\beta) \end{aligned} \quad (19)$$

$$I_{zz}\ddot{\psi} = D_r \dot{\psi} + D_\beta \frac{\dot{y}}{\dot{x}} + D_\delta \delta \cos(\epsilon) + (D_{gf} \gamma_f + D_{gr} \gamma_r) \quad (20)$$

#### V. DYNAMIC MODEL VERIFICATION.

Complete model of motorcycle considered is:

$$M(\ddot{x} - \dot{y}\dot{\psi}) = \frac{T_f}{R_{wf}} + F_{roll} + F_{aer_x} + F_w \sin(\theta_s) \sin(\beta) \quad (21)$$

$$\begin{aligned} M(\ddot{y} - \dot{x}\dot{\psi}) &= C_r \dot{\psi} + C_\beta \frac{\dot{y}}{\dot{x}} + C_\delta \delta \cos(\epsilon) \\ &+ (C_{f2} \gamma_f + C_{r2} \gamma_r) + F_{aer_y} - F_w \cos(\theta_s) \sin(\beta) \end{aligned} \quad (22)$$

$$I_{zz}\ddot{\psi} = D_r \dot{\psi} + D_\beta \frac{\dot{y}}{\dot{x}} + D_\delta \delta \cos(\epsilon) + (D_{gf} \gamma_f + D_{gr} \gamma_r) \quad (23)$$

In order to verify the behavior of the model, a virtual test scenario was build in BIKESIM software. The road designed is a "Figure 8" with two curves of a radius of 50m, a bank angle of 30% and an intersection with an elevation of 2m and a slope angle of 5. Two kinds of the test will be made: constant speed values from 5 to 20 [m/s] to notice yaw response at different speed values and variable speed in the same range to see dynamic response of error. The constants values required by the model are obtained by BIKESIM dataset and it is shown in Table II.

M	234.36 [Kg]
$I_{zz}$	14 [Kg/m <sup>2</sup> ]
g	9.81 [m/s <sup>2</sup> ]
Rwr	0.29 [m]
$\epsilon$	27.72 [°]
Lf	0.56 [m]
Lr	0.81 [m]
$C_d$	0.52
$\rho$	1.167 [Kg/m <sup>3</sup> ]
$A_f$	0.6 [m <sup>2</sup> ]
$\mu_0$	0.008
$\mu_1$	7e-6

TABLE I: Constant values.

According to [10] a motorcycle with the size and weight described by BIKESIM use pneumatics with the friction parameters reported in Table II.

The parameters values shown are useful to make an initial approximation to the motorcycle behavior. However, it is an a priori estimation. For this reason, those friction coefficients and the effect of dynamics approximation made have to

$C_{f1}$	14900
$C_{r1}$	27296
$C_{f2}$	927
$C_{r2}$	1527

TABLE II: Friction coefficients apriori estimation.

be verified in order to determine friction coefficients which cause the minimum error and the limitations of the model developed.

#### A. Constant speed test.

This test was made to verify the behavior of each component of the longitudinal equation and detect easily mistakes.

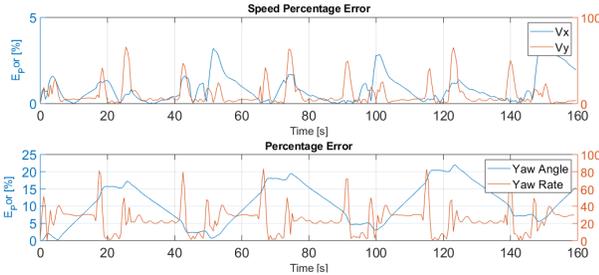


Fig. 4: Static speed profile results (error percentage).

As it can be seen in Figure 4, even when the error percentage of the longitudinal and lateral speed is lower enough (under 5%) to be considered as an acceptable estimation, the error of angle yaw increases fastly. It makes impossible to use yaw information as a constraint how it is required.

#### B. Dynamic speed test.

It is expected the error increase in face of dynamic speed references. For this reason, an estimation with this period of time is considered acceptable when the mean error is under 10% in dynamic speed conditions. In Figure Figure 5 lets to determine the prediction horizon for the current accuracy of the model. In this case, the prediction horizon is of 8 seconds to consider an error of less than 10% of the measure in longitudinal speed and yaw angle.

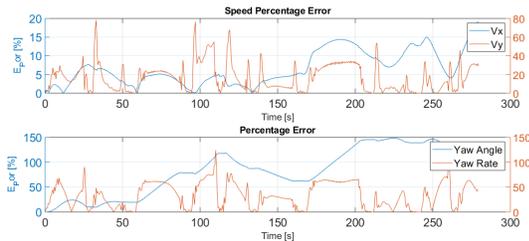


Fig. 5: Dynamic speed profile results (error percentage).

Taking into account the current results, it is important to make a sensitivity test to determine the behavior of percentage error as a function of the friction coefficients. Since the friction coefficients are assumed constant by the

theoretical model proposed, but they have a non-linear behavior described by Pacejka equation in BIKESIM, better results are expected with a better estimation of the approximated value of these parameters in an urban drive profile where the speed values are from 0 to 60Km/h.

### VI. SESIBILITY TEST.

The sensitivity test consists in the study of how each friction coefficient variation affects the mean percentage error. It is important to note that only one parameter is varied along each test, the other 3 parameters are considered constant along the test with apriori estimation value. For example in test 1, only  $C_{f1}$  varies.  $C_{r1}$ ,  $C_{r2}$  and  $C_{f2}$  are considered constant. The minimum and maximum values used in the test were chosen based in the physical restriction of the pneumatic dynamics, the values of friction coefficients found in the pneumatic industry and a minimum correlation between BIKESIM states and model sates of 75% without matters the absolute error between them. The delta values were chosen with the objective of causing a minimum variation of 0.5% in one or more model states between iterations.

	Minimum	Maximum	Delta
$C_{f1}$	5000	90000	100
$C_{r1}$	5000	90000	100
$C_{f2}$	0	10000	50
$C_{r2}$	0	10000	50

TABLE III: sensitivity test parameters.

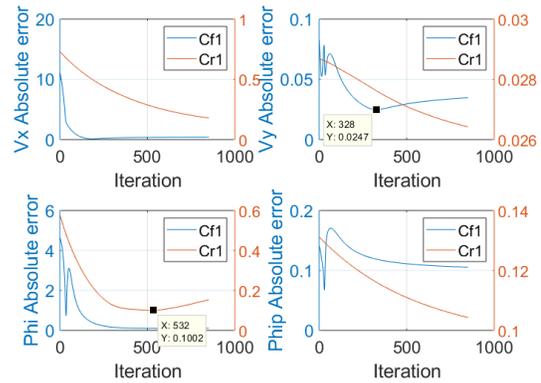


Fig. 6: sensitivity test Cf1 &amp; Cr1.

The result of sensitivity test is shown in Figures 6 and 7. The minimum error coefficient values obtained are summerized in Table IV.

### VII. RESULTS.

#### A. Static speed value

After the utilization of new friction coefficients estimated by the sensitivity test, the results show a decreasing of 73% of the mean error percentage in yaw mean error percentage. The static speed test result's Figure 8 reports a mean error percentage of 1.15% for longitudinal speed and 3.4% for

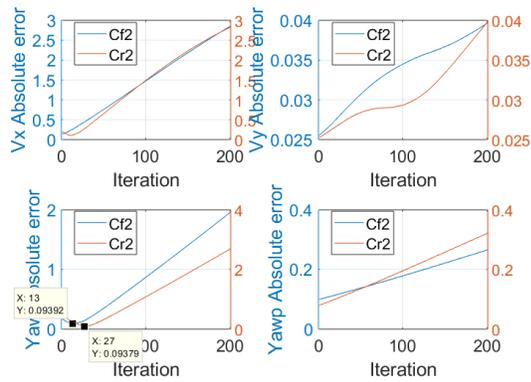


Fig. 7: sensitivity test Cf2 &amp; Cr2

Coeff	Iteration	Value
$C_{f1}$	328	37700
$C_{r1}$	532	58100
$C_{f2}$	13	600
$C_{r2}$	27	1300

TABLE IV: Friction coefficients estimation based on speed profile data.

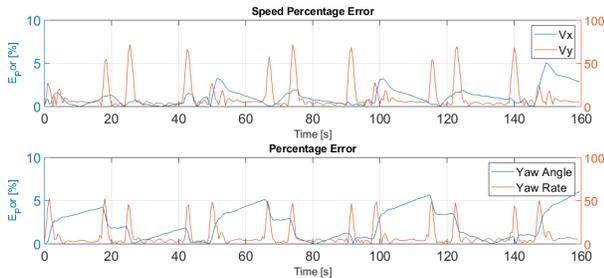


Fig. 8: Static speed profile results after sensitivity test (error percentage).

yaw angle. The Longitudinal speed and yaw angle percentage error signals in static speed value test are under the 5% and even when the mean errors are increasing along the test, their ratio is around 1% per 100 seconds in longitudinal speed and 1% per 40 seconds in the yaw angle. Those results are considered enough to continue to the dynamic test.

### B. Dynamic speed value.

In this case, the utilization of new friction coefficients estimated by the sensitivity test decreases a 71.97% of the mean error percentage in comparison with the yaw mean error percentage reported before. The dynamic speed test Figure 9 reports a mean error percentage of 4.47% for longitudinal speed and 23.12% for yaw angle. Although the yaw angle mean error percentage still being high, the new results let to obtain a prediction horizon of 66 seconds with a consideration of an acceptable yaw error under the 10%.

## VIII. CONCLUSION AND FUTURE WORK

More sophisticated equations can describe the behavior of the friction coefficient for different speed values, but

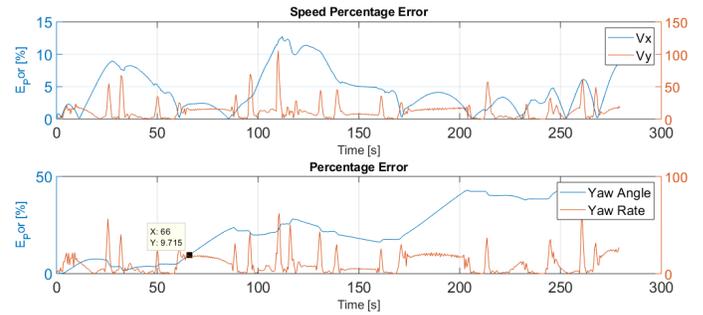


Fig. 9: Dynamic speed profile results after sensitivity test (error percentage).

with high compilation time too. By another hand, a characterization of the driving profile and road conditions like speed minimum and maximum values and slope angle lets to obtain acceptable results with linear pneumatic equations through the sensitivity test of the error signal. Those results are acceptable with the maximum speed value restriction of 60 km/h (An urban speed profile). If a highway speed profile is required, (in order to keep a linear description of pneumatic dynamics) a look-up table strategy can be used to determine the correct correspondence between different coefficient values and speed values due to the non-linear behavior described by Pacejka equation. This process would represent an expensive calculus but it can be approached with robust techniques as Montecarlo analysis, for example.

The dynamic model presented in this paper is able to recreate the mechanical behavior of a motorcycle in presence of roll, slope and bank angles with a maximal error of 10% on a period of time around 1 minute. The maximum roll angle which is acceptable to the assumptions made by the dynamic model equation is  $15^\circ$ . If the roll angle is over this value, dynamics related to the effective radius of the wheel and driver mass center coupling with the motorcycle center of mass have to be considered.

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