Modelling of all-wheel drive motorcycle for dynamics analysis and control

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ABSTRACT

Over the years, the trend of innovation and development of all-wheel drive (AWD) solutions for motorcycles has undergone a great delay compared to what occurred for cars, although four wheeled drive systems have shown to bring undoubted advantages in terms of safety, as well as in terms of stability and handling of the vehicle. The factors contributing to this delay are manifold. Jointly with market shares motivations, the most intuitive reason concerns the mechanical difficulties in distributing the torque to the front wheel, not to mention the complexity in managing the nature inherently unstable and highly nonlinear of an under actuated two-wheeled vehicle. The few commercial proposals available on the market, exploit only partially the potential of an AWD motorcycle, indeed, they are conceived to transfer to the front wheel only the exceeding part of the drive torque during the rear wheel slip. In this work, an AWD motorcycle model with attached rider is proposed. It has been developed a dynamic model in a symbolic way, and then compared with a multi-body simulation software. It allows investigating the front wheel drive effects on the motorcycle dynamics. The proposed model takes into account the lateral dynamics, the longitudinal dynamics and the tyre-road interactions.

INTRODUCTION

Since recent technologies allow simplifications of the driveline system, e.g. by housing an electric motor in the front wheel assembly, it becomes interesting reviewing the AWD system in a modern concept without excessive efforts of integration. Furthermore, state-of-the-art of sensor technologies, which are currently not deployed in AWD motorcycle models available on the market [1-2-3], allow an easier estimation of the vehicle's attitude and wheel's speed. Sophisticated control algorithms running on powerful MCU, also required by newest safety devices, can manage these estimates. In this perspective, greater research efforts would be required in AWD motorcycle modelling. At present, many researches have been done concerning the stability analysis and the dynamics behavior of bicycles and motorcycles. In [4] is reported a review on their modelling. However, few scientific works have been proposed regarding the AWD motorcycles, among which one worthy of mentioning can be found in [5], wherein a multibody simulator is proposed. On the other hand, the literature seems to be lacking AWD models described by Ordinary Differential Equations (ODE). The main objective of this work is to propose a proper ODE model for the purposes of analysis and design of AWD motorcycles as well as the testing of active torque distribution, regardless of the technology used for the front driveline system (mechanical, hydraulic or electric). It is a preliminary work and it aims to provide the foundation for further research activities.

AWD SYMBOLIC MODEL

Due to space limitations, this paper will only summarize the main equations useful to derive the complete symbolic model through the Euler-Lagrange formalism, referring for details to other publications. The Fig.1 shows the considered motorcycle's bodies and its parameters [6].

The vector of the generalized coordinates is defined as $\mathbf{q}=[x_1, y_1, \psi, \phi, \delta, \theta_r, \theta_f]^T$, where x_1, y_1 represent the reference point A displacements, whilst ψ and ϕ are the yaw and roll angles. The vertical motion is neglected. By denoting T, V the system's kinetic and potential energies, and with $\mathbf{Q_q}=[Q_{x1},Q_{y1},Q_\psi,Q_\phi,Q_\delta,Q_{\theta r},Q_{\theta f}]^T$ the vector of the generalized forces acting on $\mathbf{q}$, the Euler-Lagrange formalism (1) can be solved [7-8] by using the following equations ($i\in\{r,f\}$ refers to rear and front body)

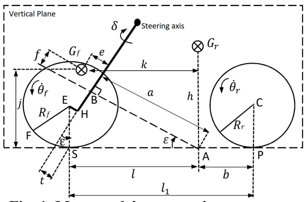

Fig. 1. Motorcycle's geometric parameters

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\partial T}{\partial \dot{\boldsymbol{q}}}\right)-\frac{\partial T}{\partial \boldsymbol{q}}+\frac{\partial V}{\partial \boldsymbol{q}}=\boldsymbol{Q_q} \quad (1)$$

$$T=T_r+T_f+T_\omega \quad (2)$$

$$T_i=\frac{1}{2}M_i v_i^2+\frac{1}{2}\boldsymbol{\omega_i^T I_i \omega_i},\ i\in\{r,f\} \quad (3)$$

$$T_{\boldsymbol{\omega}}=\sum_i T_{\boldsymbol{\omega wi}}+\sum_i T_{\boldsymbol{fly_i}},\ i\in\{r,f\} \quad (4)$$

$$V=V_r+V_f=M_r g z_r+M_f g z_f \quad (5)$$

where: $T_{\boldsymbol{\omega wi}}$=wheels' kinetic energy, and $\mathbf{T_{flyi}}$=rotational energies of flywheels
$\boldsymbol{\omega_i}, \boldsymbol{v_i}$ =bodies' angular and linear velocities

$$\begin{cases} Q_{x1}\approx X_f+X_r \\ Q_{y1}\approx Y_f+Y_r \\ Q_\psi\approx Y_f l-Y_r b+M_{z_f}+M_{z_r} \\ Q_\phi=-t\delta\left(Z_f\cos\phi-Y_f\sin\phi\right) \\ \qquad +M_{x_r}+M_{x_f}\cos\gamma \\ Q_\delta=(\tau-K\dot{\delta})+t\{[(Y_f\sin\phi-Z_f\cos\phi)\sin\varepsilon+(X_f-Y_f\gamma)\cos\varepsilon]\delta- \\ \qquad -\left(Y_f\cos\phi+Z_f\sin\phi\right)\}+M_{z_f}\cos\varepsilon\cos\phi \\ Q_{\theta i}\approx -\tau_{\omega_i}+R_i X_i,\ i\in\{r,f\},\ \tau_{\omega_i}\text{ are engine torques on the wheels} \end{cases}$$

TIRE MODEL

The AWD model here proposed adopts the Pacejka tire model [9], where tire forces (longitudinal X_i and lateral Y_i) and moments (overturning M_{xi} and aligning M_{zi}) are described in terms of the Magic Formulas, which depend on the tire slips (longitudinal slip λ_i, side slip α_i, and the theoretical slips: longitudinal σ_{xi}, lateral σ_{yi} and total σ_i). The Pacejka model adopts axes systems according to standard SAE J670 and 4976, whereas here is adopted ISO 8855 1991, which is most used as simulation oriented standard in automotive. Below a short summary of the used equations

$$X_i=\frac{\sigma_{xi}}{\sigma_i}\frac{Z_i}{Z_o}X_i^0\left(\frac{Z_o}{Z_i}\sigma_i\right) \quad (6)$$

$$Y_i=\frac{\sigma_{yi}}{\sigma_i}\frac{Z_i}{Z_o}Y_i^0\left(\frac{Z_o}{Z_i}\sigma_i,\begin{Bmatrix}\phi_r=\phi \\ \phi_f=\phi+\delta\sin\varepsilon\end{Bmatrix}\right) \quad (7)$$

$$M_{z_i}=-t_{r_i}Y_i+X_i s(Y_i,\phi)+M_{res_i} \quad (8)$$

$$M_{x_i}=R_i Z_i q(Y_i,\phi) \quad (9)$$

$$\text{with}\ \begin{cases}\lambda_i\approx-\dfrac{\dot{x}_1+R_i\dot{\vartheta}_i-R_i\sin\phi\dot{\psi}}{\dot{x}_1} \\ \alpha_r\approx-\dfrac{\dot{y}_1-b\dot{\psi}}{\dot{x}_1} \\ \alpha_f\approx\delta\cos\varepsilon-\dfrac{\dot{y}_1+l\dot{\psi}-t\dot{\delta}}{\dot{x}_1}\end{cases}\ \text{and}\ \begin{cases}\sigma_{xi}\triangleq\dfrac{\lambda_i}{1+\lambda_i} \\ \sigma_{yi}\triangleq\dfrac{\tan a_i}{1+\lambda_i} \\ \sigma_i=\sqrt{\sigma_{xi}^2+\sigma_{yi}^2}\end{cases}$$

where: $i\in\{r,f\}$,Z_0 is the nominal wheel load, Z_i is the load acting on the i^{th} wheel, and X_i^0, Y_i^0, are the longitudinal and lateral empirical Magic Formulae in pure slip condition.

AWD MULTIBODY MODEL

The AWD motorcycle has been also developed in the MSC Adams multibody environment, in order to compare both the results and model reliability. A multibody model consisting of four rigid bodies has been developed. It has 9 DOFs. The vertical displacement and pitch rotation have been considered neglectable under the assumptions in [6]. The rear and the front frames of the motorcycle were each divided into two rigid bodies, a frame without the wheel, and the wheel. This allows considering in the multibody environment the tire-road constraint. In MSC Adams the tire-road interaction makes use of the fully empirical tire model derived in [9]. Fig. 2 shows the

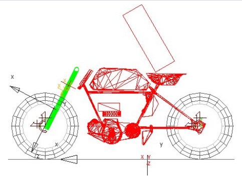
Fig. 2. AWD Multibody model

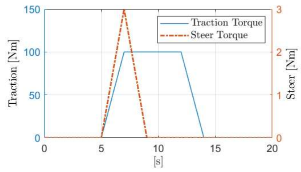

Fig. 3. The input torques

overall AWD motorcycle in the multibody environment. Both the models, in MSC Adams and the symbolic one, have the same masses, inertiae, geometric dimensions, and tires model hence, subject to same input torques, they must show similar behavior, as reported in the next section.

RESULTS

In order to point out the effects of the front wheel drive on the motorcycle's behavior, three simulations have been carried out for each model. Under the same conditions, and the same input torques depicted in Fig. 3, it has been done a comparison between the models and it was observed the effects of the front wheel drive on the vehicle.

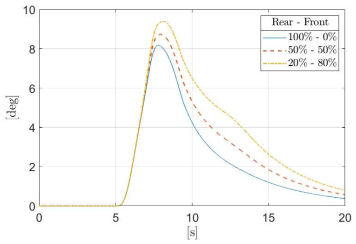

Fig. 4. Roll angle of multibody model

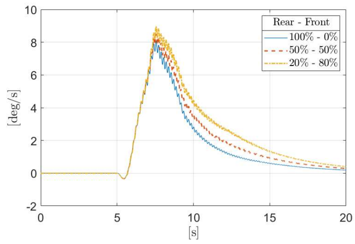

Fig. 5. Yaw rate of multibody model

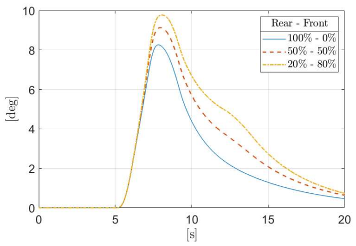

Fig. 6. Roll angle of symbolic model

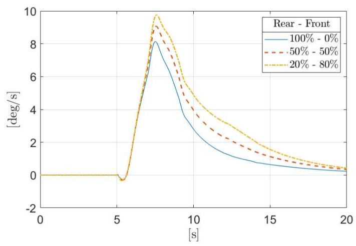

Fig. 7. Yaw rate of symbolic model

In Figures 4 and 5 are reported the cornering simulations of the multibody model, while the simulations of the symbolic model, under the same conditions, are reported in Figures 6 and 7. The figures show the trend of the most significant variables in cornering: the roll angle and the yaw rate. The same steer torque has been applied to the steering mechanism of both models, while the traction torque, showed in Fig. 3, was partitioned between the front and rear wheel with three different ratios. Both models show very similar dynamics responses with very few percentage points of difference in peak value (the yaw rate bouncing in Fig. 5 are attributable to the numeric solver of MSC Adams). Furthermore, both models show the significant effect of the front traction on the motorcycle behavior, indeed its action allows a narrow curved trajectory, by increasing both the roll angle and the yaw rate, other conditions being equals. Clearly this generates advantages in entering the curve and drawbacks in the exit of the curve.

CONCLUSIONS

In the present work an AWD motorcycle model is proposed, its symbolic formulation is briefly shown and then its behaviors have been compared with those of an AWD model developed in high realistic multibody environment. A comparison between the models' behavior has been carried out keeping the same simulation conditions and applied inputs. This allowed to highlight and to explain the effect of the front wheel drive on a two-wheeled vehicle. In brief, it occurs a rising of the roll angle and the yaw speed as more traction is given to the front wheel. Such an effect is desirable to achieve narrow curve trajectory, but may become a drawback when the vehicle is being raised in the exit of a curve. This suggest that an AWD motorcycle with a properly dynamically controlled traction torque distribution can have a noticeable handling advantage.

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