

Comparison Between Experimental and Numerical Handling Tests for a Three-Wheeled Motorcycle

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ABSTRACT

This paper deals with the analysis of the handling behaviour of a novel three-wheeled motorcycle. This vehicle has two front steering wheels and a single rear wheel and can be driven much like a common two wheeler.

In order to analyse the handling behaviour of such vehicle and to compare it to an ordinary two wheeler, an experimental campaign was conducted with the vehicle endowed with several transducers. Experimental tests included some classical handling manoeuvres. Concurrently, a simulation model was developed using a multi-body code. A simple logic was employed to drive the model; it consists in a roll follower and a longitudinal velocity follower.

The main dynamic parameters obtained from simulations, such as the steering angle and steering torque are compared to the experimental data and discussed. The effect of the driving style on the manoeuvre is also analysed with reference to steering pad manoeuvres.

INTRODUCTION

This paper deals with a research activity carried out at the Department of Mechanical, Nuclear and Production Engineering of the University of Pisa (Italy), in the framework of the European co-funded SIM ("Safety in Motion") project, within the 6th Framework Program. In the SIM project, a prototype of a motor scooter endowed with active and passive safety systems is going to be developed; with such a prototype the effect of safety systems on motorcycle will be investigated.

Piaggio MP3 (Figure 1), a novel three-wheeled motorcycle recently introduced into the market, was selected as the reference vehicle for the project. In this paper the handling behaviour of this vehicle is analysed by means of multibody simulation and experiments.

The vehicle has a classical scooter rear suspension, consisting of the engine and the CVT transmission assembly. On the other hand, MP3 has two front steering wheels connected by a four linkage system. This architecture allows the vehicle to roll like a conventional motorcycle. In addition, the two front wheels have the same steering angle and the total front vertical load is almost equally distributed between the right and the left wheel. As a consequence, MP3 can be driven like a common motor scooter (the technical literature on two wheelers is wide, see for example [1, 2]).



Figure 1: Piaggio MP3 scooter.

However, there are some interesting differences between MP3 and a common two-wheeler (a preliminary analysis of this vehicle was presented in [3, 4]). From the safety point of view, the two front wheels lower the probability of a sudden loss of grip, which, for example, can occur on uneven roads or due to unexpected events. In addition, since the friction coefficient between tires and road slightly increases if the vertical load is reduced, the two front wheels result in a higher overall friction coefficient. This, in turn, allows to shorten braking distances and achieve higher lateral accelerations. Finally, the front wheels also exert a gyroscopic effect with consequences on the stability.

In this activity, in order to analyse the handling behaviour of such a vehicle (and to compare it with a conventional two-wheeler) an experimental campaign was conducted by Piaggio & C., measuring the fundamental kinetic and dynamic quantities which describe the vehicle behaviour. On the other side, a multi-body model of the MP3 was developed using the commercial software MSC-Adams and the same manoeuvres of the experimental tests were simulated.

EXPERIMENTAL TESTS

CONSIDERED MANOEUVRES – Several experimental tests were conducted in order to analyse the dynamic behaviour of MP3. These include steering pads, lane changes, U-turns and slaloms. In the following, steering pad and lane change manoeuvres are analysed in details.

- **Steering pad:** they were executed on a 14 m radius track. The driver was asked to maintain a constant velocity and to follow the indicated track, in order to evidence the stationary behaviour of the vehicle. This kind of test was executed turning left and right at approximately: 5.6, 6.9, 8.3 and 9.7 m/s (20, 25, 30, and 35 km/h).
- **Lane change:** during this non steady state manoeuvre, the velocity was maintained almost constant at: 5.6, 8.3, 11.1, 13.9 and 16.7 m/s (20, 30, 40, 50 and 60 km/h).

In all the tests conducted, the driver was asked to maintain, as possible, a fixed position relative to the vehicle frame, so that the sprung mass could be considered as a rigid body in the first stage of the simulation tests.

EMPLOYED SENSORS - The vehicle used to perform the experimental tests was endowed with some sensors in order to achieve the fundamental quantities needed to describe the vehicle behaviour. The instrumental equipment was composed by the following sensors:

- **IMU** (Inertial Measurement Unit): it consists of three gyroscopic sensors which measure directly the three components of the angular velocity of the sprung mass in a body-fixed coordinate system. There are also three accelerometers along the same directions of the gyroscopic sensors, which give the three components of the acceleration along the same body-fixed coordinate system. In addition to these measured quantities, the inertial platform also outputs, as computed signals, the roll and the pitch angles.
- **GPS:** it gives the position of the installation point in a ground frame and evaluates the point velocity, the distance travelled and the trajectory.
- **Steering sensors:** these sensors give the steering angle and the steering torque applied by the driver.

MULTIBODY MODEL

VEHICLE MODEL – A multibody model of the vehicle was implemented in the MSC-Adams environment. The model consists of several rigid bodies (see Figure 2). The inertial properties for the main bodies were obtained on the basis of the 3D CAD drawings and technical data. The front four linkage architecture is modelled with two side arms and two horizontal arms connected by four revolute joints. The mechanism is also linked to the mainframe by other two revolute joints on the upper and lower arm. The two front suspensions can rotate around the relative side arm when a steering movement is imposed to the steering column. In the rear part of the vehicle model there is the rear suspension assembly, which includes the engine and the CVT transmission.

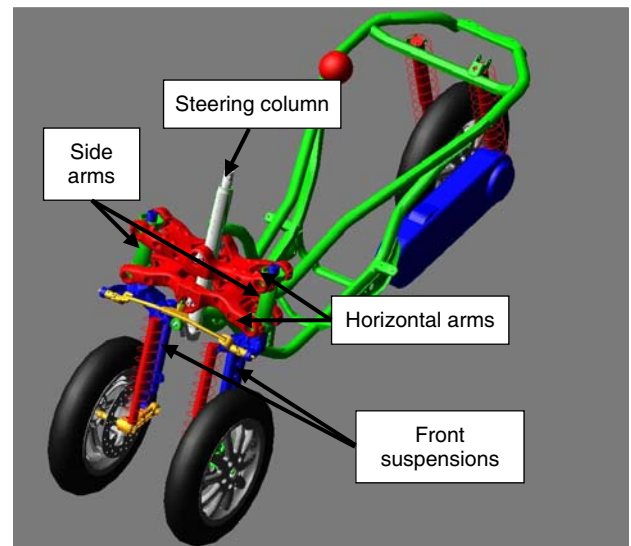


Figure 2: Multibody model of MP3.

The stiffness and damping properties of the suspensions were obtained from experimental tests. The measured data are interpolated in the multibody software using the Akima fitting method. The MF-MCtire model developed by TNO has been used in the Adams model. This tire formulation is derived from the well known Magic Formula but it is more suitable for large camber angles, typical for motorcycle applications.

DRIVER MODEL – From a physical point of view, the easiest way to model the driver, is to represent it as a rigid body fixed to the vehicle mainframe. However, the common experience suggests that a real motorcyclist usually changes his position with respect to the vehicle, thus changing the configuration of the system. For this reason, in addition to a fixed driver, a two-mass driver was also set up. This driver is composed by a first body attached to the vehicle main frame and a second body connected to the first one by a revolute joint. The lateral shifting of the driver centre of mass with respect to the scooter symmetry plane was used in order to analyse the effect of the driver position on simulation results.

In details, the driver lower body, fixed to the frame, represents the legs and the forearms, whereas the upper body, represents the head, the arms and the trunk.

The generic driver model, which can be used for all simulations, is basically a roll and forward speed follower; the desired roll angle and forward speed are denoted by $\phi_d(t)$ and $u_d(t)$. The driver logic is then composed by two controllers, based on those two variables.

The roll follower controller acts on the steering angle velocity $\dot{\delta}$, following this rule:

$$\dot{\delta}(t) = k_0 \ddot{\phi}(t) + k_1 \dot{\phi}(t) + k_2 (\phi_d(t + T_p) - \phi(t)), \quad (1)$$

where ϕ is the vehicle roll angle, T_p represents a preview time and k_i ($i=0, \dots, 2$) are constants.

During simulations, the controller evaluates the desired roll function at time $t+T_p$. If $T_p=0$, the controller follows the roll angle objective with a time delay, depending on the values of the controller constants. In this case, a very "reactive" driver (high values of k_i), able to follow closely the desired roll angle, may result too sensitive to variations, with a relatively high risk of falling. With the preview time set to a reasonable value (0.1 – 0.3 s) the driver appeared to be correctly modelled and able to reproduce even fast transient manoeuvres.

The longitudinal velocity follower controls the first derivative of the traction force $\dot{F}_t$, which is applied to the rear tire contact patch, given by:

$$\dot{F}_t(t) = c_1 \dot{u}(t) + c_2 (u_d(t) - u(t)), \quad (2)$$

where u is the vehicle longitudinal velocity and c_i ($i=0,1$) are constants.

In the steering pad manoeuvres it was also found useful to have a trajectory controller, based on the desired curvature radius R_d instead of the roll angle ϕ_d . In order to avoid the vehicle from falling, the radius follower was formulated in a way, similar to the roll follower. The following relationships were considered, which are valid in steady state conditions, for a vehicle with lenticular tires, neglecting the gyroscopic effect of wheels:

$$\tan \phi = \frac{u^2}{R g} \quad \text{and} \quad R = \frac{u}{\dot{\psi}}, \quad (3)$$

where $\dot{\psi}$ is the yaw velocity, R the curvature radius and g the gravity acceleration. With reference to a steering pad manoeuvre, considering equations (3), the following driver was also considered:

$$\begin{aligned} \dot{\delta}(t) &= b_0 \ddot{\phi}(t) + b_1 \dot{\phi}(t) + b_2 \left(\frac{u_d^2}{g R_d} - \frac{u^2(t)}{g R(t)} \right) = \\ &b_0 \ddot{\phi}(t) + b_1 \dot{\phi}(t) + b_2 \left(\frac{u_d^2}{g R_d} - \frac{u(t) \dot{\psi}(t)}{g} \right) \end{aligned} \quad (4)$$

u_d and R_d are the desired radius and forward speed, referenced to the experimental test, while $u(t)$ and $\dot{\psi}(t)$ are the run-time variables of the simulation; b_i ($i=0, \dots, 2$) are constants.

RESULTS

ANALYSIS OF SINGLE MANOEUVRES – In order to reproduce the experimental tests in the simulation environment, the virtual driver is asked to follow the measured roll angle and longitudinal velocity. The correlation between experimental data and simulation was based on the comparison between the roll and the steering angles. The former is a parameter related to the driver model behaviour, indicating how correctly the controller replicates the experimental manoeuvre, whereas the latter (steering angle) indicates how good is the correlation between experimental and simulation data.

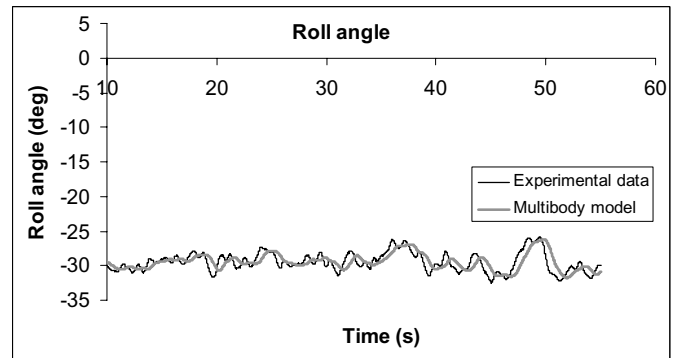


Figure 3: Roll angle in a steering pad manoeuvre at 8.3 m/s (30 km/h).

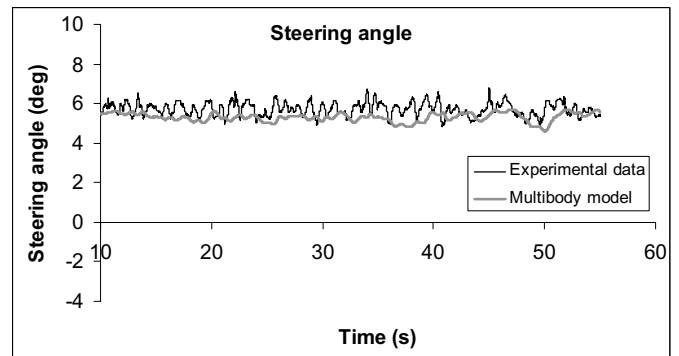


Figure 4: Steering angle in a steering pad manoeuvre at 8.3 m/s (30 km/h).