Dynamic modelling and simulation of the green vehicle PICAV

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Abstract: - PICAV is a new Personal Intelligent City Accessible Vehicle (PICAV) that is conceived to ensure city accessibility for everybody and some of its features are specifically designed for people whose mobility is restricted for different reasons, particularly (but not only) elderly and disable. Ergonomics, comfort, stability, assisted driving, eco-sustainability, parking and mobility dexterity are the main drivers of the PICAV design. The vehicle has small dimensions and tiny footprint, it is powered by four in wheel electric motors and is endowed with new frame- suspension structures. Because it is suitable for weak people and for uneven difficult city centers environment, in order to guarantee the stability and mobility dexterity in different manoeuvring operations, its design was based on a detailed dynamical model that is here presented. Further a simplified model is derived with the main objective to be used by the driver assistant module and, in the case of autonomous driving, by the vehicle motion control.

Key-Words: - Sustainable urban vehicle, Full electric vehicle, 4 independent wheels, mathematical model, Personal vehicle, Simulation

1 Introduction

Mobility and communication are markers of civilised society level: the restrictions on people or ideas always mean dark seasons. Sustainable development, however, requires today compatibility checks, with emphasis for highly pedestrianized areas. Many technical developments ought to be reconsidered, possibly, rethinking solutions accepted as obvious, that actually lay within the range of habits [1]. City-cars and personal-movers are example case, where innovations and non-conventional ideas have to be weighed to look after effectiveness. These vehicles are developed for local traffic duty, worth for low speed and bounded autonomy; several auto-makers are becoming aware of the challenge [2], still the offers do not turn out from to many conventional technicalities. PICAV, Fig. 1, is a small electric motorised vehicle, able to follow paths and slopes, with given performance (payload, speed, acceleration, autonomy, etc.). Transport systems for pedestrian areas, based on a fleet of fully-automated PICAVs have been proposed in [3] and [4]. The paper reconsiders the driving and manoeuvrability operations, assuming four independently powered wheels [5], [6]. The investigation builds up the vehicle dynamics moving from the behaviour of a driving wheel (with compliant tyre); modelling the group motor wheel suspension and then assembling the four groups to the chassis to find out the motion of the centroid and around it when the four actuators operate while the vehicle moves on varying soil surfaces.

The dynamic model is first written in the general case of six degrees of freedom vehicle with four motorized steering wheels and then non steering wheels condition is introduced.

<table>
<thead>
<tr>
<th>Legenda</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total mass</td>
<td>M</td>
<td>400 kg</td>
</tr>
<tr>
<td>Height of the gravity center</td>
<td>rGz</td>
<td>0.58 m</td>
</tr>
<tr>
<td>Wheel rolling radius</td>
<td>R</td>
<td>0.2 m</td>
</tr>
<tr>
<td>Wheel moment of inertia</td>
<td>I</td>
<td>0.1 kgm2</td>
</tr>
</tbody>
</table>

Fig.1 PICAV digital mock-up and main parameters
The actuation redundancy makes these models complex, but, at the same time, opens new opportunities, on condition that proper information is exploited. An interesting opportunity is to exploit the actuation torque supplied by each motor and the speed of the individual wheels, to reckon the useful traction force required at soil-tyre interfaces, once the adherence and creeping condition are assessed [7]. From the analysis performed by researchers in the field, has been shown that a four independent motor wheels vehicle can be driven along any path by simply controlling the slip, without necessarily impressing a turning angle to the wheels. On these premises, the manoeuvre dexterity and stability are considered the main performances for all the tasks a city vehicle is required to perform at low speed over urban roads [8].

2 Formulation of the PICAV dynamic model

The study deals with electrically powered personal vehicles, used to transport one person with limited autonomy range (some 60 km), at low speed (max 25 km/h), assuming proper soil upkeep. The four wheels are individually actuated, but the driver, as usual, acts on two commands: steering and acceleration, while the controller needs modify the torque and the speed, so that the (four) soil/tyre interfaces grant the required manoeuvrability and stability [9]. The problem is consistent, by itself, with several issues, on condition that the vehicle dynamics is modelled with account of the pertinent degrees of freedom.

Quite often, indeed, the reference models are limited to ‘quarter car’ or to ‘bicycle’ cases, totally or partially neglecting roll and pitch motion. In this case, due to the PICAV special aim to guarantee the accessibility to city centers to all, including elders and mobility impaired people, stability and manoeuvrability are the main issues [10]. So detailed reliable models are needed as a base for the vehicle mechatronic design and real time control system set-up [11].

The vehicle is modelled as a multi bodies system: the chassis connected, through viscous-elastic joints, to four masses including suspensions and motorised wheels, each one coupled with the road.

The basic modules involved in the analysis are summarised by the Fig. 2. Due to the low speed and the comparatively smooth soil, finally, a 14-degrees-of-freedom model is obtained: 6 for the car body (3 of the centroid and 3 around it); 2 for each of the four suspended masses (the linear motion at joints and the rotation of the wheels); namely, in body-axes and assuming small angular deflections, the reference dynamics is set, as in the following sections.

Fig.2 Modular view of the vehicle model. The grey boxes are the basic modules, the white ones are composition of the basic modules.

2.1 The motor-wheel-suspension group

The motor wheel dynamic model considers all the degrees of freedom supplied from the rotations around three non orthogonal axes: spin \( \theta \) around the current y axis \( (j) \), corresponding to spinning torque \( M_s \), steering \( \delta \) around the z fixed axis \( (e_3) \), corresponding to steering torque \( M_{st} \), camber \( \gamma \) around the current x axis \( (i) \), corresponding to camber torque \( M_c \), see Fig. 3a. \( \mathbf{R} \) is the road reaction on the tyre and \( F_{wh} \) is the force applied by the chassis suspension to the wheel.

Fig.3 Wheel (a) and motor-wheel (b) schemes; wheel-chassis outline (c)

The motor wheel model is hereafter written in the general case as derived in [4]:

\[
\begin{align*}
\mathbf{r} &= \mathbf{e}_1 \times \mathbf{r} \\
\mathbf{F}_{wh} &= \mathbf{R} + \mathbf{F}_{wh} - \mathbf{M}_{wh} \times \\
\mathbf{M}_{wh} &= M_{wh} \mathbf{e}_2 \\
\end{align*}
\]
Translation equilibrium:

\[
\begin{align*}
R_s - F_{s_{\text{veh}}} & = m R \begin{bmatrix}
\alpha_s & \omega_s \omega_c & + 2 \omega_s \omega_c c_j + \alpha_s \\
\alpha_s & \omega_s \omega_c & + 2 \omega_s \omega_c c_j + \alpha_s \\
\alpha_s & \omega_s \omega_c & + 2 \omega_s \omega_c c_j + \alpha_s \\
\end{bmatrix} \\
R_i - F_{s_{\text{veh}}} & = m R \begin{bmatrix}
\alpha_i & \omega_i \omega_c & + 2 \omega_i \omega_c c_j + \alpha_i \\
\alpha_i & \omega_i \omega_c & + 2 \omega_i \omega_c c_j + \alpha_i \\
\alpha_i & \omega_i \omega_c & + 2 \omega_i \omega_c c_j + \alpha_i \\
\end{bmatrix} \\
R_c - F_{s_{\text{veh}}} & = m R \begin{bmatrix}
\alpha_c & \omega_c \omega_c & + 2 \omega_c \omega_c c_j + \alpha_c \\
\alpha_c & \omega_c \omega_c & + 2 \omega_c \omega_c c_j + \alpha_c \\
\alpha_c & \omega_c \omega_c & + 2 \omega_c \omega_c c_j + \alpha_c \\
\end{bmatrix}
\end{align*}
\]

Rotation equilibrium:

\[
\begin{align*}
M_i + RR_s & + RR_s - M_{\text{veh}} = M_{\text{c}} - RR_c \\
M_i - RR_i & + M_{\text{c}} - MR_{\text{veh}} = -I \omega_i \\
M_i - M_{\text{veh}} & = I \omega_i \\
\end{align*}
\]

Where \( \omega \) and \( \alpha \) are the wheel angular velocity and acceleration while their subscripts \( s, \text{st}, \text{c} \) refer respectively to spin, steering, camber; the wheel-motor mass is \( m' \) and the mass quadratic moment \( I \) is diagonal with \( I_x = I_z = I \); \( s, \text{st}, \text{c} \) refer to rectilinear motion, \( s \), \( \text{st} \), \( \text{c} \).

The influence of the tire slip at the interface between soil and wheel is really important for the car dynamics and stability behavior. In technical literature different definitions of tires slip \( s \) are given. In the present work the Magic formula of Pacejka [12] was used to model and simulate the tire behaviour [8]. With reference to the \( i^{th} \) wheel:

\[
s_j = \frac{\omega_j R - v_j}{|v_j|} \quad \text{with} \quad |v'_j| = \max(|v_j|, 0.01)
\]

\( \omega_j \) is the wheel angular velocity, \( R \) is the wheel radius, \( v_j \) is the wheel longitudinal velocity.

This formula presents the advantage to be free of singularity in zero, to be independent from the operative behaviour and velocity direction.

The wheel kinematic model, with reference to Fig. 3c, is:

\[
v_a = v_G + \dot{r}_a = v_G + r_i \dot{\psi}_a \\
v_i = v_G + \dot{r}_i = v_G + r_i \dot{\psi}_a \\
\dot{\psi}_a = \dot{\alpha}_G + a_G = a_G + r_i \dot{\psi}_a - r_i \psi'_a
\]

\[
v_i = v_G + \dot{r}_i = v_G + r_i \dot{\psi}_a \\
\dot{\psi}_a = \dot{\alpha}_G + a_G = a_G + r_i \dot{\psi}_a - r_i \psi'_a
\]

Where \( \dot{\psi} \) is the vehicle yaw angle, \( v_G \) and \( a_G \) are the velocity and acceleration vectors of the vehicle center of mass: \( v_G = [v_G, v_G]^T = [\dot{x}, \dot{y}]^T \)

The relation between the reaction force \( R_{Xi} \) and the slip \( s_i \) can be approximated by a symmetric saturation law:

\[
F_{si} = k_i s_i = \mu F_G \quad \text{if} \quad |s_i| \leq s_{sat}; \quad F_{si} = k_i s_{sat} = \mu F_G \quad \text{if} \quad |s_i| > s_{sat}
\]

Where: \( k_i \) is the \( i \) wheel slip law constant, \( s_{sat} \) is the saturation slip and \( \mu_i \) is the wheel adherence coefficient.

The lateral reaction \( F_Y \) is considered proportional to the slip angle \( \beta_i \):

\[
R_{yi} = C_i \beta_i \quad \text{where:}
\]

\( C_i \) is the \( i \) tire cornering stiffness;

\[
\beta_i = \text{sgn}(v_y) \cdot \text{atan}
\]

The model of wheel dynamics, including the slip non linear behaviour, allows to find the wheel torque \( T_i \) and the maximum torque that can be applied to the wheel before creep.

The forces exchanged between tire and soil depend on complex phenomena but the basic technical literature [9], [12] suggests simple proportionality relations for micro-slip contacts (say, out of extended creep situations). So, within the slip saturation limits, the wheel dynamics model, in case of rectilinear motion, is simplified in:

\[
T_i - R_i = \dot{r}_i \\
R_i = k_i \dot{s}_i = m \dot{v}_i
\]

where \( m \) and \( I \) are mass of the wheel and its moment of inertia referred to the wheel axis; \( k \) is the slip law constant.

Then, taking into account the expressions of \( s \) (5) and:

\[
\dot{\alpha}_i = \frac{\dot{v}_a + m \dot{v}_a v_a + v_a^2}{R} \frac{m}{k_i} \left( \dot{v}_a + v_a^2 \right) \frac{1}{k_i R} \frac{m}{k_i} \left( \dot{v}_a + v_a^2 \right) + \dot{v}_a
\]
The schema of the suspensions and terminology are given in Fig. 4, where $\psi$ is the yaw angle, $F_{\text{wh}}$ and $M_{\text{wh}}$ are force and moment due to the motor wheel within the hypothesis of stiff motion; $m_{\text{susp}}$ and $I_{\text{susp}}$ are the mass properties of the suspension, $k_{\text{susp}}$ and $c_{\text{susp}}$ are the elastic and damping parameters of the suspensions; $z_s$ is the quote of the suspension mass $m_{\text{susp}}$ considered concentrated; $\mathbf{r}^i_{\text{GS}}$ is the vector between the chassis center of mass $G$ and the suspension contact point $S$; $\mathbf{a}_G$ is the $G$ acceleration; the superscript indicates the reference frame $\{i', j', k'\}$.

![Fig. 4 Schema of the visco-elastic elements between the road and the vehicle body (chassis); 1 and 3 indicate the left side wheels, 2 and 4 indicate the right side wheels, 1 and 2 are front wheels.](image)

The proposed dynamic model of the individual suspension is:

$$
\begin{bmatrix}
F_{\text{swh}} - F_z \\
F_{\text{swh}} - F_z \\
F_{\text{swh}} - c_{\text{susp}}v_z - k_{\text{susp}}v_z
\end{bmatrix} = m_{\text{susp}}
\begin{bmatrix}
\alpha_{\text{swh}} - \psi r_{\text{ch}} - \psi' r_{\text{ch}} \\
\alpha_{\text{swh}} - \psi r_{\text{ch}} - \psi' r_{\text{ch}} \\
\alpha_{\text{swh}} - \psi r_{\text{ch}} - \psi' r_{\text{ch}}
\end{bmatrix}
$$

(11)

$$
\begin{bmatrix}
zF_{\text{swh}} + M_z - M_{\text{swh}} \\
zF_{\text{swh}} + M_z - M_{\text{swh}} \\
M_z - M_{\text{swh}}
\end{bmatrix} = 0
$$

(12)

In the case of stiff suspensions the vertical acceleration is zero: $\ddot{z}_s = 0$

The equations of motor wheels and suspensions can be assembled together to have the model of the group motor wheel suspension.

### 2.2 The chassis

The chassis is considered as a rigid body with 6 degrees of freedom. The reference frames are shown in Fig. 5.

![Fig. 5 Co-ordinate frames, roll ($\phi$), pitch ($\theta$), yaw ($\psi$) and steering ($\delta$) angles for basic rotations. The subscript ch refers to the chassis, wh refers to the wheel, $\{i', j', k'\}$ is the fixed frame. These frames and angles refer to the general vehicle model. In the case of PICAV (no steering wheels) $\delta = \psi$ and $\{i', j', k'\}_\text{wh}$](image)

Taking into account the forces and moments applied by the four wheels, reported in the chassis reference frame, the dynamic model of the chassis, in the case of wheels null camber and chassis inertia diagonal, is given by the two following vector equations that represent the equilibrium to translation and respectively to rotation of the chassis centroid along any three-dimensional path:

$$
\sum_{j=1}^{4} \left( F_j \psi (\theta - \delta) - F_j s (\theta - \delta) \right) = 0
$$

$$
\sum_{j=1}^{4} \left( F_j s (\theta - \delta) + F_j c (\theta - \delta) \right) = 0
$$

$$
\sum_{j=1}^{4} F_j - m_{\text{ch}} g = 0
$$

$$
\begin{bmatrix}
\alpha_{\text{Gch}} \\
\alpha_{\text{Gch}} \\
\alpha_{\text{Gch}}
\end{bmatrix} + m_{\text{ch}} \begin{bmatrix}
(\phi c + \psi c \psi s \phi) v_{\text{ch}} - (\psi c \phi - \theta s \phi) v_{\text{ch}} \\
(\psi c \phi - \theta s \phi) v_{\text{ch}} - (\phi - \psi s \theta) v_{\text{ch}} \\
(\phi - \psi s \theta) v_{\text{ch}} - (\theta c \phi + \psi c \theta \phi) v_{\text{ch}}
\end{bmatrix}
$$

Setting $I_{\text{ch}} = \text{diag}[I_x, I_y, I_z]$ the inertia matrix of the chassis; $m_{\text{ch}}$ the chassis mass; $[r_x, r_y, r_z]^T$ the arm vector of the force $F_k$ applied to the chassis from the motor wheel $k$ to the chassis center of mass, the dynamic equilibrium to the rotation is:

$$
\sum_{j=1}^{4} \left( F_j c (\theta - \delta) - F_j s (\theta - \delta) \right) = 0
$$

$$
\sum_{j=1}^{4} \left( F_j s (\theta - \delta) + F_j c (\theta - \delta) \right) = 0
$$

$$
\sum_{j=1}^{4} F_j - m_{\text{ch}} g = 0
$$

$$
\begin{bmatrix}
\alpha_{\text{Gch}} \\
\alpha_{\text{Gch}} \\
\alpha_{\text{Gch}}
\end{bmatrix} + m_{\text{ch}} \begin{bmatrix}
(\phi c + \psi c \psi s \phi) v_{\text{ch}} - (\psi c \phi - \theta s \phi) v_{\text{ch}} \\
(\psi c \phi - \theta s \phi) v_{\text{ch}} - (\phi - \psi s \theta) v_{\text{ch}} \\
(\phi - \psi s \theta) v_{\text{ch}} - (\theta c \phi + \psi c \theta \phi) v_{\text{ch}}
\end{bmatrix}
$$

Setting $I_{\text{ch}} = \text{diag}[I_x, I_y, I_z]$ the inertia matrix of the chassis; $m_{\text{ch}}$ the chassis mass; $[r_x, r_y, r_z]^T$ the arm vector of the force $F_k$ applied to the chassis from the motor wheel $k$ to the chassis center of mass, the dynamic equilibrium to the rotation is:
\[
\begin{align*}
\sum_{i=1}^{\frac{3}{2}} M_{wi}\psi_i + M_{wi}y_{wi} &= M_{wi}y_{wi} + \sum_{i=1}^{\frac{3}{2}} M_{wi}\psi_i + M_{wi}y_{wi} \\
+ \sum_{i=1}^{\frac{3}{2}} M_{wi}y_{wi} \psi_i &= M_{wi}y_{wi} + \sum_{i=1}^{\frac{3}{2}} M_{wi}\psi_i + M_{wi}y_{wi} \\
\sum_{i=1}^{\frac{3}{2}} M_{wi}\psi_i + M_{wi}y_{wi} &= M_{wi}y_{wi} + \sum_{i=1}^{\frac{3}{2}} M_{wi}\psi_i + M_{wi}y_{wi}
\end{align*}
\]

This model is hyper static and it will be solved admitting the simplifying hypotheses on the suspensions/chassis relative stiffness. It will be here admitted that the suspension stiffness is very high so that roll and pitch motions can be neglected (\(0=0, \phi=0\)). Further, considering the case of PICAV with no steering wheels, \(\delta=\psi\), the simplified model is:

\[
\begin{bmatrix}
\sum_{i=1}^{\frac{3}{2}} F_{xi} \\
\sum_{i=1}^{\frac{3}{2}} F_{yi} \\
-\sum_{i=1}^{\frac{3}{2}} F_{zi} - m_i g
\end{bmatrix}
= m_0 \begin{bmatrix}
a_{\theta} \\
a_{\psi} \\
0
\end{bmatrix}
+ m_0 \begin{bmatrix}
-\psi y_i \\
y_i v_i \\
0
\end{bmatrix}
+ m_0 \begin{bmatrix}
0 \\
I_{\psi i} \dot{\psi}_i \\
0
\end{bmatrix}
\]

(15)

| \(F_{xi} = r_i F_{x1} + r_i F_{x2} + r_i F_{x3} + r_i F_{x4} + r_i F_{x5} + r_i F_{x6} + r_i F_{x7} + r_i F_{x8} + \sum_{i=1}^{\frac{3}{2}} M_{zi} \) |
| \(F_{yi} = r_i F_{y1} + r_i F_{y2} + r_i F_{y3} + r_i F_{y4} + r_i F_{y5} + r_i F_{y6} + r_i F_{y7} + r_i F_{y8} + \sum_{i=1}^{\frac{3}{2}} M_{zi} \) |
| \(F_{zi} = r_i F_{z1} + r_i F_{z2} + r_i F_{z3} + r_i F_{z4} + r_i F_{z5} + r_i F_{z6} + r_i F_{z7} + r_i F_{z8} + \sum_{i=1}^{\frac{3}{2}} M_{zi} \) |

(16)

2.1 PICAV vehicle simplified model

The dynamic models of the vehicle single modules have been presented with different levels of approximation. The vehicle complete detailed model obtained by linking these modules equations is complex. This section is dedicated to present a further simplified version of the 2D model written in terms of state equation, useful for developing control algorithms. In the case of: - chassis suspensions connections sufficiently stiff compared to the forcing band due to the ground unevenness; - short suspension arms; - adopting the ground model proposed in [8] and working within the slip saturation limits; - zero steering angle relative to the chassis; the adopted methodology in accordance with the one addressed by [9].

It is considered the state vector

\[ \mathbf{x} = [\omega_1, \omega_2, \omega_3, \omega_4, \dot{x}, \dot{y}, \psi]^T, \]

where the \(\omega_i\) is the wheel i angular velocity and:

\[ \dot{x} = v_x; \dot{y} = v_y; \psi = \frac{d\psi}{dt}. \]

The dynamic equations for the four wheels are derived from (9), substituting the longitudinal force \(F_x\) due to slip:

\[ \dot{\omega}_i = \frac{T_i}{J_i} \left( \frac{\omega_i R_i - \dot{x} + \psi r_i}{\dot{x} + \psi r_i} \right) \frac{R_i}{I_i} \]

(17)

The two translation equations of the chassis are obtained by substituting in (15) the \(F_{xi}\) and \(F_{yi}\) from (7) and (9), repeated in (19). The dynamic equilibrium to rotation derives from (16) taking into account the expressions of \(F_{xi}\) and \(F_{yi}\).

The state equation can be written in the standard non linear form \(\dot{\mathbf{x}} = \mathbf{f}(\mathbf{x}) + \mathbf{B} \cdot \mathbf{u}\) where:

(18)
where: $T_j$ is the actuation torque reduced at each wheel axis and subscripts 1÷4 refer to the wheels. The model is non-linear and includes dynamic couplings. The forces exchanged between tyre and soil depend on complex phenomena, as addressed in section 2.1, but the basic technical literature [8],[12] suggests simple proportionality relations for micro-slip contacts (say, out of extended creep situations):

$$F_x = k_s s = \frac{\omega R}{\dot{x} + \dot{\psi}_y} - 1 \quad F_y = C_{\psi} \tan \left( \frac{\dot{y} - \dot{\psi}_x}{\dot{x} + \dot{\psi}_y} \right)$$  \hspace{1cm} (19)

Happily enough, the PICAV manoeuvring tasks remain inside these conditions, thereafter, adapting the actuation law, the body applied forces vary in proportion with the supplied torques and the vehicle trajectory is accordingly modified. Of course, front and rear axes should be balanced in the average (to avoid forced creep due to kinematic constraints); while bent angles are properly obtained by impressing the pertinent speed to, either, the left or the right train of wheels. Within the proposed scheme, the angular deflections of the vehicle is related to the mentioned differential modulation of the left- by respect to the right-side actuation and the car speed depends on the overall delivered power.

The feasibility of practical manoeuvres can be shown by solving the equations of the modelled dynamics. The solution has been done using Matlab/Simulink, resorting to a library purposely written for vehicle dynamics, by fully exploiting the modelling modular approach. In the PICAV dynamics simulation the external aerodynamic forces were neglected due to the PICAV low velocity.

3 Simulation results

The models of the single bodies of the vehicle have been used to generate the various Simulink modules, with different level of approximation [8], that suitably linked as in Fig.2 give the virtual dynamic mock-up of our vehicle PICAV.

The models were validated in different ground friction and trajectory conditions. Some results are reported in [4].

The vehicle PICAV is thought for weak, elderly and motion impaired, people to move even in difficult city centers grounds, so the stability is a main concern.

3.1 PICAV behavior to impulsive torques actuation

To study the intrinsic stability of the vehicle, before to apply any control strategy, in order to assess its behaviour to reactions and yaw moments on the chassis, some simulations have been performed applying differential torques of short duration to the motor wheels, leaving the vehicle then move without actuation.

As example, applying a very high torque, mimicking an impulsive torque: 170 Nm to the outer train and -170 to the inner from start to 0.2 s, then suddenly removing the torques, the results are shown in Fig. 6.

![PICAV path](image1)

![Chassis acceleration and velocity](image2)

Fig.6 PICAV path (a); chassis acceleration and velocity (b).
independent from the transversal reactions and its yaw dynamics is mastered by the only longitudinal reactions. The PICAV behaviour is congruent with the one investigated for another city car [8] in similar operative conditions.

3.2 PICAV behavior to bang bang torques actuation

As a second example case the vehicle behaviour under the application of bang bang pulses of torques on the four wheels is illustrated in Fig.7. The path tracked shows that the effect of the counter-torques is the vehicle stabilization, Fig.7(b), that happens very quickly; in effect the counter-torques are applied when all the wheels are skidding and the yaw dynamics is mainly influenced by the longitudinal reactions, different from the applied torques.

Only the behaviour of wheels 1 and 4 are reported because the slip on the front wheel 2 has the same trend as in front wheel 1; similar trends are obtained for the two rear wheels; the soil reactions are similar for the left side wheels and for the right side wheels. In Fig.8 it can be seen that the moment due to longitudinal reactions $M_x$ nullifies before the end of the skidding and the corrective action is sufficient to take again the PICAV on the route.

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Fig.7 Torque laws applied to wheels (a), performed path (b.), slip (c) and soil reactions (d) on wheels 1 (front left) and 4 (rear right).
4 Conclusion

The paper proposes the model for a 4 non steering motor-wheels vehicle. The model is applied to the new personal vehicle PICAV to study its open loop performances about dexterity in manoeuvring and stability. First the general mathematical models are derived for the single modules composing a car with different level of approximation; then the preliminary architecture and geometry mass parameters of PICAV have been used in the simulation to explore its behaviour and results are shown and commented. Further a simplified model in terms of state variables is proposed: it is useful as a base for the driving assistant set up [12] and new control modes.

The most simplified models represent the knowledge base of the PICAV dynamics used within the discrete event simulation package, following the same microsimulation approach used in [13], developed as a detailed analysis tool of the PICAV system management, described in [3] and [4].

The simulation results addressed many ideas to exploit the actuation redundancy of four powered wheels to accomplish standard manoeuvres by directly controlling the torque and the speed actually supplied by each tyre-soil interaction. The actuation redundancy might also be exploited to have higher driving stability even when the road conditions vary abruptly and this conditions are typical of the PICAV environments.

PICAV is designed to be driven also by weak people in semi-autonomous way under the guide of the driver assistant module and, in autonomous way in special cases, such as the re-allocation of the vehicle fleet to the parking lots, during the night. Off line and on line control logics are then needed in order to fulfill the requirements on manoeuvrability and stability. For this reason the dynamic detailed model of the vehicle plays a unique important role allowing the definition and set up of the mechatronic system also in virtual critical conditions. The paper concerns the contribution of the authors to these aspects.

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