Multiple-wheel all-terrain vehicles without a steering system must use great amounts of power when skid steering. Skid steering is modeled with emphasis put on the ground contact forces of the wheels according to the mass distribution of the vehicle. To increase steering efficiency, it is possible to modify the distribution of the normal contact forces on the wheels. This paper focuses on two aspects: first, it provides a model and an experimental study of skid steering on an all-road 6×6 electric wheelchair, the Kokoon mobile platform. Second, it studies two configurations of the distribution of the normal forces on the six wheels, obtained via suspension adjustments. This was both modeled and experimented. Contact forces were measured with a six-component force plate. The first results show that skid steering can be substantially improved by only minor adjustments to the suspensions. This setting decreases the required longitudinal forces applied by the engines and improves the steering ability of the vehicle or robot. Skid-steering characteristic parameters, such as the position of the center of rotation and absorbed skid power, are also dealt with in this paper. © 2010 Wiley Periodicals, Inc.

1. INTRODUCTION

This paper presents a model and experimental results of skid steering with a 6×6 all-terrain vehicle (ATV). A correct understanding of the phenomena that occur during steering would allow the modeling of the contact forces during skid steering with this vehicle and propose adjustments to improve steering capabilities and decrease energy loss due to friction.

Skid is a phenomenon that appears with every type of ground vehicle when the external forces applied to the vehicle exceed the capabilities of the vehicle–ground interface (Kecceci & Tao, 2006). Skid may be due to longitudinal inertial forces when accelerating/braking or to lateral inertial forces when steering at high speed and low radius. It may also be due to the design of the vehicle.

Skid always appears with tracked vehicles during turns, even if some of them have front steering tracks (Watanabe, Kitano, & Fugishima, 1995) because the long contact surface of the track with the ground requires a given torque to steer. Conversely, wheels ensure a reduced contact surface on a plane ground: a point contact with toroidal tires such as motorbike tires and a linear contact with cylindrical tires such as those used for cars. In reality, because of tire deformation, the contact point or contact line becomes a contact patch and a moderate steering torque may be noted. However, wheels give excellent steering capability while maintaining ground contact. For both tracks and wheels, grip strongly depends on the normal force values and distribution (Mokhiamar & Abe, 2006).

The large majority of wheeled vehicles have steering wheels, which can be the front wheels on classical cars; the rear wheels on power lift trucks or lawn mowers (Besselink, 2003, 2004); all the wheels on some types of mobile robots and sport cars (Shoichi, Yoshimi, & Yutaka, 1986); two front and two rear wheels of six (FNSS Corp., 2008) or four front wheels of eight on military wheeled armored vehicles (Patra Corp., 2008) or truck-mounted cranes. The steering mechanism may be complex, particularly when there are more than two steering wheels. The initial constraint is to respect the Ackermann steering geometry (1817), also known as Jeantaud geometry (1851) in Europe, that minimizes skid during low-speed turns. This condition requires that all wheels share the same center of rotation in every position. However, vehicles with more than two axes generally do not completely respect Ackermann geometry (Figure 1). As an example, a semitrailer does not respect Ackermann geometry and the three fixed rear axles generate severe wear of the tires. The second constraint is that the steering system must be compatible with other functions such as transmission and suspension. This increases mechanical complexity. Another drawback of architectures with steering wheels is that they generally do not allow the rotation of the vehicle on itself (null turning radius). For instance, with two steering wheels, this would require a high
steering angle, which is technically complex to design and dangerous at high speeds.

For this reason, many ATVs still rely on fixed wheels with no steering mechanism and with an optional suspension system (Figure 2). These vehicles have a robust and reliable behavior on rough terrain. Most of them have a 4×4 transmission, such as the Pioneer3-AT robot (Robosoft Corp., 2009), and some have a 6×6 one, such as multipurpose amphibian vehicles (Oasis LLC Corp., 2009). They must turn by skid steering and behave like tracked vehicles (Maclaurin, 2007). During skid steering, the wheels that are not tangent to the curved trajectory have to skid laterally, which generates friction forces that are opposed to the rotation.

The purpose of this work is to model and experiment skid steering in a 6×6 configuration. This paper also explores a solution to reduce energy loss during skid steering. Although lateral friction is a well-known problem of such types of vehicles with nondirectional wheels, it appears that very few studies have tried to reduce lateral friction forces. Most research is focused on the improvement of longitudinal adherence to improve traction and occasionally stability on rough terrain, such as the work on the Gofor Mars exploration robot done by Sreenivasan and Wilcox (1994). Reducing steering friction forces could enhance the interest in this class of simple, robust, and inexpensive vehicles.

2. DESCRIPTION OF THE 6×6 MOBILE PLATFORM

The Kokoon mobile platform is an all-road 6×6 electric wheelchair (Fauroux, Charlat, & Limenitakis, 2004a) designed by several students of the French Institute for Advanced Mechanics (IFMA) from 1999 onward (Figure 3). Kokoon is made of a modular aluminum frame on which are fixed composite body panels (removed during experiments). Kokoon is driven by two direct-current, permanent-magnet electric motors of 1,330 W each (Motovario 24V) and is capable of moving at 8 km/h on 20% slopes and of climbing easily over 15-cm obstacles. Two
lead-acid batteries (Hawker-Oldham 12 V, 160 Ah, 70 kg each) ensure 4 h of autonomy. Each motor is controlled by a speed controller (Curtis 1227) that allows current peaks of around 200 A. The driver interface is either a joystick or separate levers, one for each side.

Kokoon is 175 cm long and 103 cm wide, and it is equipped with six wheels of 20-cm radius (denoted \( r \)) and with 7-cm-wide air-inflated tires (Figure 4). The average wheelbase is denoted \( e \) and measures 47 cm, but this may undergo change when the suspension is compressed. The track width denoted \( 2v \) is 93 cm long. This is supposed to be a fixed value, although the lateral deflection of the suspension arms when skid steering may be as high as 2 cm. However, as the suspension arms of the same axle have approximately the same deflection during steering, track width may be considered constant.

Each motor drives synchronously the three wheels of one side thanks to a belt transmission using six pulleys and five belts (Figure 5). The motor is directly connected to pulley \( P_2 \) by a clutching system not represented in the figure. Belts \( B_{12} \) and \( B_{23} \) transmit the driving torque to front pulley \( P_1 \) and rear pulley \( P_3 \), respectively. Belts \( B_1, B_2, B_3 \) are located on the three independent swing arms and drive the power to the last pulleys \( P_{1w}, P_{2w}, P_{3w} \) that are linked to the wheels. For kinematic compatibility of transmission movement with suspension movements, pulleys \( P_i \) are mounted...
Figure 6. Adjustable swing-arm suspension with oleo-pneumatic shock absorber.

free and coaxial on swing-arm axes. It should be noted that a suspension movement generates an additional coupling torque on the wheel. However, this phenomenon did not appear in our case as the experiments were made on flat ground at constant speed. Pulley-belt transmissions require careful belt tension for proper operation.

The six independent swing-arm suspensions use oleo-pneumatic shock absorbers (Figure 6). They are easily adjustable thanks to the T-slots on the sides of the aluminum profiles. The top end of the shock absorber is named $T$ and can be longitudinally translated along the T-slots or vertically elevated by spacers. The shock absorbers (Fournales, Inc.) are designed to be inflated at 10 bars using an air pump. Adjusting pressure alters both the preconstraint and the stiffness.

The first field tests with Kokoon showed excellent climbing abilities but some steering difficulties. Even with small tires (tire tread width 7 cm), the vehicle could not steer on itself on highly adherent grounds such as tarmac. In this case, steering was still possible with high turning radii and a nonnull longitudinal speed. On less adherent grounds such as grass or tiled floor, the vehicle could easily turn on itself.

Initially designed for disabled people, Kokoon is an interesting research platform because it has a modular design and can be easily reconfigured (Fauroux, Charlat, & Limenitakis, 2004b). Parameters such as transmission, suspension geometry, and mass distribution can be adjusted rapidly. As mentioned above, this 6×6 vehicle shows diverse behaviors during skid steering according to the type of ground. This phenomenon is studied in detail in the following sections, and the results obtained below are easily transposable to comparable vehicles and mobile robots with three or more axles.

3. MODELING SKID STEERING

Dynamic modeling of four-wheel vehicles often uses an “equivalent” bicycle model that assumes that the internal and external wheels of each axle are combined into a single one. This assumption is acceptable provided that the radius of gyration is large enough and the slip angle small. Apart from the fact that Kokoon has three axles instead of two, these hypotheses could not be made during the experiments performed with the vehicle as the turning radius was small and slip angles were high.

3.1. Symmetrical Skid-Steering Model

A preliminary planar model of the Kokoon platform including six distinct wheels is shown in Figure 7 (Mendonca & Nait Hadi, 2007), where the vehicle is represented during a turn of radius $R$ and center $O$ (see Table I). $C$ is the central reference point of the vehicle, located in the middle of the second axle. The vehicle has a local reference frame $(C, X_C, Y_C, Z_C)$ with $X_C$ directed forward, $Z_C$ directed upward, and $Y_C$ to the left so that the frame is direct. The centers of external wheels move at speed $V_{sa}$, with index $s$ denoting the side of the vehicle ($e$ for “external” and $i$ for “internal”) and index $a$ standing for the axle number (1, 2, or 3 from front to rear).

Figure 7. Top view of skid-steering model with symmetrical front and rear slip angles.
### Table I. Symbols.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Type</th>
<th>Meaning</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha_l$</td>
<td>Scalar</td>
<td>Limit of slip angle after which the lateral force $F_y$ stops growing linearly</td>
</tr>
<tr>
<td>$\alpha_{sa}$</td>
<td>Scalar</td>
<td>Slip angle of the wheel located on side $s$ and axle $a$</td>
</tr>
<tr>
<td>$\beta_{a0}$</td>
<td>Scalar</td>
<td>Slip angle of the wheel located on side $s$ and axle $a$ for a null turn radius $R$</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>Scalar</td>
<td>Gyration angle of the vehicle around axis $(O, Z_O)$</td>
</tr>
<tr>
<td>$C$</td>
<td>Point</td>
<td>Center of the middle axle, reference point of the vehicle</td>
</tr>
<tr>
<td>$C_{sa}$</td>
<td>Point</td>
<td>Center of the wheel located on side $s$ and axle $a$</td>
</tr>
<tr>
<td>$\Delta$</td>
<td>Scalar</td>
<td>Decrement of the unloaded length of the front and rear suspension springs</td>
</tr>
<tr>
<td>$e$</td>
<td>Scalar</td>
<td>Wheelbase of the vehicle (longitudinal distance between consecutive axles)</td>
</tr>
<tr>
<td>$F_{sa}$</td>
<td>Vector</td>
<td>Force of the wheel-center point $C_{sa}$</td>
</tr>
<tr>
<td>$F_{sa}$</td>
<td>Scalar</td>
<td>Longitudinal force applied on the wheel-center point $C_{sa}$</td>
</tr>
<tr>
<td>$F_{Max}$</td>
<td>Scalar</td>
<td>Maximal lateral force corresponding to value $\alpha_l$</td>
</tr>
<tr>
<td>$F_{sa}$</td>
<td>Scalar</td>
<td>Lateral force applied on the wheel-center point $C_{sa}$</td>
</tr>
<tr>
<td>$G$</td>
<td>Point</td>
<td>Center of gravity of the vehicle</td>
</tr>
<tr>
<td>$g$</td>
<td>Scalar</td>
<td>Acceleration of gravity</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>Scalar</td>
<td>Pitch angle of the vehicle</td>
</tr>
<tr>
<td>$h_S$</td>
<td>Scalar</td>
<td>Adjustable distance on suspension that allows change of behavior</td>
</tr>
<tr>
<td>$h_S^{\text{std}}$</td>
<td>Scalar</td>
<td>Standard value of $h_S$</td>
</tr>
<tr>
<td>$h_S^{\text{mod}}$</td>
<td>Scalar</td>
<td>Modified value of $h_S$</td>
</tr>
<tr>
<td>$K$</td>
<td>Point</td>
<td>Projection of the center of gyration $O$ on the sagittal plane $(C, X_C)$ of the vehicle</td>
</tr>
<tr>
<td>$k$</td>
<td>Scalar</td>
<td>Stiffness of suspension springs</td>
</tr>
<tr>
<td>$l_{a0}$</td>
<td>Scalar</td>
<td>Unloaded length of the suspension spring of axle $a$</td>
</tr>
<tr>
<td>$l_a$</td>
<td>Scalar</td>
<td>Loaded length of the suspension spring of axle $a$</td>
</tr>
<tr>
<td>$l_a'$</td>
<td>Scalar</td>
<td>Loaded length of the modified suspension spring of axle $a$</td>
</tr>
<tr>
<td>$l_{a0}'$</td>
<td>Scalar</td>
<td>Unloaded length of the modified suspension spring of axle $a$</td>
</tr>
<tr>
<td>$l_0$</td>
<td>Scalar</td>
<td>Standard unloaded length of a suspension spring</td>
</tr>
<tr>
<td>$m$</td>
<td>Scalar</td>
<td>Mass of the vehicle</td>
</tr>
<tr>
<td>$O$</td>
<td>Point</td>
<td>Center of gyration of the vehicle during the turn</td>
</tr>
<tr>
<td>$P_{skid}$</td>
<td>Scalar</td>
<td>Power absorbed by the skid-steering process</td>
</tr>
<tr>
<td>$R$</td>
<td>Scalar</td>
<td>Gyration radius of vehicle during the turn</td>
</tr>
<tr>
<td>$R_x, R_y, R_z$</td>
<td>Scalar</td>
<td>Components of the reaction forces measured on the force plate</td>
</tr>
<tr>
<td>$r$</td>
<td>Scalar</td>
<td>Wheel radius</td>
</tr>
<tr>
<td>$T_a$</td>
<td>Point</td>
<td>Top fixation point of the suspension spring of axle $a$</td>
</tr>
<tr>
<td>$\tau_i$</td>
<td>Scalar</td>
<td>Torques of motors (internal or external according to $s$ value)</td>
</tr>
<tr>
<td>$v$</td>
<td>Scalar</td>
<td>Half track width of the vehicle</td>
</tr>
<tr>
<td>$V_{sa}$</td>
<td>Vector</td>
<td>Speed of the wheel-center point $C_{sa}$</td>
</tr>
<tr>
<td>$V_{sa}$</td>
<td>Scalar</td>
<td>Longitudinal speed of the wheel-center point $C_{sa}$</td>
</tr>
<tr>
<td>$V_{sa}$</td>
<td>Scalar</td>
<td>Lateral speed of the wheel-center point $C_{sa}$</td>
</tr>
<tr>
<td>$x_a$</td>
<td>Scalar</td>
<td>Longitudinal position of axle $a$ with respect to point $K$</td>
</tr>
<tr>
<td>$x_{COP}, y_{COP}$</td>
<td>Scalar</td>
<td>Coordinates of the center of pressure on top of the force plate</td>
</tr>
<tr>
<td>$x_G$</td>
<td>Scalar</td>
<td>Longitudinal position of the center of gravity $G$ relative to $(C, X_C)$</td>
</tr>
<tr>
<td>$x_K$</td>
<td>Scalar</td>
<td>Longitudinal position of the projected center of gyration $K$ relative to $(C, X_C)$</td>
</tr>
</tbody>
</table>

### Indices
- $e$: External side of the turn (where the circle is bigger)
- $i$: Internal side of the turn (where the circle is smaller)
- $s$: Side of the turn: can be $e$ for external or $i$ for internal
- $a$: Axle number: can be 1 for front, 2 for middle, 3 for rear

### Frames
- $R_O(0, X_D, Y_D, Z_O)$: Global frame connected to the ground
- $R_C(C, X_C, Y_C, Z_C)$: Local frame connected to the vehicle $(X_C$ forward, $Z_C$ ascending)
- $R_P(C, X_P, Y_P, Z_P)$: Frame connected to the force plate $(X_P$ forward, $Z_P$ ascending)
The second case is when:

\[ \alpha_{e1} = \text{atan}\left(\frac{e}{R + v}\right) = -\alpha_{e3}, \]

\[ \alpha_{i1} = \text{atan}\left(\frac{e}{R - v}\right) = -\alpha_{i3}. \]  

(1)

Table II gives the values of the slip angles for the Kokoon platform. It can be noted that when the gyration radius \( R \) decreases, the slip angles increase, and consequently the lateral forces and the energy required to turn also increase. Figure 8 represents the graph of the slip angles \( \alpha_{e1} \) and \( \alpha_{i1} \) with respect to the gyration radius \( R \), with fixed values of \( e = 0.47 \text{ m} \) and \( v = 0.93 \text{ m} \).

It can be noted that each slip angle has a different extremum (2). Two cases must be distinguished:

\[ \alpha_{e1} = \pm 90 \text{ deg} = -\alpha_{e3}, \quad \text{when} \quad R = v, \]

\[ \alpha_{i1} = \pm \text{atan}\left(\frac{e}{v}\right) = -\alpha_{i3}, \quad \text{when} \quad R = 0. \]  

(2)

- The first case is when \( R = v \). This forces the internal wheels to move orthogonally to their usual rolling direction, which is the worst-case scenario. At the same time, the external wheels are submitted to a high (but not extreme) slip angle.
- The second case is when \( R = 0 \). The vehicle self-rotates around point \( C \). The front and rear slip angles are denoted \( \alpha_{e0} \) and \( \alpha_{i0} \). They are very high and depend solely on the vehicle geometry:

\[ \alpha_{e0} = \text{atan}\left(\frac{e}{v}\right) = -\alpha_{e30}, \]

\[ \alpha_{i0} = \frac{\pi}{2} + \text{atan}\left(\frac{v}{e}\right) = -\alpha_{i30}. \]  

(3)

The absolute value of \( \alpha_{e10} \) increases with the wheelbase \( e \) and decreases with the half-track width \( v \). It is the contrary for \( \alpha_{i10} \). In the case of Kokoon, as \( e \) and \( v \) have similar values, the slip angles \( \alpha_{e10} \) reach around \( \pm 45 \text{ deg} \), which is extremely high and largely over the classical values measured for cars.

### 3.2. Nonsymmetrical Skid-Steering Model

The former symmetrical model would be perfectly acceptable with a balanced vehicle having its center of gravity \( G \) located at the center \( C \) of the vehicle. As this is rarely the case, a more realistic nonsymmetrical skid-steering model has recently been introduced (Mousset & Chervet, 2008).

It is important to keep in mind that the vehicle can turn because of the longitudinal and lateral forces. Longitudinal forces are provided by the engines, whereas lateral forces result from friction when skid steering. The relation between lateral forces \( F_y \) and slip angle \( \alpha \) is quasi-linear up to a limit value \( \alpha_l \), as represented in Figure 9 (Halconruy, 1995). The angle \( \alpha_l \) is of the order of 10 deg for a typical car tire. Above this threshold, there is a transition zone and the tire starts to slip on the ground. So with identical front and rear slip angles, the vehicle should be submitted to equivalent front and rear lateral forces and the front and rear slip angles should remain equal throughout the whole process.

But Figure 9 also shows that the lateral force \( F_y \) depends on the normal force \( F_z \): an increase of \( F_z \) generates an increase of \( F_y \). With a nonbalanced vehicle, the load on each axle is no longer identical and the lateral forces vary accordingly. For instance, a vehicle that is heavier on axle.

### Table II. Kokoon slip angles computed from a symmetrical skid-steering model.

<table>
<thead>
<tr>
<th>Gyration radius ( R ) (m)</th>
<th>Front external slip angle ( \alpha_{e1} ) (deg)</th>
<th>Front internal slip angle ( \alpha_{i1} ) (deg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>45.3</td>
<td>-45.3</td>
</tr>
<tr>
<td>0.47 (( = v ))</td>
<td>26.7</td>
<td>-90/+/90</td>
</tr>
<tr>
<td>3</td>
<td>7.7</td>
<td>10.5</td>
</tr>
<tr>
<td>6</td>
<td>4.2</td>
<td>4.8</td>
</tr>
</tbody>
</table>

The graph of the slip angles \( \alpha_{e1} \) and \( \alpha_{i1} \) as a function of the turning radius \( R \). A singularity appears when \( R = v \).
The exact value of the longitudinal forces applied on the ground to the vehicle are represented by vectors applied by the engine of the propulsion force generated because of slip angles (cf. Figure 9) on rugged ground. Consequently, the only available information on the longitudinal forces is that they obey Eqs. (4):

\[
F_{sx1} + F_{sx2} + F_{sx3} = \tau_e, \\
F_{xi1} + F_{xi2} + F_{xi3} = \tau_i,
\]

where \(\tau_e\) and \(\tau_i\) are the torques of the external and internal motors, respectively, and \(r\) is the wheel radius.

Concerning the lateral reaction forces, and assuming that no slip angle is over \(\alpha_i\),

- The lateral internal forces \(F_{xia}\) should be higher with respect to \(F_{xia}\) than the lateral external forces \(F_{yxa}\) because the slip angles are always larger on the internal side.
- The lateral forces ahead of point \(K\) are directed to the outside of the turn. Those behind point \(K\) are directed to the inside of the turn. For a given vertical load on all the wheels, the higher the longitudinal distance between \(K\) and \(Csa\) projected along \(XC\), the higher the slip angle and the higher the lateral forces \(F_{yxa}\).

In Figure 10, the speeds \(V_{sa}\) are represented on the top position of the vehicle with a magnitude that is proportional to the distance between \(O\) and the considered wheel center \(Csa\). Generally, this means that external wheels turn faster than internal ones. Even on the same side, each wheel center \(Csa\) moves around a separate circle. The speed vectors \(V_{sa}\) are constructed tangent to the circular trajectory of \(Csa\), with the following components:

- The longitudinal speeds \(V_{xsa}\) are oriented to the front of the vehicle.
- The lateral speeds are denoted \(V_{yxa}\). Each time \(V_{yxa}\) is nonnull, there is lateral slipping of the wheel, which generates a lateral force in the opposite direction.

The slip angles \(\alpha_{sa}\) are obtained via Eq. (5) as a function of \(s_a\), which is the longitudinal distance between axle \(a\) and the projected gyration point \(K\). The value of \(s_a\) depends on wheelbase \(e\) and on the longitudinal position \(x_K\) of point \(K\) relative to \((C, XC)\), which is negative in Figure 10:

\[
\begin{align*}
\alpha_{ea} &= \text{atan} \left( \frac{s_a}{R + v} \right) \\
\alpha_{ia} &= \text{atan} \left( \frac{s_a}{R - v} \right)
\end{align*}
\]

with \(s_a \in \{ -x_K, x_K, -x_K, -e - x_K \} \).

Concerning the longitudinal reaction forces:

- The longitudinal internal forces \(F_{xia}\) must be smaller than the longitudinal external forces \(F_{yxa}\) in order to generate a positive torque around the axis \((K, ZC)\). \(F_{xia}\) may even be negative if necessary.
- The exact value of the longitudinal forces applied on each axle is unknown because it depends on the local contact properties. Indeed, the great advantage of synchronous propulsion of all the wheels on the same side is that the engine torque is automatically distributed where the contact is best: i.e., if one wheel is on slippery ground, the torque is distributed to the two others. This principle is similar to differential locking on an all-terrain car, which proves to be extremely efficient on rugged ground. Consequently, the only available information on the longitudinal forces is that they obey Eqs. (4):

\[
F_{sx1} + F_{sx2} + F_{sx3} = \tau_e, \\
F_{xi1} + F_{xi2} + F_{xi3} = \tau_i,
\]

where \(\tau_e\) and \(\tau_i\) are the torques of the external and internal motors, respectively, and \(r\) is the wheel radius.

Concerning the lateral reaction forces, and assuming that no slip angle is over \(\alpha_i\),

- The lateral internal forces \(F_{xia}\) should be higher with respect to \(F_{xia}\) than the lateral external forces \(F_{yxa}\) because the slip angles are always larger on the internal side.
- The lateral forces ahead of point \(K\) are directed to the outside of the turn. Those behind point \(K\) are directed to the inside of the turn. For a given vertical load on all the wheels, the higher the longitudinal distance between \(K\) and \(Csa\) projected along \(XC\), the higher the slip angle and the higher the lateral forces \(F_{yxa}\).

In Figure 10, the speeds \(V_{sa}\) are represented on the top position of the vehicle with a magnitude that is proportional to the distance between \(O\) and the considered wheel center \(Csa\). Generally, this means that external wheels turn faster than internal ones. Even on the same side, each wheel center \(Csa\) moves around a separate circle. The speed vectors \(V_{sa}\) are constructed tangent to the circular trajectory of \(Csa\), with the following components:

- The longitudinal speeds \(V_{xsa}\) are oriented to the front of the vehicle.
- The lateral speeds are denoted \(V_{yxa}\). Each time \(V_{yxa}\) is nonnull, there is lateral slipping of the wheel, which generates a lateral force in the opposite direction.

The slip angles \(\alpha_{sa}\) are obtained via Eq. (5) as a function of \(s_a\), which is the longitudinal distance between axle \(a\) and the projected gyration point \(K\). The value of \(s_a\) depends on wheelbase \(e\) and on the longitudinal position \(x_K\) of point \(K\) relative to \((C, XC)\), which is negative in Figure 10:

\[
\begin{align*}
\alpha_{ea} &= \text{atan} \left( \frac{s_a}{R + v} \right) \\
\alpha_{ia} &= \text{atan} \left( \frac{s_a}{R - v} \right)
\end{align*}
\]

with \(s_a \in \{ -x_K, x_K, -x_K, -e - x_K \} \).

Concerning the longitudinal reaction forces:

- The longitudinal internal forces \(F_{xia}\) must be smaller than the longitudinal external forces \(F_{yxa}\) in order to generate a positive torque around the axis \((K, ZC)\). \(F_{xia}\) may even be negative if necessary.
- The exact value of the longitudinal forces applied on each axle is unknown because it depends on the local contact properties. Indeed, the great advantage of synchronous propulsion of all the wheels on the same side is that the engine torque is automatically distributed where the contact is best: i.e., if one wheel is on slippery ground, the torque is distributed to the two others. This principle is similar to differential locking on an all-terrain car, which proves to be extremely efficient on rugged ground. Consequently, the only available information on the longitudinal forces is that they obey Eqs. (4):

\[
F_{sx1} + F_{sx2} + F_{sx3} = \tau_e, \\
F_{xi1} + F_{xi2} + F_{xi3} = \tau_i,
\]

where \(\tau_e\) and \(\tau_i\) are the torques of the external and internal motors, respectively, and \(r\) is the wheel radius.
3.3. Skid-Steering Modeling

Assuming that the skid-steering vehicle has a constant rotation speed, the fundamental principle of dynamics can be applied with a null rotational acceleration around axis $(O, Z_O)$.

If the friction in the transmission is initially ignored, it can be interpreted in the following way: the steering torque generated by the longitudinal forces created by the motors is used to compensate exactly for the resisting torque created by the slipping lateral forces. This results in Eqs. (6)–(7):

$$M_{O,F_{\text{in}}} + M_{O,F_{\text{sia}}} = 0, \quad (6)$$

with $x_a \in \{e-x_K, -x_K, -e-x_K\}$.

Equation (4) allows us to replace longitudinal forces by motor torques and to obtain

$$\left[ \sum_{a=1}^{a=3} F_{\text{xe}a} \cdot (R + v) + \sum_{a=1}^{a=3} F_{\text{xia}} \cdot (R - v) \right] + \left[ \sum_{a=1}^{a=3} (F_{\text{ye}a} + F_{\text{yia}}) \cdot x_a \right] = 0, \quad (7)$$

with $x_a \in \{e-x_K, -x_K, -e-x_K\}$.
Equation (8) governs the skid-steering behavior and may help to characterize it, provided that sufficient data are gathered from experiments.

In Section 4, we present an original solution to reduce the friction forces $F_{ysa}$ during skid steering. By decreasing the absolute value of the second term in Eq. (8), which represents the skid-steering resisting torque, it appears that the driving torque represented by the first term will simultaneously decrease as an absolute value. This could be an advantageous improvement on this class of vehicles.

Sections 5 and 6 present the experimental part of this work. Field tests on the Kokoon vehicle intend to confirm the nonsymmetrical skid-steering model presented above.

4. IMPROVING SKID STEERING BY SUSPENSION ADJUSTMENT

To improve the steering efficiency of multiaxle vehicles such as Kokoon, one solution could be to modify the mass distribution. This could be done by physically adding mass or moving existing components and payload in the frame.

Another, simpler solution is to modify the suspension characteristics. In this section, we choose to modify the front and rear suspensions with respect to the central suspension. Our purpose is to increase the vertical load on the central axle, which is equivalent to unloading the front and rear axles.

4.1. Model of the Vehicle with Standard Suspensions

To demonstrate this phenomenon, a simplified model of the vehicle is considered, with the frame supported by three suspensions, each one being represented by a vertical spring with linear behavior [Figure 11(a)]. When the vehicle is put on the ground, it finds a new equilibrium [Figure 11(b)]. The initial unloaded length $l_{a0}$ of each spring changes to a new loaded value $l_{a}$. The three unknown values of $l_{a}$ can be found by solving a set of three equations.

The first equation comes from the fact that the top fixation points $T_a$ of the springs remain aligned at the bottom of the frame, which is assumed to be nondeformable. The wheel centers $C_a$ also remain aligned because the ground is flat. The angle between both lines is the pitch angle $\gamma$ of the vehicle and relation (9) can therefore be written

$$\tan(\gamma) = \frac{l_1 - l_2}{e} = \frac{l_2 - l_3}{e},$$

which simplifies into

$$2l_2 = l_1 + l_3.$$  (10)

As the vehicle is balanced, the fundamental principle of statics give the two other equations. The sum of the reaction forces $F_{za}$ and the weight $mg$ gives relation (11):

$$\sum F = F_{z1} + F_{z2} + F_{z3} - mg = 0.$$  (11)

The sum of the moments reduced to the central point $C$ of the vehicle gives relation (12):

$$\sum MC = -eF_{z1} + eF_{z3} + mgx_G \cos(\gamma) = 0.$$  (12)

with $x_G$ the abscissa of $G$ within the local frame $(C, X_C, Y_C)$, negative in Figure 11(b). Each spring has a linear relationship between its force $F_{za}$ and displacement $(l_{a0} - l_a)$ via constant stiffness $k$ (13):

$$F_{za} = k(l_{a0} - l_a).$$  (13)

Relation (13) is used to replace unknown forces $F_{za}$ in relation (11) to obtain

$$k(l_{10} - l_1) + k(l_{20} - l_2) + k(l_{30} - l_3) = mg.$$  (14)

Relation (13) is also used in relation (12) to obtain

$$-ke(l_{10} - l_1) + ke(l_{30} - l_3) + mgx_G \cos(\gamma) = 0.$$  (15)

Equations (10), (14), and (15) form a system of three equations and three unknowns $l_a$ that can be solved to find the equilibrium position of the vehicle. First, Eq. (14) can be simplified into Eq. (16):

$$l_1 + l_2 + l_3 = l_{10} + l_{20} + l_{30} - \frac{mg}{k}.$$  (16)
Then $l_2$ can be extracted from Eq. (16) with the help of Eq. (10):
\[
l_2 = \frac{l_{10} + l_{20} + l_{30}}{3} - \frac{mg}{3k} = \frac{mg}{3k}. \tag{17}
\]

On the other hand, Eq. (15) is reformulated into Eq. (18) provided that $\gamma$ remains small enough:
\[
l_1 - l_3 = l_{10} - l_{30} - \frac{mg x_G}{k e}. \tag{18}
\]

Equations (16) and (17) are condensed into Eq. (19):
\[
l_1 + l_3 = \frac{2(l_{10} + l_{20} + l_{30})}{3} - \frac{2mg}{3k}. \tag{19}
\]

Equations (18) and (19) form a system of two equations with two unknowns ($l_1, l_3$) that are extracted:
\[
\begin{align*}
l_1 &= \frac{5}{6}l_{10} + \frac{1}{3}l_{20} - \frac{1}{6}l_{30} - \frac{mg}{k} \left( \frac{1}{3} + \frac{x_G}{2e} \right), \tag{20} \\
l_3 &= -\frac{1}{6}l_{10} + \frac{1}{3}l_{20} + \frac{5}{6}l_{30} - \frac{mg}{k} \left( \frac{1}{3} - \frac{x_G}{2e} \right). \tag{21}
\end{align*}
\]

If all the unloaded lengths $l_{a0}$ are assumed to be equal to the same reference length $l_0$, all the unknown lengths $l_a$ expressed in Eqs. (17), (20), and (21) can be simplified into
\[
\begin{align*}
l_1 &= l_0 - \frac{mg}{k} \left( \frac{1}{3} + \frac{x_G}{2e} \right) = l_2 - \frac{mg x_G}{2e}, \\
l_2 &= l_0 - \frac{mg}{3k}, \\
l_3 &= l_0 - \frac{mg}{k} \left( \frac{1}{3} - \frac{x_G}{2e} \right) = l_2 + \frac{mg x_G}{2e}. \tag{22}
\end{align*}
\]

It can be seen that because the center of gravity $G$ is located at the rear, $x_G$ is negative and
\[
l_0 \geq l_1 \geq l_2 \geq l_3 > 0. \tag{23}
\]

Using relation (13), the unknown forces $F_{za}$ can be deduced from the lengths $l_a$ found in Eqs. (22) and expressed in Eqs. (24):
\[
F_{z1} = mg \left( \frac{1}{3} + \frac{x_G}{2e} \right) = F_{z2} + mg \frac{x_G}{2e}. \tag{24}
\]

Expression (24) allows us to sort the contact forces by intensity on the standard vehicle, which appears to be overloaded at the rear [relation (25)], which is logical:
\[
F_{z3} \geq F_{z2} \geq F_{z1} > 0. \tag{25}
\]

As $F_{z3}$ is high, it generates a strong lateral ripping force $F_{z3}$ during skid steering.

### 4.2. Model of the Vehicle with Modified Suspensions

To improve skid steering, the front and rear normal forces must be lowered according to their initial value. This can be achieved by modifying the suspensions. For instance, the modified unloaded lengths $l’_{10}$ and $l’_{30}$ of the front and rear springs will be shortened by a given value $\Delta$ with respect to their original unloaded lengths $l_{10}$ and $l_{30}$ [Figure 12(a)]. After equilibrium is achieved, the unknown spring lengths are named $l’_a$ and the corresponding contact forces $F’_{za}$.

The modified unloaded spring lengths are enumerated in Eqs. (26):
\[
\begin{align*}
l’_{10} &= l_{10} - \Delta = l_0 - \Delta, \\
l’_{20} &= l_{20} = l_0, \\
l’_{30} &= l_{30} - \Delta = l_0 - \Delta. \tag{26}
\end{align*}
\]

Equations (17), (20), and (21) have simply to be reevaluated with the modified unloaded lengths obtained from Eqs. (26). The modified spring-loaded lengths $l’_a$ are then calculated and are presented in Eqs. (27):
\[
\begin{align*}
l’_1 &= l_1 - \frac{2}{3} \Delta, \\
l’_2 &= l_2 - \frac{2}{3} \Delta, \\
l’_3 &= l_3 - \frac{2}{3} \Delta. \tag{27}
\end{align*}
\]

![Figure 12.](image-url) Simplified model of a vehicle with modified suspensions. Front and rear springs have shortened unloaded length $l’_{a0}$.

---

*Journal of Field Robotics* DOI 10.1002/rob
It appears clearly that adjusting the suspension of value \( \Delta \) generates an identical compression of \( 2\Delta/3 \) of all the springs [Figure 12(b)]. This also means that the pitch angle \( \gamma \) remains unaltered. The normal contact forces are derived from Eqs. (13) and (27) and appear in Eqs. (28):

\[
\begin{align*}
F'_{z1} &= F_{z1} - k \frac{\Delta}{3}, \\
F'_{z2} &= F_{z2} + 2k \frac{\Delta}{3}, \\
F'_{z3} &= F_{z3} - k \frac{\Delta}{3}.
\end{align*}
\]

Although the sum of the forces is unchanged, the front and rear normal contact forces \( F'_{z1} \) and \( F'_{z3} \) are lowered of \( k\Delta/3 \). This generates a simultaneous lowering of the resisting lateral forces \( F'_{x1} \) and \( F'_{y3} \), thus facilitating the skid-steering process and justifying the type of adjustment made. It also allows us to find the required \( \Delta \) displacement for a given decrease in the normal contact forces \( F'_{z1} \) and \( F'_{z3} \).

5. EXPERIMENTAL SETTINGS

The steering process of a vehicle is a complex phenomenon that may be better understood from an experimental preliminary approach (Foster, Ayers, Lombardi-Przybylowicz, & Simmons, 2006; Itoh, Oida, & Yamazaki, 1995). For this 6×6 vehicle, it was decided to measure experimentally the contact forces of the wheels on the ground (Fauroux, Vaslin, & Douarre, 2007).

5.1. The Six-Component Force Plate

During Kokoon displacement, the wheels roll on the top plate of a six-component force plate (TSR, Mérignac, France), rigidly fixed in a wooden box buried in the ground so that the top plate is at ground level (Figure 13).

The six-component force plate used in this study (dimensions 60 × 80 cm; measurement ranges \( R_x = 1,000 \) N, \( R_y = 900 \) N, \( R_z = 2,000 \) N; resolution 10 N) is composed of a rigid composite top plate (carbon/aluminum) fixed on three two-component strain gauge force transducers \( T_i \) (Figure 14), which are also firmly fixed on the base plate (Couétard, 1993; 2000). Each of these transducers measures

![Figure 13.](image1)

**Figure 13.** (a) Placing the force plate in the ground. (b) Integration of the wooden box into the ground. (c) The entire experimental system.

![Figure 14.](image2)

**Figure 14.** The force plate including the three transducers to measure tangential and normal force components.
one component of the resultant force in the plane of the
top plate (“shearing” component \( R_x, R_y \)) and the other in
the direction perpendicular to the top plate (“compressive”
component \( R_z \)). The signal values \( V_i \) produced by the six
force transducers are multiplied by the 36 coefficients of
the sensitivity matrix \([S]\) for calculating the six components
\((R_x, R_y, R_z, M_x, M_y, M_z)\) of the wrench applied on the top
plate in the reference frame linked to the force plate:

\[
(R_x \ R_y \ R_z \ M_x \ M_y \ M_z)^T = [S](V_1 \ V_2 \ V_3 \ V_4 \ V_5 \ V_6)^T. \tag{29}
\]

In normal use, the wrench components allow computation
of the horizontal coordinates \((x_{\text{COP}}, y_{\text{COP}})\) of the point of
force application on top of the force plate, which is usually
called “center of pressure” (COP). The signals of the force-
plate transducers are simultaneously sampled at 100 Hz
by a 16-bit A/D conversion card (AT-MIO-16X; National
Instruments) slotted into a PC, and experimental data are
recorded using acquisition software (LabView 5.1; National
Instruments). The acquisition PC and signal conditioner are
brought close to the force plate [Figure 13(c)]. Further post-
processing is performed in a spreadsheet (Open Office).

5.2. Center of Gravity

The mass distribution on the wheels has a great influence
on the ground contact. It has been determined independ-
ently by the three following different methods with con-
sistent results.

The first method uses the computer-aided design (CAD)
model of Kokoon [Figure 15(a)]. Each component
is given a uniform density, and the Solid Edge CAD soft-
ware can evaluate the volume and calculate the weight
of the component. The whole assembly, including several
hundreds of parts, reaches a total weight of 367 kg without
the external composite panels.

The second method consists of carefully putting the ve-
hicle on six scales using a winch [Figure 15(b)]. The results
are summarized in Table III and are very close to those ob-
tained with the first method. The longitudinal position \(x_G\)
of the center of gravity \(G\) is also given relative to the middle
axle. \(G\) is located 133 mm behind the middle axle without
driver and only 78 mm behind with a 83-kg driver, for a
total weight of 450 kg. Consequently, the load is higher on
the rear axle and secondly on the middle axle, even if the
driver’s mass contributes to recentering point \(G\).

The third method relies on the force plate to measure
static loads on each wheel. The results are also consistent

<table>
<thead>
<tr>
<th>Table III. Mass distribution on the three axles and longitudinal position of the vehicle center of gravity.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Front axle (kg)</td>
</tr>
<tr>
<td>-----------------</td>
</tr>
<tr>
<td>Without driver</td>
</tr>
<tr>
<td>With a 83-kg driver</td>
</tr>
</tbody>
</table>
with the preceding ones and will be commented on later (Figures 19 and 21).

The results in Table III confirm Eqs. (24) and (25).

5.3. Modifying the Real Swing-Arm Suspension

The real swing-arm suspension mechanism is slightly more complex than the single spring used in Sections 4.1 and 4.2. It is represented in Figure 16, and its main dimensions are provided in Table IV. The shock absorber has an oleo-pneumatic structure and a nonlinear law between the spring force $F_s$ and the spring length $l_s$, with hysteresis. The swing-arm mechanism also adds nonlinearity to the evolution law of the normal contact force $F_{za}$ with respect to length $l_a$. This double nonlinearity poses no problem for this work as the vehicle runs only on horizontal ground, with quasi-null variations of the contact force $F_{za}$.

For reasons of simplicity, the modification consisted of decreasing length $h_T$. This was an easy operation because the top-attachment points $T$ of the front and rear shock absorbers can be adjusted in translation in their T-slot. Distance $h_T$ was reduced from $h_{T\text{std}} = 145$ mm to the minimal possible length $h_{T\text{mod}} = 45$ mm, which is a 100-mm motion that was performed only on the front and rear suspensions. The consequences of this modification were double:

- The unloaded length $l_a$ [Figure 16(a)] decreased to $l_a'$ [Figure 16(b)] with $\alpha \in \{1, 3\}$.
- The average stiffness also decreased because of the smaller moment arm of the spring force $F_s$ with respect to the rotation point $A$ of the swing arm.

Because of both simultaneous changes, the vertical forces $F_{za}$ on the front and rear wheels were noticeably lowered (Figure 17). As a consequence, the central axle was strongly overloaded and the central shock absorber underwent visible compression [Figure 17(b)].

This suspension adjustment is equivalent to change in mass distribution, as summarized in Table V. Front and rear

Table IV. Main dimensions of the suspension model.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$v_A$</td>
<td>180</td>
</tr>
<tr>
<td>$h_{T\text{std}}$</td>
<td>145</td>
</tr>
<tr>
<td>$h_{T\text{mod}}$</td>
<td>45</td>
</tr>
<tr>
<td>$l_s$</td>
<td>210–280</td>
</tr>
<tr>
<td>$l_B$</td>
<td>170</td>
</tr>
<tr>
<td>$l_C$</td>
<td>350</td>
</tr>
<tr>
<td>$d$</td>
<td>15</td>
</tr>
</tbody>
</table>

Figure 16. Model of the real swing-arm suspension mechanism (a) before and (b) after modification.

Figure 17. (a) Standard suspension configuration. (b) Modified configuration after adjustment.
Table V. Effects of suspension adjustments on the equivalent mass distribution (including an 83-kg driver).

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Front axle (kg)</th>
<th>Middle axle (kg)</th>
<th>Rear axle (kg)</th>
<th>Total (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Standard</td>
<td>108</td>
<td>158</td>
<td>183</td>
<td>450</td>
</tr>
<tr>
<td>Modified</td>
<td>80 (−26%)</td>
<td>250 (+58%)</td>
<td>120 (−34%)</td>
<td>450</td>
</tr>
</tbody>
</table>

Adjusting the suspensions was an extremely interesting option because of the small amount of work required and the significant changes generated. The following section presents the real experiments with and without suspension adjustments.

6. EXPERIMENTAL RESULTS

The experimental field can be seen in Figure 18. Lines showing the desired trajectories were drawn on the ground using flour. Three types of trajectories have been considered in this study: a straight line (which is equivalent to a turn with infinite radius), a turn with a 6-m radius, and a turn with a 3-m radius.

Several experiments were performed with the aim of following as closely as possible the desired trajectories (Douarre, 2006). Results are summarized in Figures 19–21.

The force-plate sample frequency was set at 100 Hz, which was sufficient for a slow vehicle such as Kokoon. Test duration varied according to the trajectory: driving in a straight line took generally no more than 1.5 s, whereas turning was slower and required up to 2.5 s because steering demanded greater power from the electric motors, and this power had to be adjusted on each side of the vehicle in real time by the driver if the correct path was to be followed.

Figure 18. Experimental field with reference frame and trajectories.

Figure 19. A typical graph obtained when the 6×6 vehicle drives over the force plate. The time axis can be divided into five intervals according to the number of wheels simultaneously present on the force plate.
Figure 20. Trace of the center of pressure for the modified vehicle and for three types of trajectories.

Figure 19 is a zoom of Figure 21(b) and shows the typical reaction forces applied to the vehicle when it drives over the platform. Because the vehicle wheelbase (47 cm) is smaller than the force-plate width along the rolling direction (60 cm), sometimes only one and sometimes two wheels may be on the force plate at the same time. This explains the shape of the curves of the reaction forces applied to the vehicle when it crossed the force plate (Figure 19). The time axis of each trial can be divided into five intervals:

1. First, only wheel 1 applies efforts on the force plate;
2. Then wheel 2 climbs onto the force plate (left transparent area), and the vertical component of reaction force $R_z$ increases suddenly;
3. After that, wheel 1 leaves the force plate and only wheel 2 remains on it;
4. Then, it is up to wheel 3 to cross the force plate (right transparent area) and a second peak on $R_z$ appears;
5. Finally, wheel 2 leaves the force plate and only wheel 3 remains on it until the end of the crossing.

Another type of experimental result is represented in Figure 20: the trace of the center of pressure on the force plate for various trajectories of the modified vehicle. Phases 1, 3, and 5 are represented by curve segments directed upward. Phases 2 and 4, during which two wheels are present at the same time on the force plate, are represented by a sudden downward inflexion of the curve. It is interesting to see that the center of pressure follows almost perfect circular arcs during phases 1–3–5. This confirms that the trajectory was correctly followed by the vehicle (see arrows in Figure 20).

Figure 21 gives typical results obtained with standard suspensions (panels a, b, c on the left) and modified suspensions (panels d, e, f on the right) for different values of the gyration radius $R$. Tables VI and VII are obtained by averaging the $R_x$, $R_y$, $R_z$ values on the single-wheel intervals and give an order of magnitude of the reaction components, thus eliminating the small variations in the signal due to electrical perturbations and vehicle vibrations on small pieces of gravel.

### 6.1. Reference 6×6 Vehicle with Standard Suspension

For normal force $R_z$ in standard configuration, it can be seen in Figure 21(a) and Table VI that wheel 3 (930 N) bears more weight than wheel 2 (848 N), which, in turn, bears more weight than wheel 1 (635 N). These results include the driver’s weight and confirm the previous calculations of the center of gravity. Assuming that the grip coefficient

| Figure 20. Trace of the center of pressure for the modified vehicle and for three types of trajectories. |
is identical on every wheel, this means that the rear and central wheels are able to apply a higher propulsion force $R_x$ and to undergo a higher lateral force $R_y$. When the trajectory varies [Figures 21(b) and 21(c)], the overall shape of the normal force $R_z$ does not differ considerably; the first peak is identical; the second peak changes slightly, probably due to transient phenomena.

Propulsion force $R_x$ has an original shape: it appears that only the central wheel applies a propulsion force. This may be caused by insufficient tension in the front and rear transmission belts. This problem must be corrected in future work because the central belts cannot transmit the complete torque; hence only one-third of the potential propulsion force is currently used by the vehicle. The planned solution is to replace belt transmission by chain transmission. Another interesting result is the evolution of $R_x$ with respect to the steering radius $R$: when $R$ decreases, $R_x$ must increase to make the vehicle rotate, as expected from Eq. (7). Along a straight line, $R_x$ does not need to be very high in order to generate vehicle movement. But during a turn with $R = 6\,\text{m}$ (respectively, 3 m), $R_x$ reaches 454 N (respectively, 553 N). This increase in $R_x$ force was clearly experienced by the driver, who needed to increase the power during short turns.

As expected, the lateral force $R_y$ has a negligible value when driving in a straight line. However, this value

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{forces.png}
\caption{Forces on the right wheels. (a) (b), and (c) For the standard suspension; (d) (e), and (f) for the modified suspension.}
\end{figure}
increases particularly on the front and rear wheels when the turning radius $R$ decreases. For instance, for the 3-m turn, $R_y$ reaches $-321$ N (respectively, $332$ N) on wheel 1 (respectively, wheel 3). The opposite signs of $R_y$ between wheels 1 and 3 logically reflect the opposite lateral efforts applied on these wheels during the turn. The absolute values of $R_y$ are not symmetrical on front and rear wheels. One explanation is that axle 3 loads more weight than axle 1. This is also the case for the 3-m turn, where $R_y$ is nonnull on the central wheel ($-146$ N). These results seem to confirm that the center of gyration of the vehicle is not located on the central axle, as predicted by the nonsymmetrical skid-steering model presented in Section 3.2.

### 6.2. Modified 6 × 6 Vehicle

With the modified vehicle, the duration of turn trials was only 2 s instead of 2.5 s with standard suspensions, because steering and power control were much easier for the vehicle and the driver, respectively. These improvements were clearly experienced by the driver during these trials. Results are summarized in Table VII.

In Figures 21(d)–21(f), it can be noted that the $R_y$ curve does not change a lot with the turning radius $R$. As expected on the modified vehicle, the highest normal load is supported by axle 2. During the straight-line trajectory (Figure 21(d)), the normal force $R_z$ shows a strange shape with a peak during Phase 3. This could be caused by the driver leaning forward or by the vehicle tilting on a small obstacle. Apart from this phenomenon, the total of normal forces on the external wheels remains constant at around 2,500 N (Figure 22). The total weight of the vehicle was supposed to be 4,500 N, as shown in Table III. This means that the external wheels bear approximately 56% of the weight of the vehicle, which was probably not perfectly horizontal, with a little roll angle. Figure 22(b) clearly shows the important part of the load borne by wheel 2 with the modified suspensions.

The curves of force $R_z$ still show that only the central wheel applies an effective propulsion force. However, the modified suspension seems to have decreased the required propulsion force. This could mean that a smaller longitudinal force is able to generate the same movement of the vehicle. For a 6-m turn, $R_x$ decreases from 454 to 282 N on the central wheel, which means a gain of 38% with respect to the initial suspension adjustment. The sum of the propulsive forces $R_x$ of all the external wheels is visible in Figure 23. The driver clearly needs to inject less energy into the electric motors. The force decrease is quantified at 43.5%, for radius 6 m as well as radius 3 m. The overall turning time was observed to be shorter than with a classical suspension. This suggests that the global turning efficiency is improved with the modified suspension. Further analysis is required to quantify this improvement.
The lateral force $R_y$ has approximately the same shape in the initial and adjusted configurations. For a 6-m-turn with the modified vehicle, there is a 28% decrease of $R_y$ on the front axle and a 36% decrease on the rear axle. This means that less energy is dissipated during skid steering with the modified suspensions.

6.3. Important Parameters to Characterize Skid Steering

From the skid-steering model and the corresponding experimental results, it appears that the nonsymmetrical model for skid steering on the Kokoon vehicle is verified. To quantify this nonsymmetry, it is important to find the projected center of gyration $K$ and its longitudinal position $x_K$.

$x_K$ becomes a characteristic parameter that can be extracted by first developing Eq. (8) into Eq. (30):

$$
\begin{align*}
\tau_e \cdot \frac{(R + v)}{r} + \tau_i \cdot \frac{(R - v)}{r} + (F_{y_1e} + F_{y_1i}) \cdot (e - x_K) \\
+ (F_{y_2e} + F_{y_2i}) \cdot (-x_K) \\
+ (F_{y_3e} + F_{y_3i}) \cdot (-e - x_K) = 0.
\end{align*}
$$

(30)

Then $x_K$ is factorized and extracted:

$$x_K = \frac{\left[\tau_e \cdot \frac{(R + v)}{r} + \tau_i \cdot \frac{(R - v)}{r} + e \cdot (F_{y_1e} + F_{y_1i}) - F_{y_2e} - F_{y_2i} - F_{y_3e} - F_{y_3i}\right]}{F_{y_1e} + F_{y_1i} + F_{y_2e} + F_{y_2i} + F_{y_3e} + F_{y_3i}}.
$$

(31)

However, obtaining $x_K$ theoretically from Eq. (31) is slightly awkward as the values of $F_{y_{ia}}$ are not precisely known and depend on the model used for slip angles. The best way would be to ascertain $x_K$ experimentally by making the vehicle turn on itself (with a zero turning radius) and finding the invariant point of its sagittal plane. This would require high-resolution photographs of the vehicle taken from the top and will be done in future work.

Section 5 introduced the concept of suspension modification in order to lower friction forces during skid steering. This decrease can be quantified by introducing the supplement of power $P_{skid}$ absorbed by skid steering:

$$P_{skid} = \left[\tau_e \cdot \frac{(R + v)}{r} + \tau_i \cdot \frac{(R - v)}{r}\right], \quad \text{or}
$$

$$P_{skid} = - \sum_{a=1}^{3}(F_{y_{ea}} + F_{y_{ia}}) \cdot x_a
$$

(32)

with $x_a \in \{e - x_K, -x_K, -e - x_K\}$. Equation (32) gives two ways of calculating $P_{skid}$:

**The second one is based on the knowledge of all the lateral forces. A simple solution would be to use a second force plate buried in the ground to obtain experimental results. But an even better solution would be to integrate force sensors into each wheel for continuous measurement. This would provide precise knowledge of the law $f$ that governs lateral forces $F_y$, as shown in Eq. (33) and Figure 9:

$$F_y = f(\alpha, F_z).
$$

(33)

As the slip angles $\alpha_{sa}$ are fixed for a given gyration radius $R$, as the mass distribution on the axles and the values of $F_z$ may be considered constant at uniform speed, the knowledge of this law would allow the prediction of the power consumption for skid steering. It requires, however, a careful fitting of the law to as many experiments as possible.

These preliminary results are very encouraging and confirm that turning efficiency is highly sensitive to the normal component $F_z$ of the contact force, as expected in Eq. (33). $F_z$ can be adjusted either by the mass distribution in the vehicle or by suspension adjustments. Changing mass distribution requires either adding ballast or moving sufficient masses inside the vehicle using a suitable mechanism. In both cases, the adjustment is not very practical and not suitable to selectively overload only the central axle of the vehicle, which is required to improve skid-steering ability. On the other hand, suspension adjustments can lead to the expected results with only simple modifications that may concern spring stiffness or the fixture position of the spring. Future work will have to confirm this result and to quantify it more precisely by using parameters $x_K$ and $P_{skid}$ first introduced here in Eqs. (31) and (32).

More generally, it now appears feasible to improve skid steering of many comparable vehicles by using this method. The main idea presented in this work is to reduce the energy loss due to friction during skid steering by overloading the axle that is closest to point $K$, whereas the other axles are underloaded proportionally to their distance from point $K$. Knowledge of parameter $x_K$ is therefore critical in adopting a correct strategy for adjusting the suspensions. The sum of normal forces $F_z$ remains constant but their distribution changes, and likewise for the lateral forces $F_y$. This allows the steering torque to decrease.

For vehicles with two axles, it appears to be difficult to improve skid steering with our method if the vehicle is balanced (i.e., with the same mass on the front and rear axles) as point $K$ is in the middle of the vehicle and no axle is
closer to \( K \). So it should be noted that this method is particularly suitable for vehicles and robots with three or more axles.

7. CONCLUSION

This work has presented models and experiments of the skid-steering phenomenon on a 6×6 vehicle. It characterized the lateral skid forces that are responsible for a high level of energy dissipation during steering. It also characterized the projected center of gyration \( K \) and its position relative to the vehicle. This center depends on the vehicle mass distribution as well as its propulsion and suspension systems. Preliminary results seem to confirm that point \( K \) is not located on the central axle of the vehicle. This suggests that the proposed nonsymmetrical skid-steering model is correct. Locating point \( K \) is thus a priority in order to achieve an efficient control of the vehicle or mobile robot during skid steering.

The experimental work also suggested that skid-steering efficiency of a 6×6 ATV can be substantially improved by only minor adjustments on the vehicle suspension. An important modification in contact force distribution was obtained with a 10-cm adjustment of damper fixtures. The driver reported that he felt a substantial improvement in the steering capacity during the trials performed with the modified vehicle. This minor adjustment of the suspensions allowed reduction of the propulsion forces by around 40% and also brought down the lateral forces in the same proportion.

The absence of any steering system on a vehicle is a guarantee of robustness and control simplicity, but it has the major drawback of consuming too much power during steering phases. The method and solutions presented in this paper could be generalized to many types of multiaxe vehicles and robots in order to improve their skid-steering performance. Indeed, on vehicles with three or more axles, one can imagine an adaptive suspension capable of modifying the normal force distribution on the wheels without changing either mass or payload distribution in the vehicle. This is currently done manually in the Kokoon vehicle. In a future version, the suspension adjustment could be automatically performed only during turns by using a dedicated mechanism, resulting in lower energy consumption during skidding. When driving in a straight line, the adjusting mechanism would reset the initial normal force distribution for better balancing of traction forces on all the axles together with improved pitch stability.

Further work will focus on extended experimental results and improved modeling. The first thing for experiment will be to film the vehicle motion from a top view during self-rotation with a zero turning radius in order to precisely locate point \( K \).

A new version of the vehicle is also currently being constructed with chain transmissions instead of belts. This will prevent sliding inside the transmission and will allow us to obtain more precise experimental results. Another work in progress concerns the measurement of contact forces, because the force plate buried in the ground cannot simultaneously measure the contact forces on all six wheels. Moreover, when the wheel is rolling over the force plate instead of on the normal ground, there is also a change in contact parameters. For these reasons, a more sophisticated experiment is planned in the short term, with each wheel including a six-component force sensor. Hopefully, these two improvements should allow us to refine our skid-steering models and to obtain more precise experimental results.

ACKNOWLEDGMENT

The Kokoon prototype was designed, built, tested, and continuously improved with the extensive help and constant motivation of IFMA (French Institute for Advanced Mechanics) and UBP (Blaise Pascal University) students. The authors also acknowledge the financial support of OSEO-ANVAR (French National Agency for Development of Research), the Michelin Company, and the TIMS Research Federation. The other sponsors and people involved in Kokoon development are given on the Kokoon Project Web page: http://www.kokoon.fr.st.

REFERENCES


