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Multibody simulation of a small size farming tracked vehicle

Francesco Mocera[a],*, Andrea Nicolini[a]

[a]Politecnico di Torino, Corso Duca degli Abruzzi 24, Torino - 10129, Italy

Abstract

In this paper, the Multibody (MTB) model of a small size tracked vehicle for farming applications is shown. These machines may encounter several working scenarios in their operating life when equipped with different working tools. Moreover, they are used in unstructured environments that are very difficult to predict in terms of terrain conditions and slope. Depending on these factors, the actual tractive force may vary a lot requiring often a high number of field tests to qualify the vehicle performance. The numerical model built in MSC ADAMS, wants to be a software environment where several working conditions can be exploited considering the dynamic properties of the vehicle. This work focuses on the global kinematic behavior, considering the difference between imposed motion laws and the actual one.

Keywords: Multibody; Simulations; Tracked vehicle

1. Introduction

In the design process of a complex mechanical system, numerical simulations are becoming more and more helpful thanks also to higher computational resources. In a software environment, it is possible to model real operating conditions that allows for exploiting the behavior of the system, reducing the amount of field tests required. This leads to a higher data availability since the early design stage and thus to higher optimization opportunities.

To study its kinematic and dynamic behavior, a mechanical system can be seen as a group of rigid body connected together with rotational, translational or more complex joints. Thus, the obtained multibody (MTB)

* Corresponding author. Tel.: +39 011 0906897; fax: +39 011 564 6999.
E-mail address: francesco.mocera@polito.it

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system can be solved starting from external applied load and kinematic initial conditions. Several commercial software allow for a MTB simulation: it is possible to choose between general purpose multi-body codes such as MSC ADAMS, LMS-DADS and others, or dedicated solutions optimized to study specific aspects of a more complex system. Considering vehicle dynamic performance analysis, MTB based studies are among the most discussed topics as highlighted by Blundell and Harty (2014). One of the more discussed topic in this field is the wheel-terrain interaction due to the intrinsic deformable behavior of the two elements. The maximum tractive force of the wheel depends on the real pressure distribution all over the involved contact area.

Analytical models have been proposed in the literature to describe the wheel-terrain interaction. Bekker (1956) considered a terrain with plastic behavior correlating the pressure distribution to the contact area and to the terrain deformation. The "hardening" effect and other specific characteristics of each terrain are taken into account by mean of empirical constants applied to the vertical deformation. Wong (1989), Pacejka (2002) and other researchers have proposed analytical models with different assumptions on the terrain behavior. All these models usually requires a proper characterization of the terrain in order to identify the main parameters of interest.

Tracked vehicles represent a particular subsystem of ground vehicles. They cover a wide range of applications from the military field to the construction and agricultural field. Performance are strongly related to the track-terrain interaction since they usually operate in unstructured environments. The combination of high payloads and terrain variability make the dynamic analysis of these vehicles a much more difficult task. Moreover, tracks can be classified in two main categories: rigid and flexible. This difference plays a crucial role when it comes to investigate the dynamic behavior of the track and the real shape of the contact area, thus the pressure distribution. Steel links made tracks are usually classified as rigid tracks. They are usually adopted on medium-high size vehicles in military, construction and agricultural field, characterized by low linear speeds. In this case, the track is usually modeled as a series of rigid links connected together with low friction rotational joints as reported by Gao and Wong (1994). Rubber tracks are commonly classified as flexible tracks, with a defined bending stiffness. They can be found on medium-small size farming and construction machines like the one addressed in this work. The flexibility characteristic of the track strongly affects the estimation of the actual deformation of the terrain below the track and thus the tractive force as found by Wong and Garber (1984).

Multibody codes have been widely used in the latest years to model complex tracked vehicles for off-road application as in Rubinstein and Hitron (2004). These tools allows for better modeling of the main mechanisms involved such as the tensioner, the supporting wheels with their suspension system and the sprocket-idler coupling. Moreover, careful modeling also allows for a good track representation. It is usual to consider a track made by a sequence of rigid elements, with proper mass and inertia properties, linked together with rotational joints with or without joint friction applied or with bushing elements. Contact forces must be considered between each rigid body: each mesh interacts with the sprocket-idler mechanism, with the idle wheels, and with the modeled terrain. Usually, the formula proposed by Janosi-Hanamoto (1961) is implemented in prebuilt multibody codes for tracked vehicles to consider the effects of a deformable terrain on the tractive effort. The high number of meshes and thus applied contact forces allows to increase the accuracy when modeling the pressure distribution below the track. This leads to a better representation of the tractive force but implies a very high computational effort at each integration step. Thus, a proper attention on the integration parameters set up is required for optimization between precision and computational effort.

In this paper, the multibody model of a small size, multipurpose tracked vehicle for farming application is shown. Since the weight of each tool is not negligible compared to the one of the vehicle, it is possible to reach unsafe operating conditions with unstable dynamic operations. Moreover, the variability of the terrain may vary significantly the traction performance of the vehicle and thus its functionality. Therefore, two main aspects require greater attention in modeling this machines: mass and inertia distribution of each component and the track-terrain interaction modeling. In this work, greater attention has been paid to the kinematic and dynamic part of the model considering a track-terrain interaction based on standard contact laws available in the MTB code used. The obtained model has been tested studying the global behavior in a set of representative driving operating modes. This activity is preliminary for a future Hardware-In-the-Loop simulation of a full vehicle on a mechatronic real time test bench as reported in Bosso et al (2013) and Mocera and Somà (2017).

Nomenclature	
x, y, z	Linear coordinates
$\alpha_\lambda\ \alpha_\mu\ \alpha_\nu$	Angular coordinates
$\theta\ \varphi\ \psi$	Eulero coordinates
ϕ	Kinematic constraint equation
$[M]$	Inertia Matrix
$[F]$	Generalized Force vector
$\{\lambda\}$	Lagrange Multipliers
v_t	Theoretical longitudinal vehicle speed
$\omega_{sprocket}$	Sprocket angular speed
$r_{sprocket}$	Sprocket radius
s_{track}	Track thickness
i	Slip coefficient
v	Actual longitudinal vehicle speed

2. Multibody modeling

A multibody (MTB) system is a representation of a mechanical system based on a group of rigid bodies linked together by mean of a certain number of joints and subjected to a set of external forces. Joints can be seen as kinematic constraints which limit the Degrees of Freedom (DOF) of the system. This approach is particularly useful to study the kinematic and dynamic behavior of a complex mechanical system, where the initial configuration may vary a lot during the simulation. For this reason, the methodology is often used to study vehicle dynamic problems.

In a MTB system, each rigid body is defined when the following are assigned:
- Local Reference Frame (LRF)
- Inertia properties
- Initial conditions.

For each new body, a specific set of state variables are introduced in the model

$$\boldsymbol{q} = [x\ y\ z\ \alpha_\lambda\ \alpha_\mu\ \alpha_\nu]^T \tag{1}$$

$$\dot{\boldsymbol{q}} = [\dot{x}\ \dot{y}\ \dot{z}\ \omega_\lambda\ \omega_\mu\ \omega_\nu]^T \tag{2}$$

$$\boldsymbol{\Theta} = [\theta\ \varphi\ \psi]^T \tag{3}$$

that can be written as a state vector

$$\boldsymbol{y} = [\dot{\boldsymbol{q}}^T\ \dot{\boldsymbol{\Theta}}^{\boldsymbol{T}} \boldsymbol{q}^{\boldsymbol{T}}\ \boldsymbol{\Theta}^{\mathbf{T}}]^T \tag{4}$$

Joints are considered as a set of kinematic constraints on the state variables in eq. 4. From the mathematical point of view, they can be defined as an algebraic constraint of the type

$$\phi\ (\boldsymbol{y}) =\ 0 \tag{5}$$

Considering all the kinematic constraints induced by all the joints it is possible to write

$$\boldsymbol{\phi}\ (\boldsymbol{y}) = \left[\boldsymbol{\phi_1}^T(\boldsymbol{y}), \boldsymbol{\phi_2}^T(\boldsymbol{y}), \ldots, \boldsymbol{\phi_n}^T(\boldsymbol{y})\right] =\ \left[\phi_1{}^T(\boldsymbol{y}), \phi_2{}^T(\boldsymbol{y}), \ldots, \phi_m{}^T(\boldsymbol{y})\right] \tag{6}$$

where n, the number of joints, can be lower than the number of constraints applied by each single joints. These equations can also be defined for the first and second derivative of the state variables. Moreover, they can be used by the MTB code solver as a check to ensure that all the points of the system are moving coherently at each integration step. The set of equations in eq.6 may be seen also as a function of time since the kinematic constraints may also refer to a particular motion law that the point/body has to satisfy.

Equation of Motions (EOMs) for a n-DOF MTB system can be schematically written as

$$[\boldsymbol{M}(\boldsymbol{y})]\dot{y} = [\boldsymbol{F}(\boldsymbol{y}, \dot{\boldsymbol{y}}, t)] \tag{7}$$

To find the solution to eq. 7 that satisfies kinematic constraints expressed by eq.6, the Lagrange multipliers method can be used. In the case of holonomic constraint, a set of Differential Algebraic Equations (DAE) is obtained as explained more in detail by Shabana (1989)

$$\begin{aligned} [\boldsymbol{M}(\boldsymbol{y})]\dot{y} &= [\boldsymbol{F}] + \frac{\partial \phi}{\partial \boldsymbol{y}}^{T} \{\lambda\} \\ \phi\,(\boldsymbol{y}, \boldsymbol{t}) &= 0 \end{aligned} \tag{8}$$

DAE systems are not easy to solve, especially when involving strongly non linear systems. This is the reason why software houses have invested lots of efforts in implementing the most efficient integration techniques in their codes allowing the user for better choice between simulation time and solution precision.

2.1. Modeling the tracked vehicle

The MTB model of the tracked vehicle was developed using the MSC ADAMS® software. It consists of a main body with the inertia properties of the real vehicle and two tracks. Moreover, for each track the sprocket and idler wheel, the tensioner and the idle wheel roads were modeled.

To model the track, a series of 40 links like the one shown in Fig.1 was considered. When dealing with the power transmission from the sprocket to the track, the key was the design of the proper pitch of the equivalent chain system. The simplified shape of the link maintains the functionality of the real track since the central rail guides the idle wheels in operating conditions.

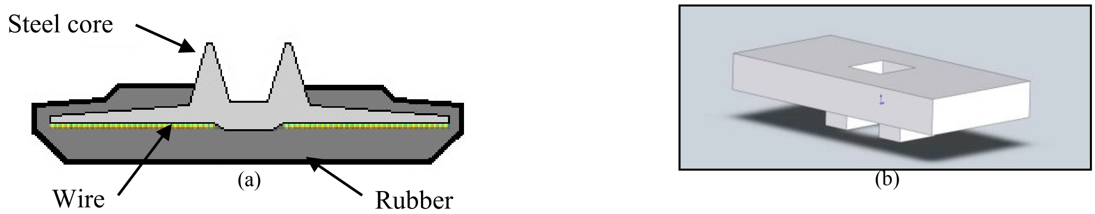


Fig. 1. (a) Track cross section, (b) Modeled link

Track of small size farming vehicles are usually rubber made with a steel reinforced soul Fig. 1(a). The single link was modeled as a simplified rectangular plate with a central hole where the sprocket teeth engage to transmit torque Fig. 1(b). Thus, a connection of each single link was necessary to replicate the properties of the real rubber track.

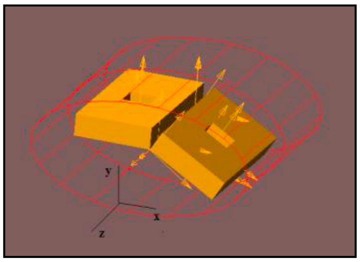


Fig. 2. Bushing element

A bushing element was used to connect each couple of links (Fig. 2). This element simulates elastic and damping forces and torques acting between two elements. Their amplitude can be defined by the user by mean of stiffness and damping coefficients due to the implicit linear relation between forces and displacements. The used values of stiffness are listed in Table1.

Table 1. Bushing parameters

	x	y	z
Translational Stiffness [N/mm]	10^4	10^4	10^4
Rotational Stiffness [Nmm/deg]	10^4	10^4	200

High values of translational stiffness and rotational stiffness in x and y direction make the angular misalignment (in x and y direction) and the displacement between two near links negligible. The value of rotational stiffness in the z direction replicates the bending stiffness of the rubber element and it was fitted comparing the deformed shape of the single track model with a real rubber track (Fig. 3). Rotational damping was also necessary to prevent undesired track vibrations during the simulation.

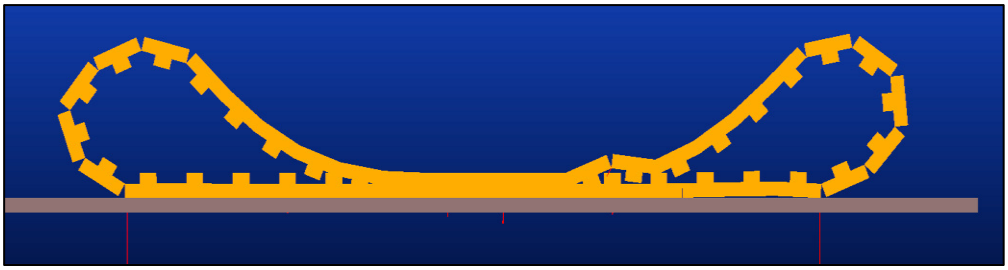
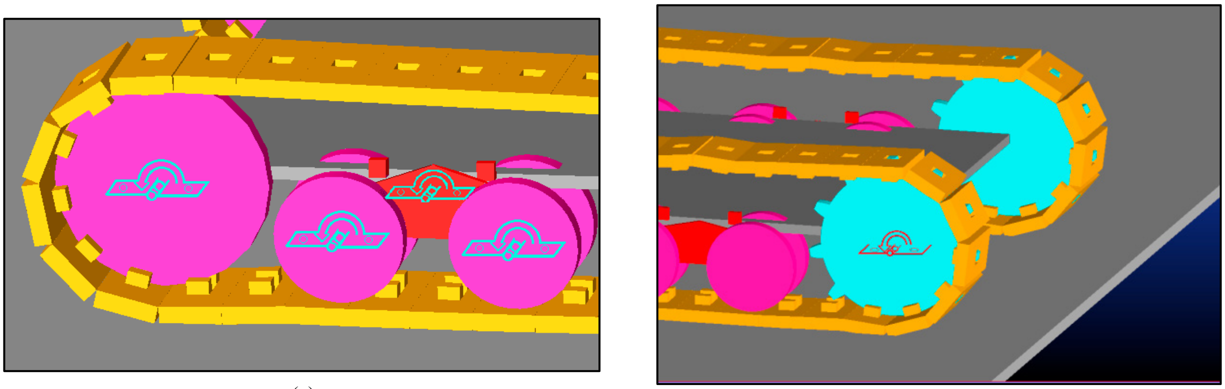


Fig. 3. Track deformed shape

Once the track properties were set, the sprocket-idler system, and the wheel road were modeled as can be seen in Fig. 4. The oscillating arm was connected to the main body through a rotational joint as also each wheel to its corresponding support. Between each track link and each wheel a contact force law was imposed. The model used considered a low penetration depth between bodies, a certain damping effect (5.0 Ns/mm), a stiffness value of 100 N/mm, and a low friction coefficient. Actually, contact forces between the sprocket teeth and each link of the track are responsible of the torque transmission.



(a) (b)

Fig. 4. (a) Idler wheels connections, (b) Sprocket gear

Finally, the tensioning system was modeled as in Fig. 5. To have relative rotation between the idle wheel and the main body, but also ensuring the right tensioning force, an intermediate pivot body was used. A translational joint connects the pivot body to the main chassis while a rotational joint allows for the rotational motion of the idler. To this pivot, a tensioning force of 5100 N was applied considering typical values used in small machines with similar applications. The system was assembled in such a way that at the beginning of the simulation, the tensioning system allow for a correct contact between the tracks and each single component.

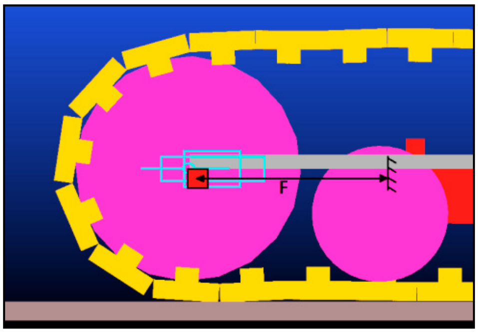


Fig. 5. Tensioning system

Finally, the two tracks were connected with the main body that replicates the chassis of the vehicle. In Fig. 6 the complete model is shown. It is important to highlight the dynamic role of the main body. Despite its simple shape, the inertia properties of this rigid body were set in order to replicate the behavior of a real farming vehicle.

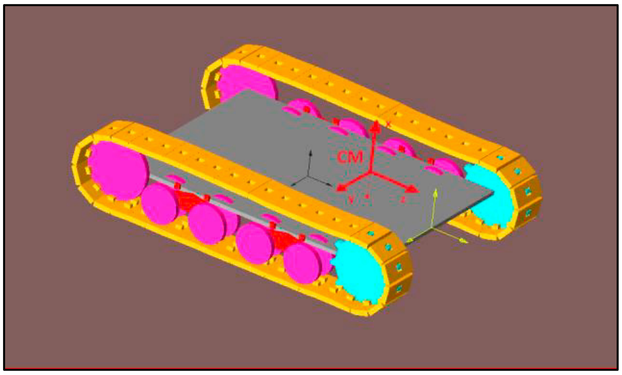


Fig. 6. Complete model of the farming vehicle

3. Simulations and results

In the following sections, the results of two simulation cases are shown. To analyze the longitudinal model behavior a straight forward was considered comparing the kinematic behavior of the vehicle on a horizontal and inclined plane. Then a counter-rotation of the tracks was analyzed to also investigate the lateral behavior. One of the goals of this work was to fix the main parameters of interest that can be investigated once a more complex contact model is implemented.

3.1. Forward motion

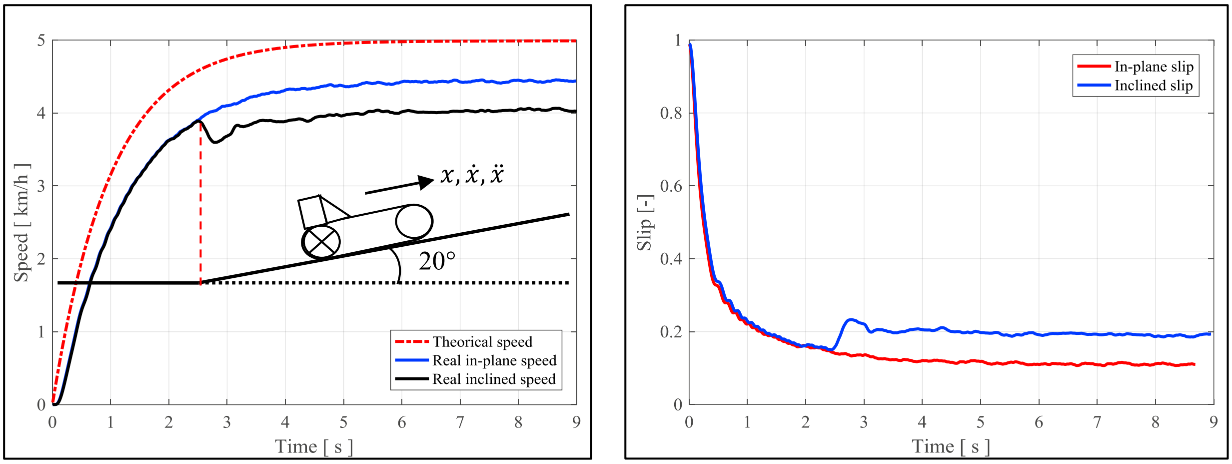
In this first test, the vehicle is simulated on a straight trajectory in order to evaluate the slip effects of the modeled contact forces between tracks and terrain. A predefined angular speed law was assigned as imposed motion on the sprockets. Thus, the expected theoretical linear speed of the vehicle, considering an ideal condition without slip, can be defined as

$$v_t = \omega_{sprocket}\left(r_{sprocket} + s_{track}\right) \quad (9)$$

and a slip factor can be defined as

$$i = 1 - \frac{v}{v_t} \quad (10)$$

Two different scenarios were simulated: a forward motion on a horizontal terrain and a mixed case where the vehicle approached a slope of 20 degrees after a horizontal start. Since the sprockets speed law was given, the linear speed of the vehicle strongly depended on the effective traction force available. This was the reason why the real linear speed differs from the theoretical one. Moreover, due to the loss of normal contact force on the inclined plane, the steady state speed reached will be different also between the two cases.



(a) (b)

Fig. 7. (a) Longitudinal speed results, (b) Slip coefficient during the simulation

In Fig. 7 it is possible to see the difference between the real speeds and the theoretical one especially when the vehicle approaches the inclined plane. Due to the loss of traction force, the slip increase from a value of 10% to 20% when the terrain changes its slope.

3.2. Counter rotation

Tracked vehicles are one of the few vehicle categories able to do a 360° turn in a very small space. These vehicles turn using the difference in speed and rotational direction between each track. Controlling the flow rate in each hydraulic motor it is possible to control the speed and the direction of the sprocket angular speed. In this case, an equal and opposite angular speed was imposed to each sprocket to simulate a 360° turn in order to evaluate how the slip affects the necessary area to realize the rotation.

In Fig. 8 the trajectories of several key points of the vehicle are shown. The center of mass (CM) trajectory is compared with other two main geometric points: the middle point of the segment connecting the sprockets' centers and the geometric center of the main body of the vehicle. As shown, the CM trajectory is much smaller than the other two. It lies within a 100x100 mm area denoting its role in determining the dynamic of the vehicle. It acts as a pivot point since the higher mass concentration around it maximizes the available tractive force.