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DESIGN AND SIMULATION OF A MOBILE PLATFORM WITH A SEMI-ACTIVE SUSPENSION FOR UNEVEN TERRAIN[1]

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The paper is focused on the design of a mobile wheeled platform able to move in uneven/unstructured terrains and particularly intended for supporting the work in agriculture. An independent double wishbone suspension is chosen to obtain a light and compact structure. Furthermore, a semi-active suspension with magnetorheological dampers and the ability to change the track of wheels is proposed to minimize uncontrolled vertical movements of the platform. A dynamic model is formulated to carry out simulations including various obstacles and cases with constant/controlled damping coefficients. As a final result, a conceptual CAD model is built with selected motors and standardized parts.

Keywords: robotics, agriculture, wheeled robot, wishbone suspension, magnetorheological damper

1. Introduction

The topic of vehicles and robots motion in uneven/harsh terrain has been studied by many researchers resulting in designing devices with various locomotion systems (Bruzzone and Quaglia, 2012). These are not only mobile platforms/robots equipped with tracks, legs (Garimella and Revzen, 2021; Raibert *et al.*, 2008) or wheels (Husti, 2019; Shamshiri *et al.*, 2018), but also more complex solutions of wheeled robots with high mobility (Shah *et al.*, 2012), including hybrid like wheel-legged devices (Niu *et al.*, 2018; Olinski and Ziemba, 2014; Sperzyński *et al.*, 2018). Mobile platforms need to move in unstructured environments (off-road terrains), hazardous surroundings, catastrophe sights (buildings, rubble), due to their various applications in transport, rescue (Niu *et al.*, 2018), exploration including for instance cultural heritage (Ceccarelli *et al.*, 2017) or space (Harrington and Voorhees, 2004), military, as well as in agriculture (Ackerman, 2015; Roldán *et al.*, 2018). Nowadays, the automation of agriculture by using mobile platforms and robots becomes crucial, since it not only helps enhancing productivity, but also improving safety by assisting human labor with heavy machinery, pesticides, etc. Therefore, the aim of this paper is to design a platform characterised by high mobility and maneuverability in uneven (off-road) terrains and particularly suitable for application/supporting work in agriculture.

The first step is to decide the type of the locomotion system. Walking systems are usually very energy consuming, complicated in construction/control, and their advantages including the ability to move in uneven terrains, to adapt the walking mode to the current terrain or regulate unit pressures on the ground do not seem to be significant for a designed light off-road mobile platform. Considered was also the possibility of using a tracked drive studied for instance for its advantageous low soil compaction in (Raper, 2004), or in (Chołodowski, 2023), where the movement resistance in tracked off-road vehicles was modeled. However, taking into account the fact of better maneuverability of wheeled vehicles and their lower costs, as well as lesser internal

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resistances which translate into the possibility of selecting smaller motors, it was decided to use a wheeled chassis. It can also move on any terrain, and wheels with tires in combination with an appropriate suspension and control systems, can provide good stability and maneuverability. Thus, the focus in this paper is placed on wheel vehicles and suspension systems.

In the process of designing a wheeled vehicle that can move efficiently in unstructured terrains (off-road), the key stage is the appropriate suspension design. It provides the possibilities of, among others, continuous generation of the traction force by minimizing the time of slip and separation of wheels from the ground. As a consequence, the device is able to overcome obstacles, reduce vibrations that lead to wear of its equipment (sensors, cameras, etc.), as well as to maintain correct height of the body, minimizing its uncontrolled movements. The suspension consists of a set of movable, rigid and elastic parts connecting the wheels with the chassis. The suspension systems can be generally divided into dependent (vertical movement of one wheel has an impact on other wheels) and independent (each wheel moves on its own); as well as into passive, active and semi-active (due to the level of controlling suspension parameters).

The most commonly applied suspension systems are passive, where the structure consists of deformable and damping elements with constant/unchanging characteristics. An active suspension is a system that uses motors and electronics to control suspension damping and stiffness in real time. Alternatively, a semi-active suspension cannot change the stiffness of its elements, but is able to modify the damping force in real time. This solution is cheaper, simpler to build, less prone to failures and consumes much less energy than fully active suspension systems (Fischer and Isermann, 2003).

Furthermore, within years, many configurations of suspension systems were developed. One of them is a multi-rocker suspension system shown in Fig. 1a and applied in Scout 2.0 robot. Others include, for instance, a swing-arm (trailing arm) as in Scout Mini from AgileX company or a rocker-bogie suspension system widely used in space vehicles like Mars rovers (Harrington and Voorhees, 2004). However, many existing wheeled mobile robots have limited capabilities of maintaining a stable position/orientation of the platform, since instead of using a suspension system they depend only on deformability of wheels/tires. Examples are BoniRob (Ackerman, 2015) or ecoRobotix (Fig. 1b), a prototype of platform monitoring agricultural fields, recognizing and spraying weeds (Ecorobotix, 2019).

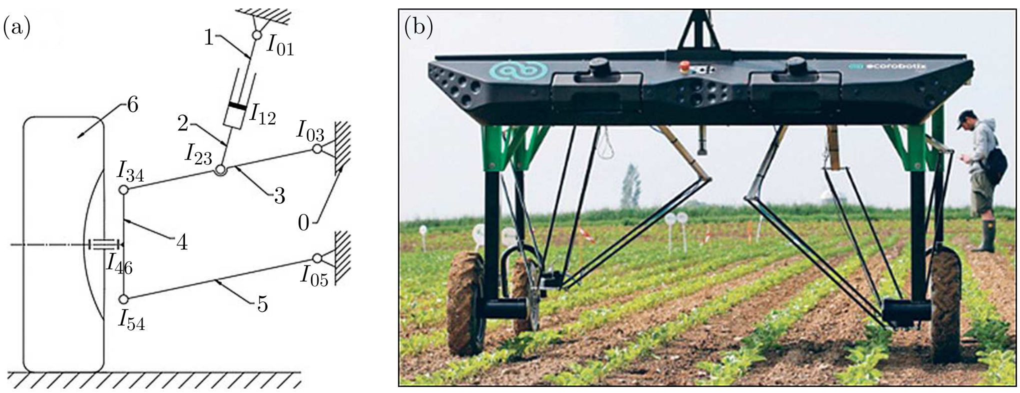

Fig. 1. Views of existing: (a) independent multi-rocker suspension with real mobility equal to 1, (b) ecoRobotix wheeled mobile platform for agricultural applications (Ecorobotix, 2019)

For the above presented reasons, it is concluded that there is still a need for lightweight mobile platforms that could move in uneven terrains and perform tasks to support agriculture. A particular aim is designing a suspension system that would allow one to minimize the uncontrolled vertical movement and vibrations, so the platform can be easily applied in various conditions to a wide range of tasks. This way it will also be ready for installation of different types of additional equipment and sensors, especially manipulators, cameras, lidars, etc. Thanks to these solutions, a complete device could achieve independence and perform planned tasks autonomously. Therefore, in the paper, the kinematic form of the chassis with the suspension and selected dimensions will be determined. This will allow building a numerical model (including the estimated masses of elements) and conduct experiments. One of the aims is also to simulate a semi-active suspension modeled with magnetorheological (MR) dampers. Basing on the outcomes of trials, elements such as drives will be selected, and the final result of work will be a 3D CAD model of the mobile platform (Cholewa, 2023).

2. Conceptual design of a mobile platform

2.1. Assumptions and designed kinematics

Developing a solution for the design of a mobile platform requires first to formulate working conditions and assumed functionalities including flexibility of application and movement in unstructured terrains (focusing on cultivated farmlands), minimizing the uncontrolled vertical movements and vibrations, supporting work in agriculture like harvesting/monitoring of crops and gathering samples.

After checking the state of the art and considerations of applications, the consecutive structural assumptions for the mobile platform and its suspension have been formulated:

- Mobile platform as a 4-wheeled vehicle,
- Independent double wishbone suspension system with a motor for each wheel,
- Semi-active suspension based on a shock absorber as a spring with an MR damper,
- Changing the dimension of track of wheels by at least 30%,
- Compact design, platform size fitting in 1 m^3 to simplify its transport,
- Lightweight design – less than 65 kg, platform mass with assumed additional equipment.

Taking into account all the requirements and assumptions, the design of the suspension system as a kinematic scheme (Fig. 2) has been created. An independent double wishbone suspension (type of earlier presented multi-rocker suspension) is chosen to obtain a light and compact structure allowing one to apply wide tires and to lower the platform center of mass to increase stability. The turning of the vehicle is going to be realized by varying the rotational speed of motors installed in each wheel.

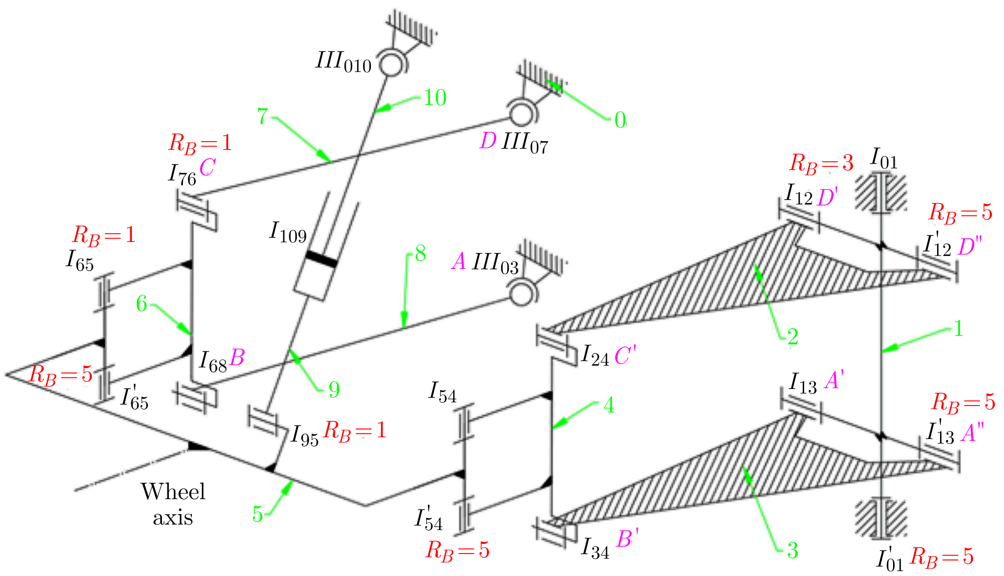

Fig. 2. Kinematic scheme of the proposed suspension mechanism (view for one wheel)

The theoretical mobility has been calculated according to the well-known structural equation as

$$W_T = 6(n-1) - 5p_1 - 4p_2 - 3p_3 = 6 \cdot 10 - 5 \cdot 16 - 3 \cdot 3 = -29 \tag{2.1}$$

and the real mobility as

$$W_R = W_T - R_B = -29 + (5 \cdot 5) + 3 + 1 + 1 + 1 = 2 \tag{2.2}$$

where: n – number of mechanism elements, p_i – number of kinematic pairs of particular i class, R_B – number of passive (excessive) constraints. The passive constraints are in this case connected with multiple parallel first class kinematic pairs (5 of these) and building plane 4-bar mechanisms in space.

The shock absorber consists of a spring and damper that are modeled together as elements 9, 10 in Fig. 2. Moreover, it is assumed that the platform is going to be equipped with a manipulator, camera and set of sensors, so in an attempt to minimize uncontrolled vibrations, vertical movements of the main body and consequently errors in its positioning, a semi-active suspension with magnetorheological dampers is proposed. Their application advantageously provides fast responsiveness, relatively low weight, as well as simple construction, since viscosity of the damper fluid and, consequently, the damping coefficient and force can be controlled in real time (Sarami, 2009). To realize this, an MR damper LORD RD-1005-3 has been selected in accordance with the planned dimensions and predicted movements. Its stroke is 35 mm and the damping coefficient ranges from 1.3 Ns/mm to 2.9 Ns/mm (Mohd Yamin *et al.*, 2022).

In addition, due to various types of terrains, the desire to avoid obstacles and to adjust the device to driving between irregular rows of plants, it was decided to include in the vehicle the ability to change its track of wheels. Therefore, the proposed suspension system is a mechanism with the mobility of two. The first one is the vertical wheel movement which enables overcoming obstacles and minimizing vertical chassis movements, whereas the second mobility is an additional rotation of the suspension around the vertical axis (element 1 in Fig. 2), resulting in altering the track of wheels with a simultaneous change of the wheelbase. This mechanism is intended to increase the flexibility of vehicle applications.

Overall, the proposed double-wishbone suspension is characterized by connecting the chassis and the wheel with two wishbones, thus the entire system takes the form of a 4-bar mechanism (Fig. 2). The wishbones are elements No. 2, 3 and the same function is fulfilled by elements No. 7, 8. The duplication of parts is required for proper operation of the system for changing the track of wheels.

2.2. Determined dimensions of the device

Considering the developed kinematics, including the suspension and system for changing the track of wheels, as well as the defined assumptions/requirements, the dimensions are determined for the platform (Fig. 3a) and suspension (Fig. 3b). A couple of additional assumptions formulated particularly for the suspension dimensions include wheel maximum vertical lifting 55 mm and maximum lowering 45 mm, wheel tilt its max/min wheel vertical position: $-4°/5°$. Rotational pairs A and D (Fig. 2 – mounting points of the lower and upper wishbones) of the 4-bar mechanism ABCD have common vertical axis of rotation to simplify the design for changing the track of wheels.

The determined dimensions are proposed also in view of the desired change of track of wheels by at least 30%. The principle of work for this mechanism is presented in Fig. 4. It proves that a 70° rotation of the suspension results in changing the track of wheels by more than 230 mm, which for the maximum width of 690 mm is a change by over 33%. During the process, the wheel all the time stays parallel to the chassis which enables a continuous change during the ride and

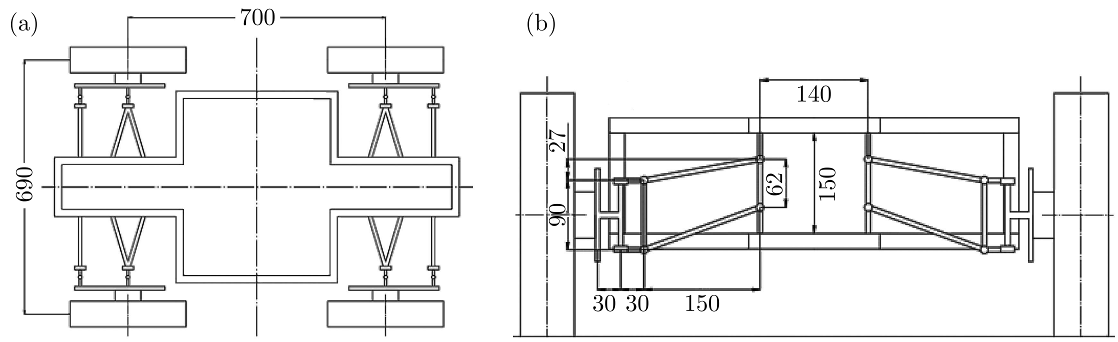

Fig. 3. Designed dimensions (in mm) indicated in: (a) an overall scheme of the mobile platform, (b) a scheme with details of the suspension system

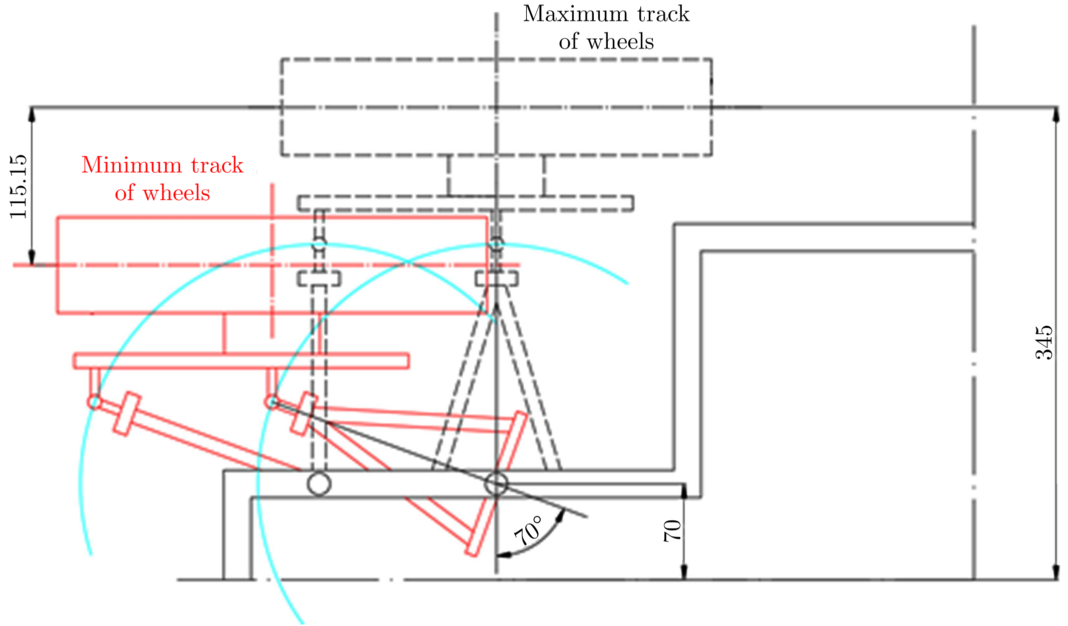

Fig. 4. Scheme presenting the working principle of the mechanism changing the track of wheels

setting any dimension in the obtainable range. It is also crucial that this process does not change length of the shock absorber and therefore does not have a negative impact on its operation. In order to ensure this, the shock absorber is mounted in accordance with the wishbone. Therefore during the process, one end of the shock absorber (Fig. 2 – pair III_{010}) is exactly in the axis of rotation (Fig. 2 vertical axis of rotation for pairs A, D), while the other end (Fig. 2 – pair I_{95}) moves in a circle (along the right arch in Fig. 4).

3. Numerical studies

3.1. Built numerical model

In order to perform test trials of the designed mobile platform, a spatial multibody dynamic numerical model with a semi-active suspension and system for changing the track of wheels has been built in Adams simulation software (Fig. 5). The model is based on elements with simple shapes, but all crucial features are reproduced according to the developed device kinematics/dimensions, as well as to the approximated mass of elements and mass distribution (possibly close to reality). Basing on diagrams and sketches, an initial 3D model of the vehicle was created, assuming the frame to be made of aluminum profiles and sheets. The mass of the preliminary 3D model was 28 kg. The weight of additional equipment in the form of drives, batteries, sensors, cameras, electronics, transported load and/or manipulator was assumed to be another 25 kg. The final mass of the modeled vehicle (64 kg) includes also a 1.2 safety factor. All the additional mass is placed in the centre of the vehicle (as yellow cylinder in Fig. 5).

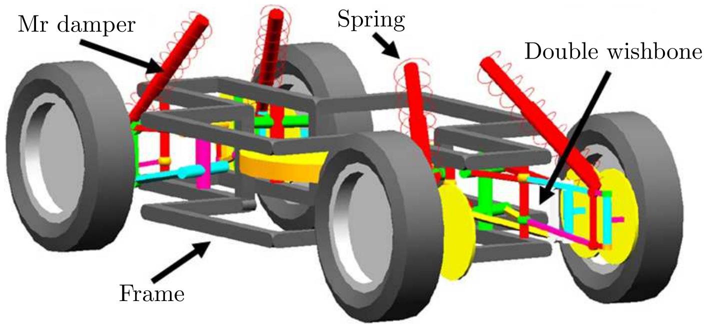

Fig. 5. Multibody dynamic numerical model of the platform built in Adams software

The model in Fig. 5 includes also visual presentation of the spring and damper, but the simulated complex force of shock absorber F_{SA}, consisting of spring F_S and MR damper F_D, is calculated in the model according to the below equation

$$F_{SA} = F_S + F_D = (F_P + k\Delta l_S) + bV_D \tag{3.1}$$

where: F_P – spring preload force, k – spring stiffness coefficient, Δl_S – spring length change, b – damper damping coefficient, V_D – velocity of damper length change.

Since MR dampers are applied, in some simulations the damping coefficient is changed and controlled. Therefore, to obtain an adaptable damping force, a control system minimizing the vertical velocity of chassis, is also implemented in the model. Specifically, for each damper, the vertical velocity of the frame corner is measured and minimised. In addition, b is limited to values between minimum and maximum (1.35 Ns/mm to 2.9 Ns/mm) achieved by the assumed MR damper. At this point of research, a proportional regulator (P regulator) is applied to calculate the damping coefficient b as

$$b = K_P(0 - V_y) \tag{3.2}$$

where: K_P – proportional gain of the regulator, V_y – vertical velocity of the chosen point (corner) of chassis.

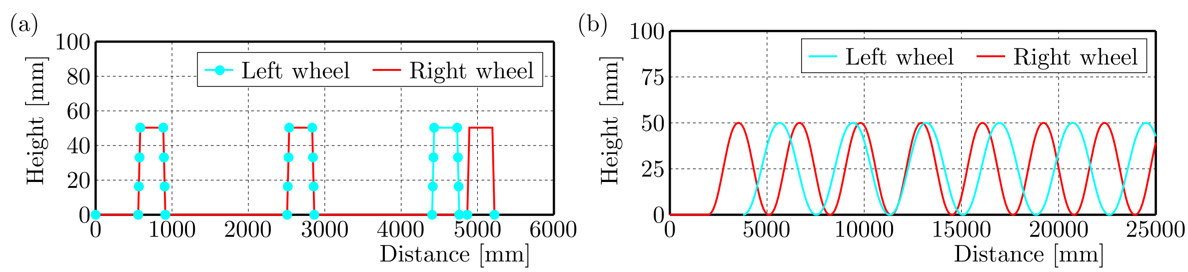

Fig. 6. Profile of road: (a) with step obstacles, (b) with sinusoidal obstacles

In order to perform the planned simulations, the numerical model is supplemented with definitions (in additional files for usage with Adams) of 2 road profiles: a) road with step obstacles (Fig. 6a) – 3 obstacles with height 50 mm, length 300 mm, third obstacle is shifted for right and left wheel; b) road with sinusoidal obstacles (Fig. 6b) – height 50 mm, length about 1600 mm, in this case the obstacles for right and left wheels are shifted unevenly.

For all types of roads the tire features and tire/road contact conditions are assumed. Basing on literature (Blundell and Harty, 2004) a tire Fial model is applied, with parameters specified in another file for Adams – unloaded tire radius ($R_1 = 157.5$ mm), width ($d = 70$ mm), vertical stiffness ($k_z = 24$ N/mm), vertical damping ($\zeta = 0.6$ Ns/mm), rolling resistance ($C_r = 27$ mm), longitudinal force with respect to the longitudinal slip ratio ($C_S = 300$ N), lateral force connected with the slip angle ($C_a = 22$ N/rad), coefficients of friction at zero slip ($U_{min} = 0.65$) and when the tire is sliding ($U_{max} = 0.95$).

3.2. Performed simulations

For the built numerical model, with the defined road profiles and tire/road contact conditions, a couple of simulations was performed. The experimental modes with a particular set and measured parameters are specified in Table 1, where: k, b are the set values of spring and damper coefficients, K_P – proportional gain of the regulator, F_P – spring preload force, a_V – vehicle acceleration, V_{max} – maximum device velocity kept after 0.5 s of acceleration, d_b, V_b, a_b – measured body displacement, velocity and acceleration, F_j, F_D – measured forces in joints and dampers, T_M – torques of motors.

Table 1. Planned experiments with specified model/simulation conditions and collected variables

<table>
<tr><th>No.</th><th>Experimental mode</th><th colspan="2">Set values of variables</th><th>Collected variables</th></tr>
<tr><td>1</td><td>Road with step obstacles
Constant damping coefficient</td><td>$b = 2.1$ Ns/mm</td><td rowspan="3">$k = 1750$ N/mm
$F_P = 156$ N
Vehicle's $V_{max} = 1.5$ m/s
Vehicle's $a_V = 3$ m/s^2</td><td rowspan="3">d_b, V_b, α_b
F_j, F_D, T_M</td></tr>
<tr><td>2</td><td>Road with step obstacles
Variable damping coefficient</td><td rowspan="2">$b-$controlled
$K_P = 10^5$</td></tr>
<tr><td>3</td><td>Road with sinusoidal obstacles
Variable damping coefficient</td></tr>
</table>

The simulations were conducted with a 100 Hz frequency, partially in order to obtain a realistic operation of MR dampers – time of damping coefficient change is therefore limited to 0.01 s. The spring stiffness ($k = 1750$ N/mm) and its preload force ($F_P = 156$ N) were selected on the basis of intermediate simulations. These values enabled the vehicle to maintain a stable vertical position of the chassis (without fluctuations) in the absence of external loads, and enabled the spring to deform freely when overcoming the obstacles. An average damping coefficient $b = 2.1$ Ns/mm was selected for the MR damper in cases when it was kept as constant.

Simulations concerning the changing of the track of wheels during ride and overcoming obstacles with various values of track of wheels were also performed in Adams. Details are not reported in this paper, but the obtained results (e.g. max. value of torque for the mechanism changing the track of wheels was 33 Nm) proved the feasibility of the system as it does not collide with the chassis and allows selecting linear motors shown in Section 5.

4. Results of simulations

According to the details presented in Table 1, the 3 planned simulations were carried out in Adams by using the built numerical model. The results include, among others, the acceleration of the body center of mass. The simulations allowed also for the assessment of the correct operation of the built P regulator, which controls the magnetorheological damper. In addition, taking as criteria the minimization of body vibrations and the maximization of safety, understood as the vehicle ability to maintain its wheels in contact with terrain, the ground reaction forces and time of possible losses of contact between the wheels and ground were verified. A part of the obtained simulation results has been presented in Figs. 7-10.

For the first simulation of the vehicle movement on the road with step obstacles and with a constant damping coefficient, the platform centre of mass accelerations in 3 axes are measured and presented in Fig. 7. Overall, the mean value of vertical acceleration as a_{RMS} (root mean square) is equal to $1.81\,\mathrm{m/s^2}$ and the maximum value does not exceed $9.8\,\mathrm{m/s^2}$ (Fig. 7). As can be seen in the plot, the road profile (Fig. 5a) is mapped in the results of vertical acceleration. At the road beginning there is the first obstacle, symmetric for right and left wheels. The vehicle is overcoming it during accelerating to max. velocity $V_{max} = 1.5\,\mathrm{m/s}$. This constant speed is kept, and after 2.5 m there is another symmetric obstacle. At the end, there are 2 asymmetric obstacles.

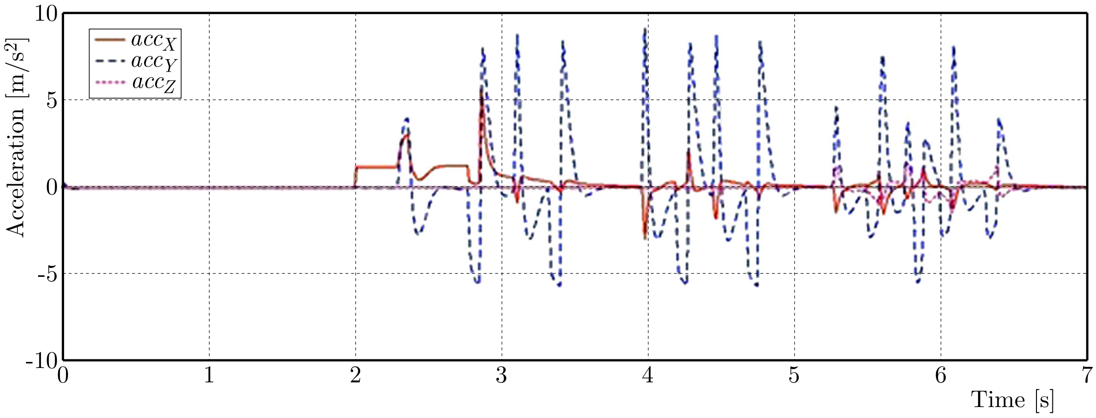

Fig. 7. Accelerations of the vehicle chassis centre of mass during the first simulation of riding on a road with step obstacles with a constant value of the damping coefficient

During the second simulation, the correct operation of the damper regulator was verified by measuring the damping coefficient value while overcoming the obstacles 2.2 s-6.7 s (Fig. 8). It can be noticed that the controller is working and the value of the damping coefficient is adjusted in real time to minimize the set error, i.e. to minimise the body vertical speed. However, the coefficient takes only the limit values in the assumed allowable range (1.35 Ns/mm to 2.9 Ns/mm), indicating the need for a more advanced control system.

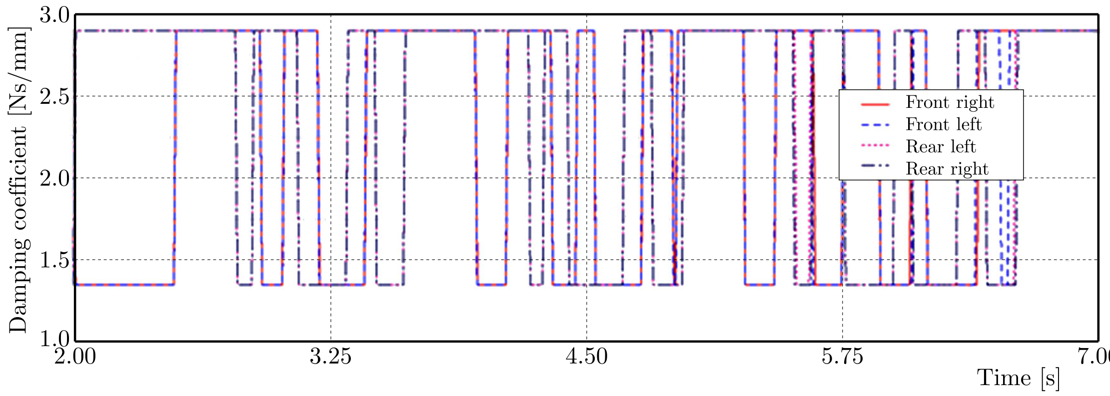

Fig. 8. Result of simulation 2 with values of controlled individually for each wheel damping coefficient while driving over the 3 step obstacles

Furthermore, in this case, the chassis accelerations were also checked and compared with the results of previous simulation. In addition, interesting values of accelerations obtained during the 3 experiments have been gathered and presented in Table 2, where a_{max} is the measured maximum value of vertical acceleration, a_{RMS} – calculated root mean square of the vertical acceleration, a_{mag} – magnitude value of accelerations calculated as a square root of the sum of squares of a_{RMS} for the acceleration in each axis.

As can be seen in Table 2, a small improvement in each of the acceleration parameters is achieved. For instance, the maximum values decreased from $9.8\,\mathrm{m/s^2}$ to $9.7\,\mathrm{m/s^2}$ for the case with the semi-active suspension and, similarly, the a_{RMS} dropped from $1.81\,\mathrm{m/s^2}$ to $1.77\,\mathrm{m/s^2}$. It can be concluded that the semi-active suspension with MR dampers proved to be useful, and even better results are expected when control of the damping coefficient is improved. However, it should be noted that these cases with very steep obstacles enabled above all evaluation of the operation of the semi-active suspension system and regulator in exceptionally unfavorable terrain conditions, which resulted in relatively high values of accelerations.

Table 2. Numerical results of chassis accelerations for each simulation

No.	Type of simulation	Acceleration of chassis in [m/s^2]		
		Vertical (in y axis)		Magnitude
		a_{max}	a_{RMS}	a_{mag}
1	Step obstacles – Constant damping coefficient	9.8	1.81	1.85
2	Step obstacles – Variable damping coefficient	9.7	1.77	1.81
3	Sinusoidal obstacles – Variable damping coefficient	0.39	0.06	0.08

Furthermore, the vehicle must be able to move in unstructured terrain, so it is important that it maintains contact with the ground even when overcoming rough obstacles. For this reason, the vertical ground reaction forces are verified for each wheel and presented in Fig. 9 (No. 2). These allowed one to check when the wheels are separated from the ground, since it corresponds to a zero force value and entails temporary inability of the wheel to generate the traction force. Duration of losses of contact is checked for each wheel: front left (0.18 s), front right (0.19 s), rear left (0.19 s), rear right (0.19 s). The sum of time when at least one wheel is not in contact with the road amounts to about 0.5s. This indicates decent vehicle behaviour during overcoming step obstacles, also in view of the fact that the design assumes a motor in each wheel. Therefore, even if one of the wheels loses traction, then the other three are still in contact with the ground (e.g. visible in Fig. 9 while overcoming the asymmetric obstacle at about 6 s).

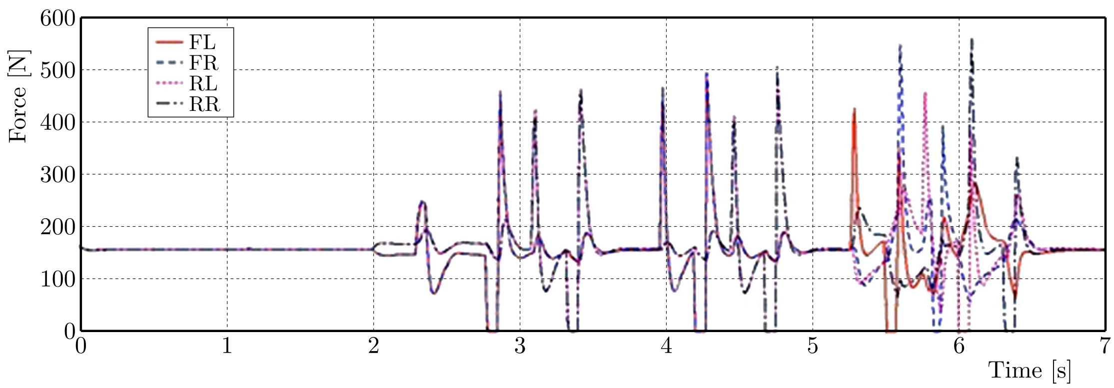

Fig. 9. Vertical ground reaction force on each wheel (FL, FR – front left and right, RL, RR – rear left and right) for simulation 2 while driving over the 3 step obstacles with the controlled damping coefficient

Additionally, thanks to the advantageous work of the suspension system and its components, for most of the time, the left and right wheels are equally loaded (Fig. 9), which enables achieving favorable contact conditions for traction force generation. Moreover, when stationary, the vertical force is about 157 N (Fig. 9) on each wheel which sums to the effect of the whole vehicle mass (64 kg).

Simulated is also movement on the road with relatively high, but not steep sinusoidal obstacles (Table 1 – No. 3). This way, a realistic terrain and work conditions for the device have been reproduced. The simulations results are among others the motor torques for each wheel (Fig. 10) and forces in joints. For the case of constant vehicle velocity, the average wheel torque is about 3.75 Nm (Fig. 10) and the maximum values do not exceed 7 Nm (omitting the results in initial seconds probably distorted by excessive slipping). In the case of this less demanding road with sinusoidal obstacles, which do not appear suddenly, but their height increases gradually, the vertical accelerations are minor and much smaller than those in previous simulations (Table 2). The maximum value does not exceed $0.4\,\mathrm{m/s^2}$ and the RMS is $0.06\,\mathrm{m/s^2}$. What is more, in this case, it was possible to continuously keep all wheels in contact with the ground, which proves the feasibility of the designed semi-active suspension system.

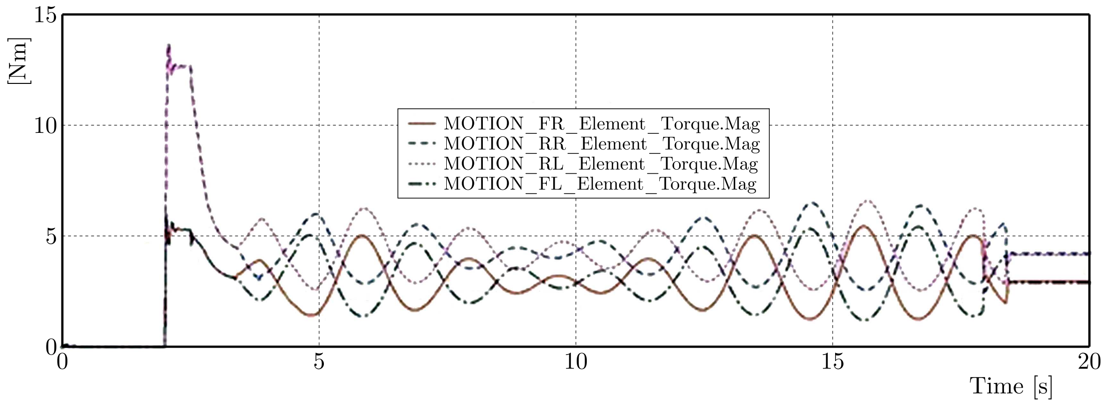

Fig. 10. Torques in wheels for simulation No. 3, moving on a road with sinusoidal obstacles while the damping coefficient is controlled and changed

All the results proved that the semi-active suspension system works properly and that the designed platform has the ability to move in uneven terrains, as well as is characterised by decent contact with the ground, overall good vertical stability, acceptable acceleration amplitude, frequency and vibration damping.

5. Built CAD model

A 3D CAD model of the designed mobile platform has been created in Autodesk Inventor software with particular emphasis put on presenting one wheel suspension (Fig. 11a) and the whole device with selected elements (Fig. 11b). It serves as an illustrative presentation of the vehicle concept and enables one to verify that no collisions occur between the main components.

The model was made taking into account the selected, on basis of simulation results, parts like: wheel drive as BDLC motor Dunkermotoren BG45X15SI with gearhead PLG40LB; clutch ROTEX KTR 98ShA,14 and electric linear cylinder FESTO EPCS BS-32-75-3P for changing the track of wheels, etc. Individual elements were also checked for their strength/stress features to select proper standardized parts that can be used in the final design and become a basis for building a prototype.

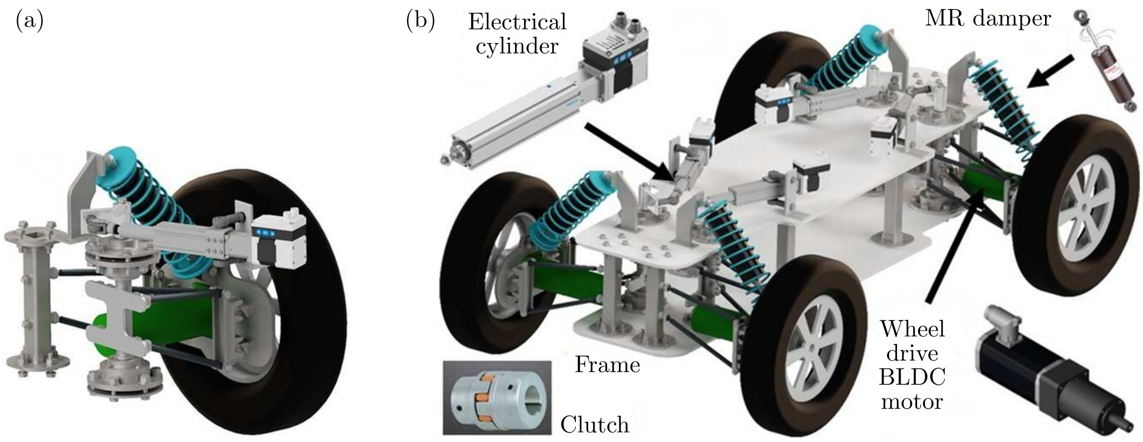

Fig. 11. A spatial 3D model built in CAD software (Inventor): (a) details of the suspension system for one wheel, (b) view of the whole vehicle with a part of selected elements

6. Conclusions

The paper is focused on the development and design of a mobile wheeled platform able to move in an uneven terrain and particularly intended for a supporting work in agriculture. The research is conducted partially, in view of the increasing demand for food production and the desire to achieve smart and precise farming with the usage of autonomous machines.

The exact planned application of the platform is combining it with a manipulator (like Kinova Gen3 Lite, or similar in workspace, payload and weight), as well as equipping it with a camera and various sensors, e.g. lidar, distance sensors, inertial IMU sensors. This way an autonomous multi-purpose mobile device could be obtained. An independent stabilization system for the camera or other elements would not be desirable, since it would limit the vehicle suitable cooperating equipment to special models. Therefore, the concept of minimizing the uncontrolled platform movements and vibrations is studied. Strictly agricultural applications could possibly focus only on providing the best ground contact conditions – the least loses of contact, omitting the overall vibrations problems, since no comfort for the operator would be necessary. This could be obtained by maximising spring stiffness and damping coefficients. However, the intended operation of the platform is wider, concerning various terrains and applications.

For these reasons, it was decided to design a 4-wheeled platform and emphasis was placed on the concept (application of MR dampers) and kinematic structure of the suspension system to provide flexibility for the vehicle mobility, as well as operation in various ground conditions and farmlands in agricultural applications. Constructional and functional assumptions were specified determining many features of the designed device. A semi-active independent double wishbone suspension mechanism with mobility of 2 was proposed, enabling not only the vertical movement of the wheel, but also a supplementary rotation of the suspension system around the vertical axis, realizing the change of track of wheels. These will allow the vehicle to avoid too large obstacles, increase its stability and elasticity of application.

Then, the vehicles dimensions were specified and in the next step, a preliminary design of the device was elaborated as a 3D dynamic numerical model of the platform (64 kg) with a 4-wheel suspension system formulated in a multibody simulation program (Adams). In order to carry out the experiments on the semi-active suspension with an adaptable damping force, a control system has also been implemented in the model. Semi-active suspensions and MR dampers have been modelled and simulated in other research like (Klockiewicz and Ślaski, 2023), where friction, hysteresis and actuation delay have been taken into consideration. However, this paper is focused on the design of the wheeled platform for uneven terrain in which the semi-active suspension system is a part of the developed idea. Therefore, at this stage of research, the MR dampers are modelled as simple elements and the control strategy for changing, in real time, the damping coefficient is based on a P controller, minimizing the body vertical velocity.

While simulating the driving on uneven terrain, the obtained vehicle parameters were verified and compared. The occurrences and durations of possible losses of contact between the individual wheels and the ground surface were also checked. The three presented experimental modes included different settings of road with obstacles, as well as cases with constant damping characteristics and with active damping regulation. Particularly, in order to evaluate the suspension system capability, plots and numerical results were derived from simulations. The vehicle kinematics and dynamics were examined, and the results were used to perform strength calculations of elements, as well as to select standardized parts and motors.

The first simulation evaluated the vehicle behaviour without the semi-active suspension in exceptionally unfavourable terrain conditions, with step obstacles. A satisfactory level of uncontrolled vertical movements was obtained. Furthermore, for comparison, the same terrain conditions were simulated with application of the controlled damping coefficient. Due to straightforward control (P regulator) the damping coefficient assumed only max/min values. Despite this fact, a slight but significant improvement in the uncontrolled vertical movements level was observed (maximum acceleration dropped from $9.8\,\mathrm{m/s^2}$ to $9.7\,\mathrm{m/s^2}$ and its RMS from $1.81\,\mathrm{m/s^2}$ to $1.77\,\mathrm{m/s^2}$). More realistic terrain with sinusoidal obstacles was also simulated. The vertical acceleration was lower than $0.4\,\mathrm{m/s^2}$ and the average wheels torque was about $3.75\,\mathrm{Nm}$. Vertical ground reaction forces were also checked and, consequently, the time when the wheels were separated from the ground. For the vehicle, favourable contact conditions for traction force generation in unstructured terrain were obtained. A decent level was observed when overcoming rough obstacles and continuous contact with the ground was kept for all wheels for sinusoidal obstacles.

To conclude, the results proved feasibility of the device and that the designed semi-active suspension enabled to decrease the main body undesired vertical movement and acceleration, as well as to possibly adjust the vehicle to various and individual cases of applications in uneven/unstructured terrains and in agriculture. The features enabling this, are the proposed two main advantages: the possibility of changing the track of wheels (functionality distinguishing the vehicle from many currently available), as well as application of MR dampers to obtain a semi-active suspension.

As the final result, a conceptual design of the vehicle has been proposed by selecting motors, constructional standardized parts, as well as some elements of electric and electronic systems. The device 3D CAD model has also been built, providing a good basis for constructing a research prototype in the future. Completing it would require suitable sensors to be selected, which combined with appropriate control would allow the device to operate autonomously. Focus should be placed on application of technologies and elements such as lidar to create maps of the environment and to detect objects, cameras for image recognition and analysis, but also ultrasonic and inertial IMU sensors, which would allow for proper control and minimizing body movements when overcoming unevenness. Moreover, the CAD model could still be used to improve the numerical model and perform simulations checking various tire/terrain contact conditions, as well as a more advanced MR damper model and control strategies (e.g. PID controller).

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