DESIGN AND ANALYSIS OF AN EFFICIENT SHOCK ABSORPTION SYSTEM FOR AN AGRICULTURAL ELECTRIC TRICYCLE
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PROIECTAREA ȘI ANALIZA UNUI SISTEM EFICIENT DE ABSORBȚIE A ȘOCURILOR PENTRU UN TRICICLU AGRICOL ELECTRIC

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ABSTRACT
Ecological transport systems must be provided with efficient vibration damping systems for the comfort and safety of the user. This paper analyses a shock absorption system that can be used in an individual three-wheeled transport vehicle. The vehicle has a complex structure, with an equal size of the front and rear wheels. This uniformity of dimensions between the rear and front wheels makes it easier to travel on rough terrain and manoeuvre in a folded shape. The tricycle allows aggregation with different agricultural equipment and can be used in small farms, greenhouses, solariums, meadows, orchards, etc. In this paper we simulate several models of absorption systems with different construction parameters. The strength of the system and the efficiency of shock absorption were taken into account. The best result of the simulation test for absorption systems will be the comparison with the actual physical model used by the electric vehicle.

REZUMAT
Sistemele de transport ecologic trebuiesc prevăzute cu sisteme eficiente de amortizare a vibrațiilor pentru confortul si siguranță utilizatorului. In lucrarea prezenta se analizează un sistem de absorbție a șocurilor care poate fi utilizat într-un vehicul individual de transport pe trei roți. Vehiculul are o structură complexă, cu o dimensiune egala a roților din față și din spate. Această uniformitate a dimensiunilor dintre roțile din spate și cele din față face mai ușoară deplasarea pe teren accidentat și manevrarea într-o formă pliată. Tricicleta permite agregarea cu diferite echipamente agricole și poate fi folosită în mici ferme, sere, solarii, pajiști, livezi etc. În această lucrare se realizează simularea mai multor modele de sisteme de absorbție cu diferiți parametri constructivi. S-a avut în vedere rezistența sistemului și eficiența absorbției de șoc. Cel mai bun rezultat al testului de simulare pentru sistemele de absorbție va fi comparația cu modelul fizic real utilizat de vehiculul electric.

INTRODUCTION
The need for agricultural equipment caused by the growth of the world's population is more than obvious. Current agricultural equipment has reached its limits of optimization in terms of complexity and efficiency with current technology. Moreover, improvements in the field of drive technology today, mainly mechanical or hydraulic drives, are limited. Therefore, the focus in this area will be on electric units in the future.

Reducing pollutant emissions and dependence on fossil fuels is a global goal. In many countries, governments promote the use of efficient vehicles such as electric and hybrid vehicles. The development of electric agricultural equipment has advantages in terms of increased energy efficiency and extended functionality. Greater efficiency means reducing fuel consumption and subsequently reducing CO_2 emissions. The new functionalities improve the quality of work and increase the comfort of the operator. The major advantages of electrifying agricultural machinery are: electric motors can have an efficiency of about 90% compared to diesel engines powered by diesel, a maximum torque from time 0, lower maintenance costs, noise reduction, low pollution and a more flexible design.

Such a vehicle can be the electric tricycle, with low production and maintenance costs, reliability for transport, its mechanics being relatively simple, having better stability than the motorcycle and possibilities of adaptation for multiple utilities such as, in this case, operation in small farms, greenhouses, solariums, meadows, orchards etc. (*Suvac A, 2019)*

The systems for suspension that integrate in their structure the rubber is encountered in many studies because this optimizes the control of the shocks and the vibrations. For example, we can mention the study conducted by *Zhao L.L. et al., (2018),* where, for improving the performance of a vehicle's seat, was created a new type of suspension composed of springs made of composite material based on rubber. This implementation has led to a low-frequency vibration attenuation generated by difficult terrain.

For vehicles used for rail transport are used, more and more often, as secondary suspension system, pneumatic elastic elements that contain rubber *(Spiroiu M.A, 2018)*. For trucks, the rubber elements are designed to provide a reduction of the engine vibration and the transmission of the vibrations to passengers. (*Hur S, 2017*).

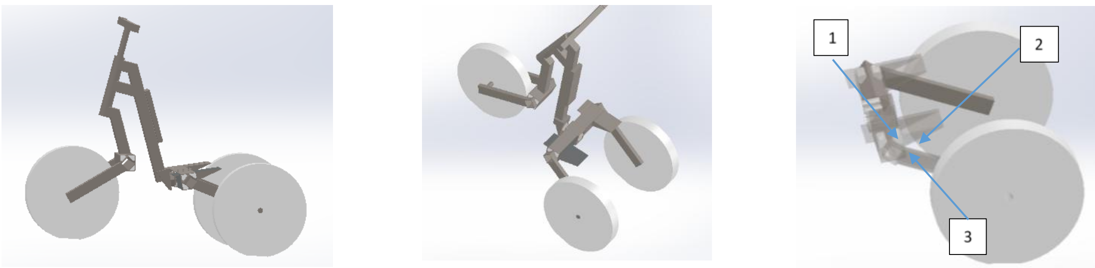
In aeronautics, the problem of complex setup can be solved by generating new models based on qualitative semantic models. The approach of this analysis requires the development of new models and methods for obtaining information regarding the shape of the modeled object, which captures greater importance than its structure (*Tomilov I., 2015*)

In the case of autonomous vehicles, in order to improve the comfort and handling achieved through the dissipation of the energy induced by terrain, have been designed other types of suspension systems. The relevant results in scientific literature, for example studies made by (*Nielens H., 2004*) are focused on mechanical simulations, as well as on structures that have a better energy absorption. Other studies (*Nielens H., 2001*) indicate that we should be cautious before using multiple suspensions regardless of terrain because the handling becomes more difficult. Also, since these systems ensure a good efficiency of shocks absorption generated by the terrain irregularities, and, as a consequence, a greater comfort is obtained, they can generate, an increase in passenger's effort, especially for cyclists.

Unfortunately, studies show that the passive vibration absorption systems, which use viscoelastic materials due to their non-linear characteristics with temperature, frequency and mechanical stress, lead to nonlinear dynamic properties (*Neto F.P.L., 2012*). Thus, in the design of the reliable vibration isolator system from the point of view of the mathematical model and of the optimal design, it is necessary to determine the degree of rigidity and absorption of the viscoelastic material influenced by shape and structure.

MATERIALS AND METHODS

In this paper, three different vibration damping structures and the mechanical vibration damping intended for use in an individual electric tricycle vehicle (Fig. 1) were analysed. The systems are used basically at the level of the foot sustaining parts, but they can be used at the direction.

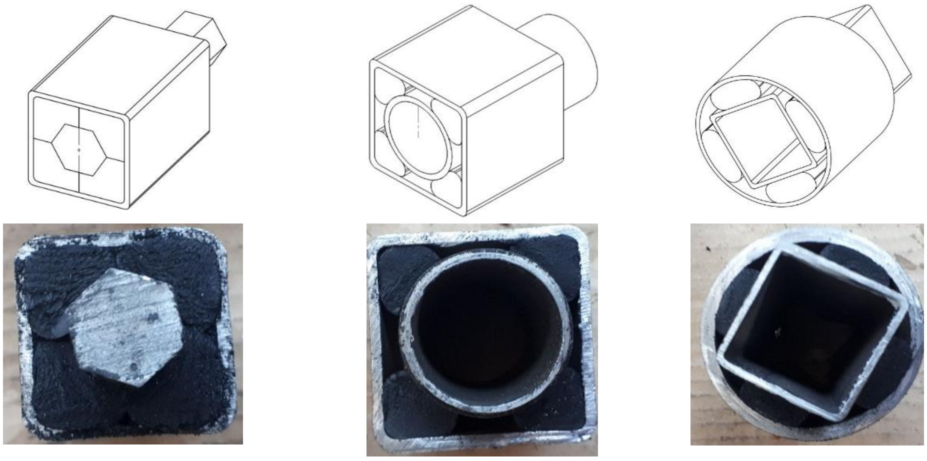


a) b) c)

Fig. 1 - 3D model of the individual electric tricycle vehicle:
a) lateral view; b) isometric view; c) system structure
1 -external part, 2 – rubber, and, 3- central part

The three constructive solutions are presented structurally and physically in Fig. 2.
The characteristic elements are:

- exterior part is a rectangular shell with hexagonal solid central part, see Fig. 2a;
- exterior part is a rectangular shell with round shell central part, see Fig. 2b;
- exterior part is a round shell with rectangular shell central part, see Fig. 2c.



a) b) c)

Fig. 2 - Different constructive solutions for vibration damping structures

a) exterior part is a rectangular shell with hexagonal solid central part; b) exterior part is a rectangular shell with round shell central part; c) exterior part is a round shell with rectangular shell central part

All the solutions for static mechanical load were tested with the sole purpose of validation of the absorption system design structure. In this first step, the performance of the system isn't taken into account, because we were interested only in the functional mechanical load, which means applying a moment of torsion on the central part. For this purpose, a testing stand was built (Fig. 3).

Three shape details were presented in Fig. 4a, 4b and 4c. Element 1 is fixed on the Table 4 and the element *2* is a bar with the length of 1m which has a system for measuring the angle *5* (considering the horizontal line as reference). The calibrated weight 3 is similar with the user's distributed weight on the legs. After a set of preliminary tests, a maximum mechanical load corresponding to 30kg was used. For this purpose, a fastening part was constructed to allow the installation of each exterior elements of the tested structures. On the central part a lever with 1m length was fastened, and to the free end we attached gauges with pre-set weights.

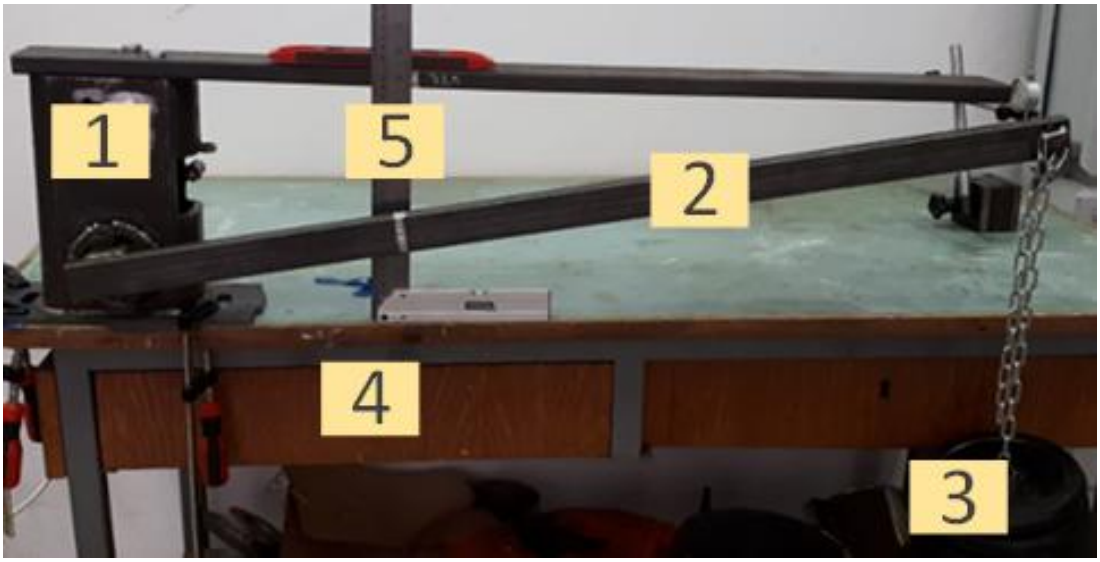


Fig. 3 - Experimental testing stand for static mechanical load (physical view)

1) fastening device for the structure to be tested; 2) bar with the length of 1m; 3) calibrated weights; 4) table; 5) system for measuring the angle of the bar (considering the horizontal as reference) and the height

In the first case, the external part is a rectangular and the central part is the solid hexagon (Fig 4a). Due to the compact structure, neither the external part, nor the central part is deforming and thus the rubber will have a large deformation under mechanical load considered and has a null elastic characteristic.

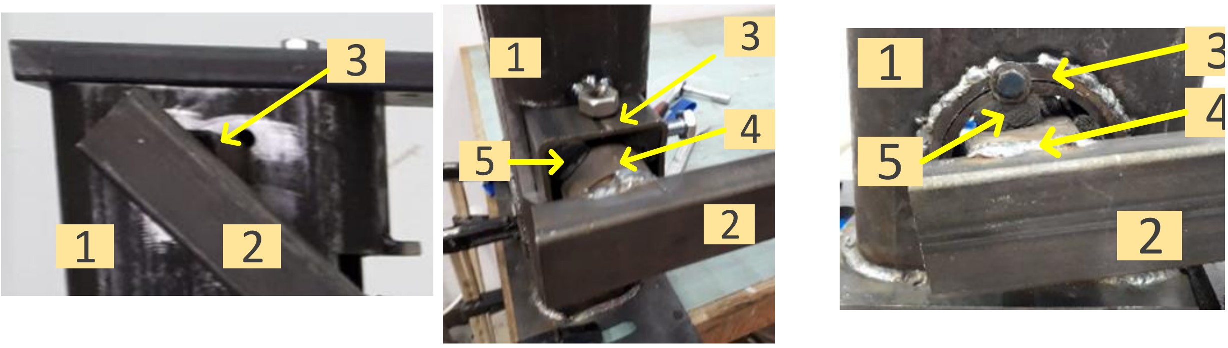
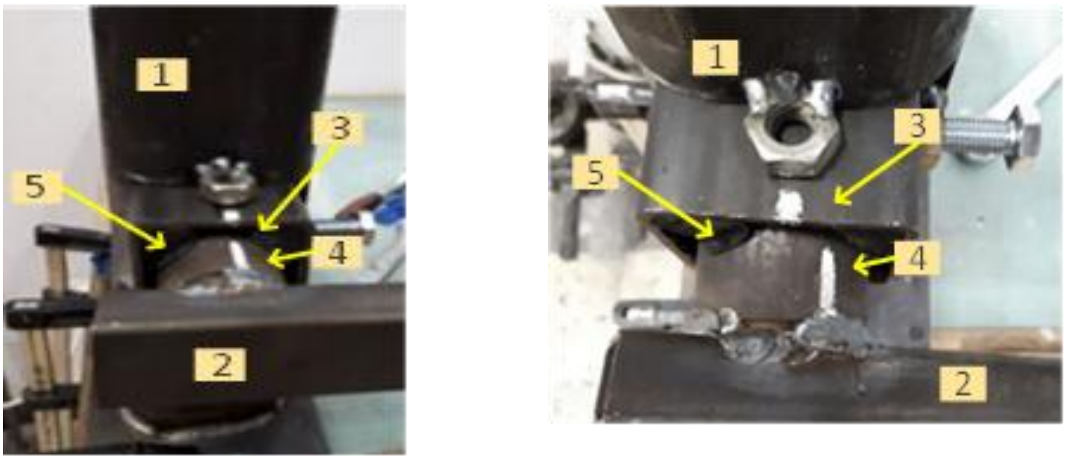


Fig. 4 - Experimental testing stand for static mechanical load

a) exterior part is a rectangular shell: 1) fastening part of the absorption structures; 2) bar used for applying the moment to the inner part; 3) hexagonal solid central part (hidden);
b) exterior part is a rectangular shell: 1) fastening part of the absorption structures; 2) bar used for applying the moment to the inner part; 3) the outer part of the damping system; 4) round shell central part; 5) rubber;
c) 1) fastening part of the absorption structures; 2) bar used for applying the moment to the inner part; 3) exterior part is a round shell; 4) with rectangular shell central part; 5) between the parts there is rubber

Considering the structure where the exterior part is a rectangular shell with a round shell central part (Fig. 4b), we didn't achieve the expected results. We expected the central part to return to the initial position. But, because the structure does not permit to the central part to have an elastic deformation, this part slides on the rubber part and the whole structure isn't returning to the initial position, as can be seen in Fig. 5.

Figure 5 is used only to demonstrate this behaviour, using a white paint on the exterior part and another on the central part.

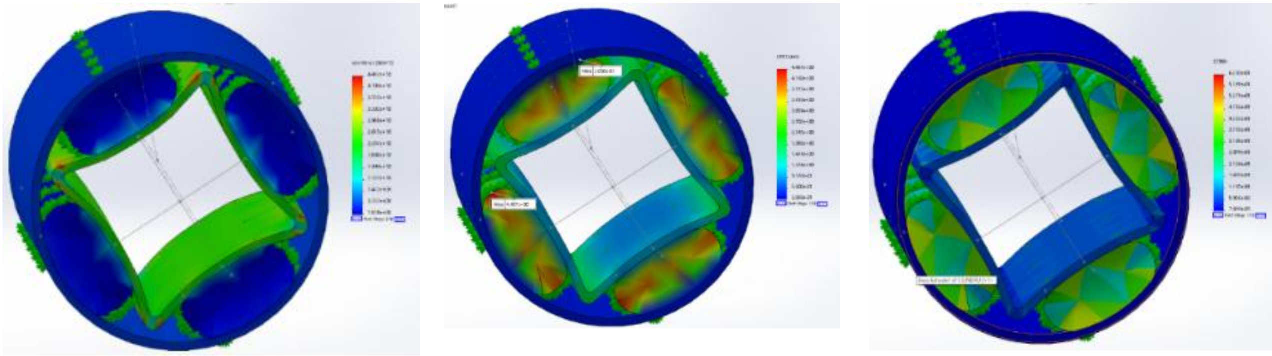


a) *b)*

Fig. 5 - Experimental testing stand for static mechanical load

a) Before applying loads; b) Testing stand under stress conditions
1) fastening part of the absorption structures; 2) bar used for applying the moment to the inner part; 3) exterior part is a rectangular shell; 4) round shell central part; 5) rubber

In the third case, the structure is made by a round shell as exterior part and rectangular shell as central part. This structure had the best results in the first mechanical test. This is the reason why a dynamic theoretical analysis was realized only for this structure. The experimental tests have been confirmed by the finite elements simulation results. For this test, a 3D model using an automated fine meshing with 12524 nodes and 4623 elements was used. Each type of elements is tetrahedral. The exterior part is fixed and a mechanical load of 300Nm was applied on the rubber central part. Fig.6 shows how the central part deforms by adding, as effect, an elastic component to the structure.



a) b) c)

Fig. 6 - Finite elements simulation results

a) mechanical strain; b) deformation, c) mechanical

Dynamic analyses

Due to the mechanical structure of the vehicle, the shape of the user's weight (dumping part which can be put under the feet sustaining part as can be seen in Fig. 7 and the oscillation of terrain level, three types of mechanical load were identified:

- compression of rubber elements;
- bending of the exterior part when the central part is fixed;
- bending of the central part when the exterior part is fixed;
- torsion of the central part when the exterior part is fixed, as in static tests and FEM presented in Figure 6.

The first case analysed is the compression caused by the system itself. The system is much longer than the dimensions of the section of the frame. The load of the user's body is uniformly distributed by the feet sustaining part (position 1 and 2, Fig. 7) which has, at the central part, a connection to the direction system. We will consider this harmonic load because although the mass is unchanged when the terrain is flat, but it has some roughness, due to the dynamics of the system the load varies because of its vertical displacement.

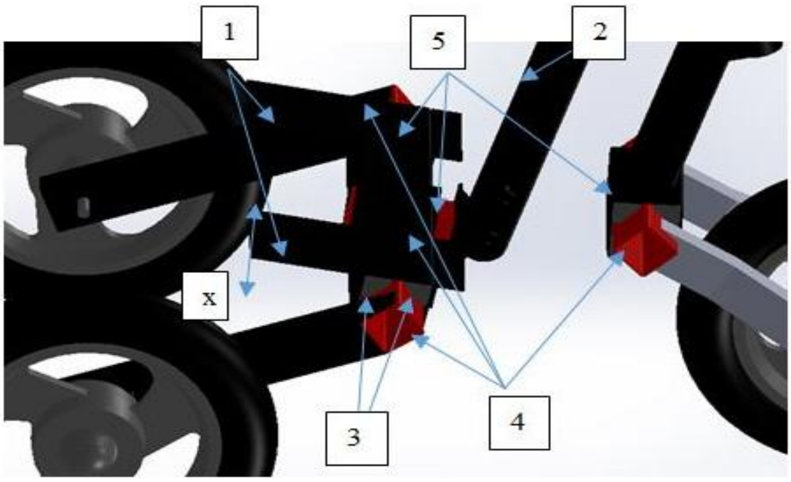


Fig. 7 - The components of the absorbing system

1) position of feet; 2) the body support part; 3) rubber;
4) central part (the profile can be: hexagonal, rectangular or round); 5) exterior part

Since the structure has no bearing and no clearance, we can say that we are dealing with a system with linear cubic rigidity. We can write, in this case, from a mathematical point of view the elastic component of the system (*Radeș M., 1985):*

$$f_e = k(x + \mu x)^3 \quad (1)$$

where f_e is the elastic force;

x - vertical deformation of the system;

k – stiffness;

μ - coefficient of negative nonlinearity since the tests showed that the system has a descending slope so, it has soft feature of the elastic component.

We consider that, when the vehicle is in motion, the force generated by the weight of the passenger is harmonic with the magnitude F_0 and a small pulse ω because of a very low frequency of the several Hz that will result in a harmonic displacement. Since we have separated the types of mechanical load, we can consider having a system with a degree of freedom consisting of a nonlinear spring because of the central part deformations and a dissipative part consisting of rubber elements.

The motion equation in the form considering only the transversal vibration of the central part is *(Radeș M., 1985)*:

$$m\ddot{x} + \frac{g\cdot k}{\omega}\cdot\dot{x} + k(x+\mu x^3) = F_0 e^{i\omega t} \quad (2)$$

In Eq. 2, g is an equivalent structural absorption factor. We're interested only in the first harmonic and the displacement can be described in function of vector of the movement which has real component (a_R) and imaginary component *(a_I):*

$$x = \tilde{a}e^{i\omega t} = (a_R + i a_I)e^{i\omega t} = a e^{i(\omega t+\theta)} \rightarrow x^3 \cong \frac{3}{4}a^2 x \quad (3)$$

The real component $(a_R)$and imaginary component $(a_I)$of the system are:

$$a_R = \left(1 + \frac{3}{4}\frac{\omega}{\omega_n}a^2 - \frac{\omega}{\omega_n}\right)\frac{k}{F_0}a^2 = F\sqrt{a^{2-\left(\frac{gka}{F_0}^2\right)^2}} \quad (4)$$

$$a_I = -g\frac{k}{F_0}a^2 \quad (5)$$

Considering that the displacement amplitude and the phase θ angle we can describe the vector of the movement:

$$a=\sqrt{a_R^2 + a_{R1}^2} \quad (6)$$

$$tg(\theta) = \frac{g}{\eta^2 - 1\frac{3}{4\mu}\mu a^2} = \frac{g}{\pm\sqrt{\frac{F_0^2}{k^2a^2} - g^2}} \quad (7)$$

Where η is the loss factor defined as the ratio between the energy dissipated in a vibration cycle and the maximum potential energy, accumulated by the system in that cycle *(Radeș M., 1985)*.

We rewrite the equations according to the imaginary and the real part, and we obtain:

$$\left(\frac{\omega}{\omega_n}\right)^2 = 1 + \frac{3}{4}\mu\frac{F_0}{k}(-a_I) \pm g\sqrt{\frac{F_0}{gk}\frac{1}{(-a_I)} - 1} \quad (8)$$

$$\left(\frac{\omega}{\omega_n}\right)^2 = 1 + 3\mu\left(\frac{F_0}{2gk}\right)^2 - \frac{3\mu\left(\frac{F_0}{2gk}\right)^2 a_R^2 + \frac{F_0}{k}a_R}{2\left(\frac{F_0}{2gk}\right)^2\left[1\pm\sqrt{1-\left(\frac{2gk}{F_0}\right)^2 a_R^2}\right]} \quad (9)$$

Equations describing a family of curves above the curve leaning towards the low frequencies:

$$\omega^2 = \omega_n^2\left(1 + \frac{3}{4}\mu a^2\right) \quad (10)$$

The pulsation of the system should be different from the natural pulsation of the system which is $\omega_n=\sqrt{\frac{k}{m}}$. In our case we obtained from the finite element model that its value was 0.07234.

We can define the stability limit as:

$$\left(\frac{\omega}{\omega_n}\right)^2 = 1 + \frac{3}{4}\mu a^2 \pm \sqrt{\frac{9}{16}\mu^2 a^4 - g^2} \quad (11)$$

$$\eta\left(\frac{\omega}{\omega_n}\right)^2 = 1 + \frac{g}{2}\left(tg(\theta) + \frac{3}{tg(\theta)}\right) \quad (12)$$

The following type of load is torsional for the entire structure. In our case the motion equations are (*Cârdei P, 2010*):

$$\ddot{\beta} + k_{0i}1\cdot\beta + k_{1i}\cdot\beta^3 + c\beta' = M(t) \quad (13)$$

where $i = \begin{cases} 1 \text{ at charging} \\ 2 \text{ at discharging} \end{cases}$

with the original conditions

$$\beta(t_0) = 0 \text{ and } \dot{\beta}(t_0) = 0 \quad (14)$$

Experimentally, we obtain the values between the moment of torsion and torsion angle and, based on these values, the chart presented in figure 8 was build. It can be seen, in figure 8, that there is a small hysteresis, which hasn't been taken into account, and using only the trend line, the resistant moment *M(t)* was identified:

$$M(t) = 7.345\beta - 15.387 \tag{15}$$

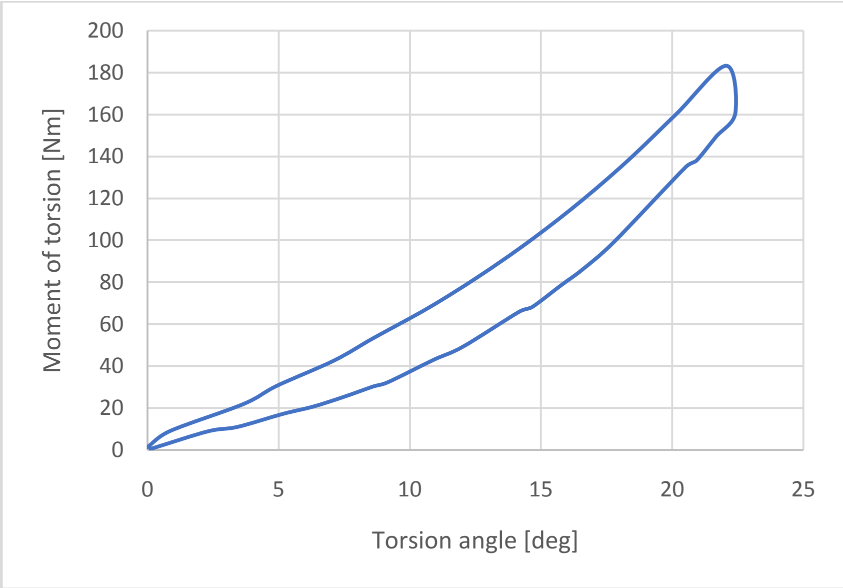
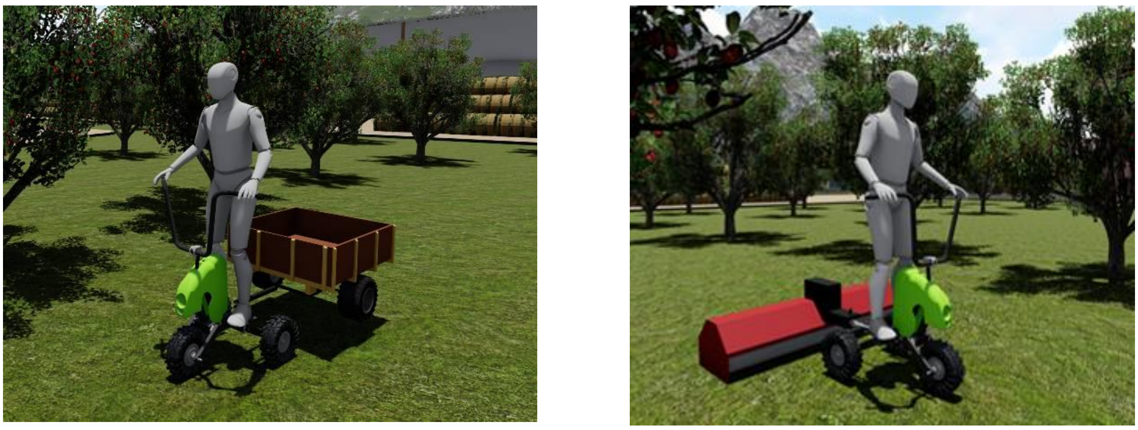


Fig. 8 - Diagram of the torsion moment vs torsion angle

CONCLUSIONS

In the article were presented three systems that can be used in individual electric vehicles, designed to absorb the mechanical shock and to reduce vibrations. The experimental tests with static mechanical loads identified the most efficient structure and it was validated by the finite element analysis. Also, for this structure were analysed different types of simple dynamic load which will be integrated into a complex mathematical model that can be used into other similar structures.

In the following research we will consider the bending and we'll analyse the simplified case in which a wheel does not move and the other moves under the action of the force generated by the deformations of the land when the vehicle is in the movement and the third type of mechanical load in the case of which we'll consider the effect of the central part.



a) *b)*

Fig. 9 – 3D tricycle rendering

a) Used for a small trailer traction *b) Used for a mower traction*

Fig. 10 – Testing tricycle prototype

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