INTRODUCTION

The solution of the passive suspension system these 2S1 tracked platform is based on torsion bars which are problematic elements. Problems with this elements are associated with their length and placement of them in the vehicle. Because of their reactive character, which during explosion of an anti-tank mine may destroy the suspension system, they should be replaced with another solution. This alternative solution may allow to replace reactive elements from under floor space to the vehicle sides. The most popular are the helical springs. They were analyzed by Taktak et al. [14], with usage of the Finite Element Method (FEM). FEM computer applications are a good solution also for the analysis of other kinds of springs. The helical springs could be problematic in application to the tracked vehicle, which was previously based on torsion bars. The FEM model application for various kinds of spring was developed in collective work edited by Shimoseki et al. [12]. The best solution is application of torsion springs. At this point, a question appears. This question is associated with the kind and construction of above mentioned torsion springs. The answer will be put forward by the authors in the first section of this article. But before this, the authors analyzed constructions of full tracked vehicles and also modeling solutions, which may be applied in the tracked vehicle model. Complex vehicle model was described by Campanelli and Shabana [2]. For analysis of the track link natural frequencies mode they use FEM. Less complex models of tracked vehicles were described by Nabagło et al. [8, 9]. The model developed in [9] was based on proving ground measurements described by Jurkiewicz et al. [4]. This model included many important elements of tracked vehicle model. One of the most important problem in this area is connected with joints between track segments. Ryu et al. [11] takes under consideration single pin track link. He describes the so-called rubber bushing model, which connects two track links. Lee [6] in his article describes contact forces between wheels, track links and ground surface. There is also explained a numerical method of force contacts calculation. Details of this connection forces, contact forces between wheels, track links and road surface and also FEM model construction of the flexible body are described in ADAMS Documentation [1]. Selected parts, with high fidelity of real parts shapes, may be designed in other CAD program. Łukaszewicz [7] in his article, explains multi-body CAD designing problem for simulation applications. For springs, the failures and fatigue damage are very serious problem. The expression of this problem was described by Prawoto et al. [10]. They analyze not only construction but also materials from which the springs are built. As it was said, the basic approach to the modeling solution is also important. The authors of the presented article used tools provided by MSC.ADAMS and MATALB programs, but they were also familiar with other solutions. These modeling solutions were described by Struski et al. [13] and Grzyb et al. [3]. Because the contact between spring arms is a serious problem in the torsional springs models, very valuable information was put forward by Lakarani et al. [5]. He described a detailed parameterized model of the contact force. This model is usually applied in the ADAMS program. Above mentioned models of subsystems may be connected.

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as parts of tracked vehicle model applied in the dynamical simulation, similar to the simulation described by Wong [15].

1 ANALYSIS OF EXISTING SOLUTIONS

During research many constructions of the torsion spring were analyzed. There were construction of Archimedes spiral shape spring (see Fig. 1a), logarithmic spiral shape spring (see Fig. 1b) and here presented hyperbolic shape spring. The main aim of research was to construct a torsion spring with stiffness characteristic as similar to initially used torsion bar characteristic as possible. Because of failures and fatigue damage of previously tested springs, it was not easy. Problems were associated with construction but also with mounting of the springs external ends. Finally, the authors used an advanced mounting system, which solved these problems. This mounting system was associated with the sliding contact between spring arms and pins placed in triangular grooves on their ends. This solution caused that stiffness characteristic were very close to the torsion bar characteristic but in very narrow range of suspension displacement. After angle about 20 degrees, in the spring, the contact forces between spring arms appeared. This phenomena caused change in the spring characteristic, turning it into a nonlinear stiffness characteristic. In the case of logarithmic shape spring also effects of the fatigue processes appeared, which finally caused damage of the spring arms in close neighborhood of the suspension arm axle.

Fig. 1. Archimedes spiral shape (a) and logarithmic spiral shape (b) springs [4]

Problems of too early appearing contacts between spring arms were overcome by the application of so called hyperbolic shape spring (see Fig. 4) with advanced system of leading links. These links lead external ends of the spring arms in order to achieve an appropriate stiffness characteristic. The shape of this characteristic is identical to characteristic of the torsion bar multiplied by constant equal to 2.6. In other words the stiffness of the torsion spring is 2.6 times bigger than stiffness of torsion bar but with the same nature of changes, when the angle of deflection increases. Contact forces appear
after angle 40 degrees, but without noticeable influence. Radical increase of stiffness, because of contact influence, appears after exceeding the value of 60 degrees.

2 MODELING OF THE SUSPENSION SYSTEM

Finally, the designed hyperbolic shape spring should be applied and verified in the tracked vehicle suspension system. This system was already built and described in articles [4,8,9]. In the articles the full tracked vehicle models are described. There are models with torsion bars, and also with torsion spring packages. Modeling assumptions

To construct the stiffness characteristic of the torsion bar, the characteristic reduced to the axle of the road wheel was measured. This characteristic initially has so called tolerant area which was associated with hysteresis loop. This loop was the result of dry friction, which caused damping effect. Therefore the mean values from this tolerant area are equal to the values of stiffness characteristic of the suspension system. The result is shown in figure 3a, as the compression spring stiffness characteristic. Because this compression spring is theoretically placed between wheel axle point and the vehicle body, the real torsional stiffness had to be calculated (see Fig 4 a and b). As a result of this calculation the torsional spring stiffness characteristic was presented in figure 3b. This characteristic has become a reference for the later obtained results.

![Fig. 3. Suspension stiffness reduced on the wheel axle (a) stiffness characteristic of torsion spring (b)](image)

The torsional stiffness characteristic was calculated according equation (1), shown below.

\[ T(\alpha(x) + \alpha_0) = d \cdot \cos(\alpha(x)) \cdot F(x) \]  

(1)

where: \(x_0 = d \cdot \sin(\alpha_0)\) and \(\alpha(x) = \arcsin \left( \frac{x-x_0}{d} \right)\).

The natural frequency of the sprung mass of the vehicle should be close to \(f_n = 1.25\) Hz. On the base of equation (2), the whole sprang mass of the vehicle was calculated.
\[ f_n = \frac{1}{2\pi} \sqrt{\frac{k_u}{M_u}} \]  (2)

where:

- \( f_n \) – natural frequency of the system,
- \( k_u \) – suspension stiffness coefficient,
- \( M_u \) – sprung mass of the single suspension unit.

\( M_u \) is part of the sprung mass associated with a single suspension unit. Because of 14 road wheels, the vehicle has 14 units like this. The unit sprung mass is described by equation (3)

\[ M_u = \frac{k_u}{4\pi^2 f_n^2} \]  (3)

When linear stiffness, based on the characteristic (see Fig 3a), is equal 50 kN/m, then single unit sprung mass is equal \( M_u = 810 \) kg. This mass must be multiplied by 14 and the whole sprung mass of the vehicle should be equal to 11340 kg. Finally 11570 kg was assumed. Based on simulation results of hyperbolic spring (see Fig 5b), the stiffness characteristic of this spring was calculated. The simulated model is described in the section 2.2. The system of the hyperbolic spring on the figure 5b is not complete. Some details of the spring mechanism is not published because of confidentiality. As is shown in figure 5a the hyperbolic shape spring is 2.6 times stiffer than the torsion bar. Then the linear spring reduced on the wheel axle, for the hyperbolic spring system is equal 130 kN/m. In continuation this way of thinking and based on equation (3), the unit sprung mass is equal to 2107 kg. These parameters may be applied in the full vehicle model. Static displacement with such mass and stiffness is described by equation (4) and it is equal 0.16 m.

\[ z_{st} = \frac{M_u \cdot g}{k_u} \]  (4)

where:

- \( g \) – gravitational acceleration, 9.81 m/s\(^2\).

This translational displacement is adequate to the angular displacement, which is equal 30 degrees.

Fig. 5. Comparison of the stiffness characteristic (a) and hyperbolic shape spring (b)

### 2.2 Simulation of the FEM spring model

To estimate technical parameters of the hyperbolic shape spring, the FEM model of the spring was built and simulated. Traditional FEM model of a spring is problematic, because during simulation when the spring is twisted, the subsequent arms of the spring interpenetrate. The model presented in this article is enriched with contact elements between subsequent arms. The contact stiffness is equal 100000 kN/m with force exponent coefficient 2.2. This contact has also damping coefficient, which is equal 10 kNs/m. This damping is reached after exceeding a penetration, which is equal to 0.1 mm.
Details of this contact model was described by Lakarani et al. [5]. These contact parameters are close to contact between two surfaces of steal elements. This contact model neglects the friction phenomena. This situation is close to the real conditions when steal torsional spring is immersed in oil and friction may be neglected. The FEM model realizes Finite Element Method algorithm. This algorithm in its generalized form is described below. It realizes subsequent steps, which lead to the final FEM model.

1. Discretization, meshing,
2. Formulation of stiffness matrices for structural elements,
3. Assembly of the global stiffness matrix,
4. Assembly of the global load vector,
5. Definition of boundary conditions.
6. Solving the system of equations,
7. Determining internal forces and reactions.

Given the stiffness matrix $K$, the static equation has the form (5).

$$ K \cdot x = F $$

where:
- $K$ – stiffness matrix,
- $x$ – displacements vector,
- $F$ – force vector.

Chosen steps of the simulation of the above described model are shown in figure 6. It contains initial position of the torsional spring for angle 0 degrees (see Fig 6a). In the figures from 6b to 6d, the results of the classic FEM simulation are presented. In figure 6d, the material stress may be noticed, especially on the external ends of the spring arms.

**Fig. 6.** Subsequent steps of simulation of the FEM model

From figure 6e, where angle of 40 degrees is exceeded, the influence of the contact forces appears. Vectors of these forces are described with red arrows and the arrow length is proportional to the values of the forces. It may be noticed in comparison with figure 5a, where the stiffness characteristic
of the torsional spring is presented, the contacts have negligible influence on the stiffness characteristic shape. It is true to the moment when external ends touch the closed spring. This condition is fulfilled when the angle exceeds 60 degrees. After exceeding this angle stiffness radically increases, which may be adequate to situation when the system could be blocked. In figure 6f, high stresses are visible. They are visible on the external ends of the spring arms. It may suggest the necessity of making some changes in the spring construction.

2.3 Full vehicle model

In the full vehicle model many factors have to be taken into consideration. There are model parameters but also environment factors, like for example ground stiffness. This element has huge influence on the ground-track contact parameters, which is tightly associated with the vehicle dynamics [15]. In the model, the authors assumed that the contact forces have linear characteristic. The contact forces appear between wheels and track links and also between the track links and the ground. The contact is characterized by two main parameters, namely stiffness and damping. The stiffness parameter of contacts between wheels and tracks equals 200 kN/m, the damping equals 2 Ns/mm. The stiffness parameter of contacts between tracks and ground equals 1000 kN/m, the damping parameter equals also 2 kNs/m.

In the final torsional spring element, because of heat exchanging problem, the above described hyperbolic shape spring will be divided into several parallel connected torsional springs. They will be connected on the so called spring packages. In the full vehicle model, the packages are mounted in place of the removed torsion bars. The spring packages are mounted outside the hull, close to cooperating suspension arms. Thus additional free space is provided. This space could be used for installing better floor armor or other equipment. This solution improves also safety, because all reactive elements of the suspension system are removed from the floor plate and they are placed at both sides of the hull, where they may be better covered.

As was said before, in the model with torsion bars, the mass of the hull was assumed $11.57 \cdot 10^3$ kg. According to calculations from section 2.1, the hull mass increased to $29.5 \cdot 10^3$ kg and this hull mass was applied in the final model (see Fig. 7). In the above described models, also other systems were applied. Idler wheel mechanism acts on the track with a constant force, which equals 39 kN. This force acts in longitudinal axis of the vehicle, as is shown in figure 7. This solution is used for maintaining a constant tension of the tracks.

![Fig. 7. Model of the full tracked vehicle](image)

It is simplified version of real idler wheel mechanism which is based on eccentric mechanism. However, this simplification realizes the main aim of the constant track tension. Forces, acting on the idler wheel from the track side during acceleration or deceleration, may be higher than the idler force value. In this situation these forces cannot be balanced with 39 kN. As a result of it, the idler wheel may be moved to the front of the vehicle. For this reason, a bumpstop is used to stop the movement of the idler wheel. It is stopped after displacement 100 mm from the construction position. This
modernized model will be applied for future simulations of the behavior of full vehicle model with applied hyperbolic shape springs.

CONCLUSIONS

Summarizing the results of the simulation, the pros and cons of the presented solution of torsion spring should be put forward. Undoubtedly, the advantages of the solution are the dimensions and the stiffness characteristic, which reflects the shape of stiffness characteristic of the torsion bar. These torsion bars were originally mounted in the vehicle. Its additional advantage is its higher stiffness, which allows to significantly increase the weight of the vehicle body. As is known in the case of military vehicles, this is a huge advantage. Using the described springs, the vehicle will be able to be used as a platform for heavy guns or as a transport vehicle for transportation of heavy equipment.

The disadvantage of this solution is a limited range of torsion of the spring, which may reach the maximum angle of 60 degrees. As is apparent from the suspension deflection in the static equilibrium state, this deflection is reached for angle of 30 degrees. Dramatic increase of the stiffness appears after exceeding angle of 60 degrees. Between these two angle positions, the range of 30 degrees still exists. Therefore, the deflection of 30 degrees is an angle from static equilibrium position to the position of the bumpstops. Thus, the above-mentioned increase of stiffness, perhaps may even support the effects of bumpstops activity. A phenomenon, which should be noted, is the material tension, which appears on the ends of the spring arms. But it can be significantly reduced by using the appropriate design solutions.

Because of higher stiffness level of spiral spring packages than stiffness of the traditional torsion bars, the constructors can increase the vehicle hull mass. This allows also to resign from axial offsets of the road wheels, so the right and left wheels may be mounted on the same axles. Usage of spiral spring packages gives more space between wheels. This space may be used for additional equipment or even additional armor. Usage of spiral springs increases also safety level, because the torsion bar solutions are associated with reactive elements of suspension under the floor of crew cabin. During explosion of a charge placed under the vehicle, these reactive elements could be harmful for the crew. The spiral spring packets are placed on both side of the vehicle and could be covered.

Abstract

In the paper the authors present an advanced model of the hyperbolic shape torsion spring, which is based on the finite element method. In this model, the contact forces between the arms of the spring were taken into consideration. It allows the simulation of its operation even in the case of a large torsion angle. Based on the FEM model of the spring, the spring stiffness characteristic was constructed. Based on this characteristic, an existing model of the 2S1 tracked platform has been redesigned. In this model, a much higher weight of the vehicle body is possible than in the original model case. Based on the flexible spring model, such parameters as the static deflection of the vehicle suspension as well as the range of suspension arm mobility were determined and analyzed. They were studied in comparison to the original suspension based on the torsion bars. In addition, phases of the spring deflection were presented to define internal stress, which are important for the stress analysis of the spring.

Keywords: torsional spring, FEM model, finite element method, ADAMS, flexible body

Modelowanie i symulacja zaawansowanej sprężyny skrętnej dla zawieszenia pojazdu gąsienicowego

Streszczenie

W artykule autorzy przedstawiają zaawansowany model hiperbolicznej sprężyny skrętnej w oparciu o metodę elementów skończonych. W modelu tym uwzględniono siły kontaktu pomiędzy ramionami sprężyny, co pozwala na symulację jej działania nawet w przypadku dużego kąta jej skręcenia. Na podstawie tego modelu MES została skonstruowana charakterystyka sztywności sprężyny. Na podstawie tej charakterystyki został przebudowany już istniejący model platformy gąsienicowej 2S1. W modelu tym dopuszczalna jest znacznie większa masa nadwozia niż w modelu pierwotnym. Na podstawie elastycznego modelu sprężyny zostały również określone takie parametry jak ugięcie statyczne zawieszenia jak również przeanalizowany został jego...
zakres ruchliwości w porównaniu do pierwotnego zawieszenia opartego na wałkach skrętnych. Dodatkowo przedstawione zostały fazy ugięcia sprężyny pod kątem naprężeń wewnętrznych, mających istotne znaczenie dla analizy wytrzymałościowej sprężyny.

**Słowa kluczowe:** sprężyna skrętna, model MES, metoda elementów skończonych, ADAMS, bryła elastyczna

**REFERENCES**