MODELLING OF TRACKED VEHICLE DYNAMICS

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Abstract

The paper presents the pr oblem of modelling dynamics of motor vehicl es in the conv ention of multi-segment systems. The ass umptions used in modelling pr ocess are presented and the characteristics of elastic - damping elements of tracked vehicle suspension system are determined. This paper presents the identified mass and the geometrical parameters of the vehicle, on the basis of these parameters in Virt ual.Lab software environment the simulation model was developed. The model describes the elements of a tank t hat does not change their position relative to the reference coordinate system, they are: the turret, muzzle, body case , and other components such as: wheels, torsion shafts, damper levers and links. The tank tracks are omitted in the simulation model. The results of numerical simulations in the form of displacement courses of characteristic points of the hull and displacements of the wheels axles in an assumed reference coordinate system are also included. The simulation results were compared with experimental results obtained on the pr oving ground. The results of experiment al measurements carried out on the tank range, w ere used to precise tune a mathematical model of the vehicle. The results of numerical simulations confirmed validity of the assumptions in t he modelling process, since good agreement was obtained with results from experimental investigations. The developed models will be used to optimize the characteristics of vehicle suspension.

Keywords: multi body, trucked vehicle, numerical simulation, dynamical analysis, suspension

1. Introduction

Issues of heavy body tracked vehicle trajectory stabilization when driving at high speeds, especially in the field conditions, are the primary concern of armoured vehicles designers.

Excessive vibration of a moving body cause crew fatigue and difficulties in stabilizing the weapons, thereby reduce the manoeuvrability and indirectly survivability of the vehicle.

In addition, the base body is usually the basis for the design of the entire family of vehicles with different masses, sizes and purposes, often additionally armoured.

It is therefore extremely important task in the design and construction process to proper align and tune the vehicle suspension components and select the elements for energy dispersion.

Tracked vehicle is a complex mechanical system, which in general case consists of several dynamic subsystems, including the various types of mechanisms.

In the operation of tracked vehicles the important role plays a structural rigidity of the vehicle body and the dynamic effects occurring in kinematic pairs of the suspension mechanisms.

Properly selected model assumptions and mathematical models of mechanical systems, therefore, play an important role in proper description of dynamic phenomena, and further simulation investigations.

Previously, when using the classical approach to design, usually one assumed that the vehicle is a system of rigid solids combination [7-9], for which the movements description were made using the generalized coordinates, the solids are linked between each one by spring - damper elements.

This approach allows only the investigation of system vibrations, which are the oscillations near the equilibrium position.

Suspensions of tracked military vehicles represent a class of mechanisms that require the description of movement of its elements, often making large displacements, which is taking into account the complex interaction with the surrounding environment.

The use of formalism for multi-segment systems understood as many-body systems interconnected by different types of kinematic pairs, with internal or external forces acting on them, allows investigating the dynamic phenomena occurring in them, while the movement of the vehicle is carried out on any trajectory defined in inertial system coordinates.

Multi-segment system can be defined as a set of rigid or deformable bodies connected by kinematic pairs or force generating components.

Individual kinematic pairs allow relative motion between the bodies, while force generating elements represent the internal forces appearing between the bodies as a result of their relative motion.

External forces applied to each system elements may be either passive, or active forces.

Schematically multi-segment system is shown in Fig. 1

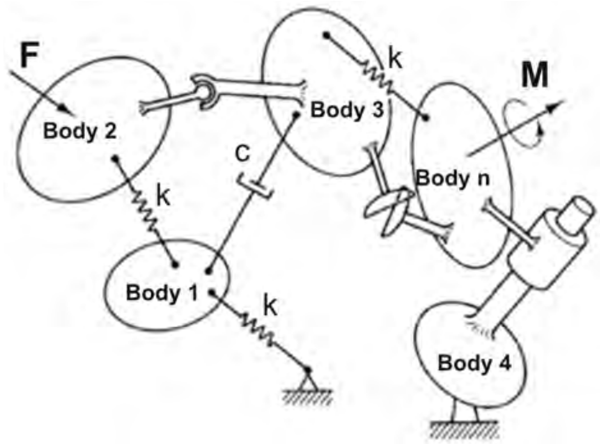

Fig. 1. Representation of solids in the multi-segment system

Analysis of multi-segment system dynamics [1] reduces to solution of equations of motion, which are second order differential equations, often occurring together with algebraic equations.

The former describe the motion of rigid or deformable bodies, while the latter are the constraints equations derived from analysis of system configuration and kinematic pairs.

The bodies occurring in the system may be rigid, in this case they have six degrees of freedom, or deformable, then generalized coordinates are added needed to fully describe their deformation [2].

Depending on the modelled system type and the type of coordinates system used, the number of coordinates may be greater than the number of degrees of freedom of multi-segment system.

In this case, the additional equations are required, to determine the relationships between the coordinates.

The system of equations derived from kinematic and driving constraints is compactly grouped into a global constraints vector Φ, written in the form:

$$\mathbf{\Phi}(\mathbf{q}, t) = \mathbf{0}\,, \tag{1}$$

where: **q** is a vector of generalized coordinates, and t represents time.

Time derivative of the constraints vector defines the relationship:

$$\dot{\mathbf{\Phi}}(\mathbf{q}, \dot{\mathbf{q}}, t) = \mathbf{0} \quad \equiv \quad \mathbf{\Phi}_{\mathbf{q}}\dot{\mathbf{q}} = \boldsymbol{\nu}\,, \tag{2}$$

where: $\mathbf{\Phi_q}$ is the Jacobian from a constraints matrix, $\dot{\mathbf{q}}$ is a vector of generalized velocities, and $\mathbf{\nu}$ is a vector containing partial derivatives of the equations differentiated respective to time:

$$\mathbf{\nu} = -\frac{\partial \mathbf{\Phi}}{\partial t}. \tag{3}$$

In the solids system subjected to movement restrictions, solids are linked by internal kinematic constraints.

Every kinematic constraint brings the reaction between solids combined, which corrects this movement so that it complies with the constraints terms [1].

Reaction forces in the constraints, also with respect to movement limiting forces, are determined by a vector gΦ.

The sum of active and passive forces g describes all the forces acting on the system, motion of which describes the equation:

$$\mathbf{M}\dot{\mathbf{h}} = \mathbf{g} + \mathbf{g}^{\Phi}, \tag{4}$$

where:
M - the global inertia matrix, containing the masses and mass moments of inertia of all bodies,
$\dot{\mathbf{h}}$ - a vector of accelerations.

The g symbol represents the generalized forces vector.

For multi-segment system with constraints, consisting of n solids, equations of motion of a single solid may be repeated n times, to find the equation of the whole system.

This leads to the equations of motion for the multi-segment system with constraints, written in the form:

$$\mathbf{M}\dot{\mathbf{h}} - \mathbf{B}^T\mathbf{\lambda} = \mathbf{g}. \tag{5}$$

It should be noted, that equation (5) represents the system of *n* second order differential equations of *n*+*m* unknowns, corresponding to the $\dot{\mathbf{h}}$ acceleration vector and the vector of Lagrange multipliers.

To obtain a solution for these equations, the m additional equations are required.

Ideally, if the additional equations were obtained from the constraints equations (1) so as to ensure compliance with both the movement equations and kinematic constraints.

However, in doing so, the solution of system of n+m differential-algebraic equations is difficult to achieve.

Instead the second derivative from the constraints equations, the constraints equation at the acceleration level is the most frequently used:

$$\ddot{\mathbf{\Phi}} = \mathbf{0} \quad \equiv \quad \mathbf{B}\dot{\mathbf{h}} = \mathbf{\gamma}^*. \tag{6}$$

Equations (6) shall be attached to the equations of motion, resulting equation for the system with the constraints in the matrix form take the form:

$$\begin{bmatrix} \mathbf{M} & \mathbf{B}^T \\ \mathbf{B} & \mathbf{0} \end{bmatrix} \begin{bmatrix} \dot{\mathbf{h}} \\ -\mathbf{\lambda} \end{bmatrix} = \begin{bmatrix} \mathbf{g} \\ \mathbf{\gamma}^* \end{bmatrix}, \tag{7}$$

where: B is a modified Jacobian matrix defined in the form of equation (8):

$$\mathbf{B} = \left[\mathbf{\Phi}_{\mathbf{r}_1}; \tfrac{1}{2}\mathbf{\Phi}_{\mathbf{p}_1}\mathbf{L}_1^T; \cdots; \mathbf{\Phi}_{\mathbf{r}_{nb}}; \tfrac{1}{2}\mathbf{\Phi}_{\mathbf{p}_{nb}}\mathbf{L}_{nb}^T\right], \tag{8}$$

and $\mathbf{\gamma}^*$ from equation (7), is a modified right side of the acceleration equations.

The total number of equations is now equal to the total number of unknowns, corresponding to the accelerations and Lagrange multipliers.

2. Modelling the system

Tracked vehicle dynamics analysis was performed in LMS Virtual.Lab Motion software, in which a model of a tank on the PT-91 chassis was developed.

In the modelling process was assumed, that the vehicle was fully equipped with all operating systems fluid-filled, with training ammunition and replacement mass (100 kg per crew member).

Whole vehicle mass - 46220 [kg], including the track weight - 2400 [kg].

Centre of gravity was determined in a coordinate system, shown in Fig. 2.

The coordinate system origin lies in the axis of the drive wheel, X-axis lies in the longitudinal plane parallel to the ground (level), Z-axis sets the vertical direction of the vehicle, and the Y-axis completes the coordinate system clockwise.

The coordinates of the centre of gravity are summarized in Tab. 1.

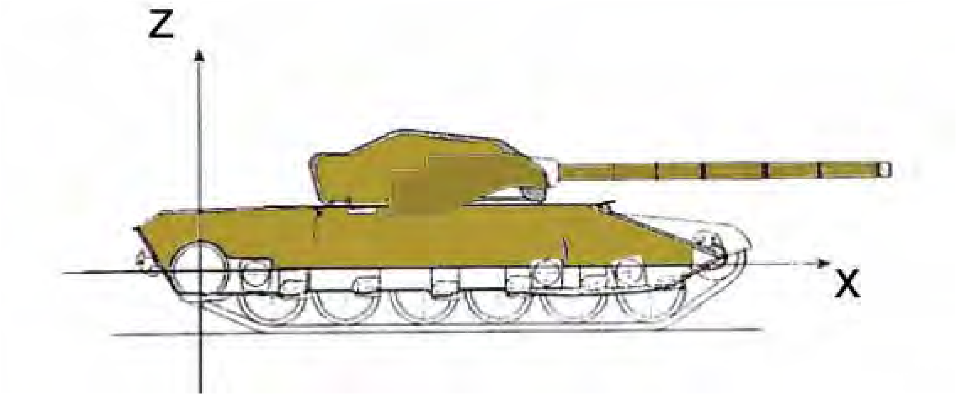

Fig. 2. Location of the coordinate system

Tab. 1. Placement of the vehicle centre of gravity

Xc [mm]	Yc [mm]	Zc [mm]
2654	5	270

Based on the identified mass-geometric parameters, a model of tracked vehicle was developed.

The software environment Virtual.Lab used in modelling process allows kinematic and dynamic simulations of mechanical devices at the design and construction stage process before making a prototype.

The model tank was divided into elements that do not change their positions relative to the assumed coordinate system in the vehicle movement (turret, muzzle, armour or dampers bodies), and the elements that change position relative to the body during motion (wheels, torsion shafts, damper levers, links).

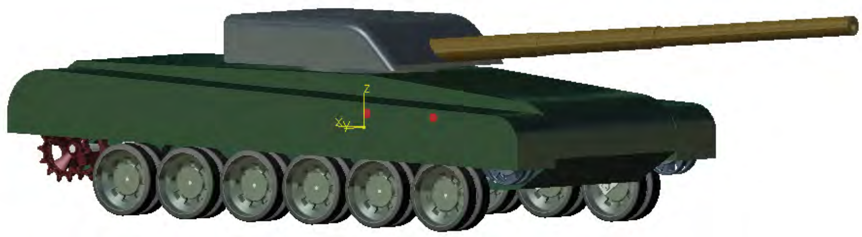

Fig. 3. Assemblage of all CAD model elements of the tank including suspension

Torsion shafts that are linked with the holes in the tank corps were modelled as rotational kinematic pair.

The constraints occurring between the individual elements of the suspension system and the characteristics of torsion shafts, modelled as elastic-damping elements, their torsional stiffness determined on the basis of experimental investigations and estimated material damping were taken into account.

Figure 4 shows the method of affixing the torsion shaft in the vehicle and the graphical interpretation of the imposed constraints.

Figure 5 shows the characteristics of torsion shaft reaction force as a function of shaft deflection in the direction normal to the ground surface.

These characteristics were obtained for the torsional stiffness equal 17 kNm/rad, and the application of the variable force, in accordance with the ***z*** direction of the axis of the global coordinate system.

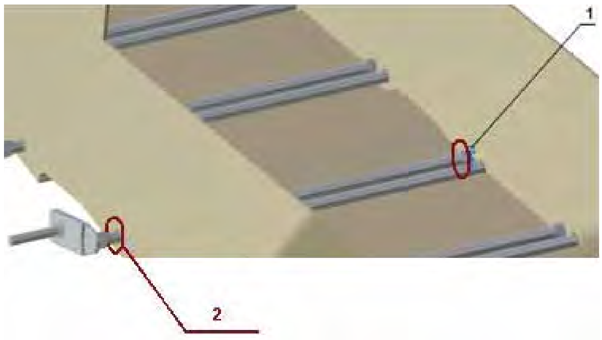

Fig. 4. Schematics of torsion shaft mounting; 1 - torsion elastic-damping element (RSDA), 2 – the kinematic rotation pair

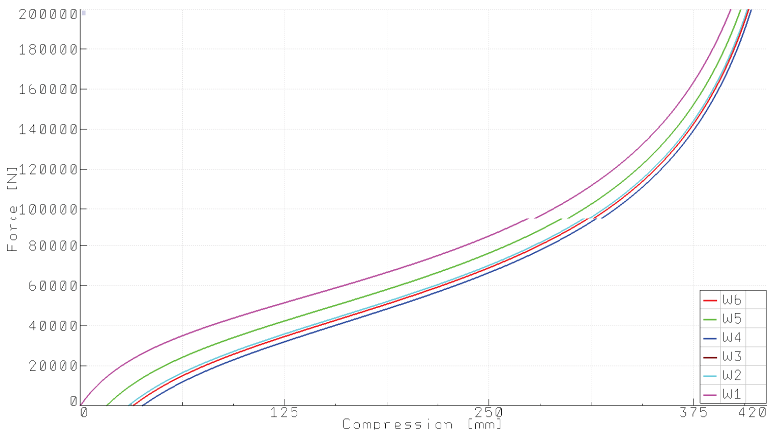

Fig. 5. Spring characteristic of torsion shafts, w1 - w6 torsion shafts numbers

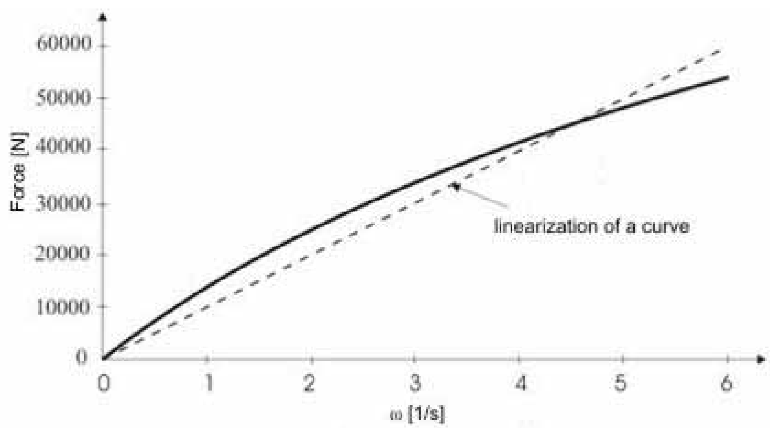

Fig. 6. Shock absorber damping characteristics

Torsion dampers levers were linked with their cases, as in the case of shafts, with rotational kinematic pair, and imposed on them the elastic-damping torsion elements, therewith, they were given a damping property only.

Entered value of damping is equal to 2662 m2kg rad/s, which was determined from the linearized torsion damper characteristics obtained from experimental studies (Fig. 6.).

Figure 7 shows the rocking arm movement transfer system on damper crank movement.

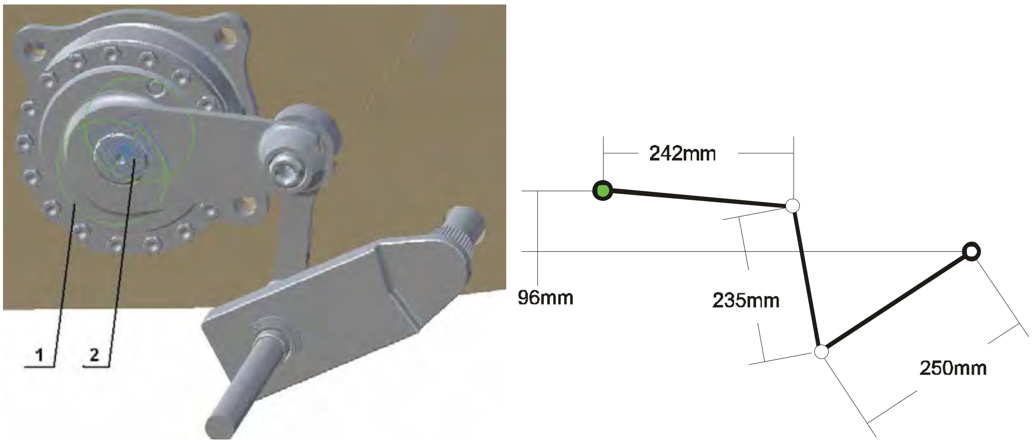

Fig. 7. Arrangement of rods transferring rocking arm movement to the shock damper rocker, 1 - articulation of Revolute Joint type, 2 - elastic-damping torsion element (RSDA)

The developed model takes into account the driving wheels, while the tracks are omitted.

For each wheel the shape of the ground surface was defined, surface after which it moves.

Six wheels on the left side of the vehicle is moving on a plane, while the right side overcomes obstacle with a height of about 30 cm, what corresponds in a proving ground conditions to tank right tread transiting the obstacle.

This required modelling of six trajectories (different for each wheel) running according to the results obtained in the process of identification in proving ground conditions.

All trajectories are generated using the Road program module included in the software.

Driving wheels were modelled by the Simple Tire module.

Model of the tank which overcomes an obstacle is shown in Fig. 8.

Fig. 8. Simulation of tank passing through the obstacle

3. Model verification

In experimental studies, two characteristic points were introduced on the body of the tank, which served to designation of the vehicle body kinematic parameters while overcoming obstacles.

Also in the computer simulation model the points P1 and P2 has been introduced, so you can compare the results obtained from the simulation to results obtained in investigations on the training ground.

Figure 9 shows the deployment of the previously mentioned points and another points related to the wheels axes.

a)

b)

Fig. 9. Distribution of points on the body and wheels axles: a) model, b) the real object

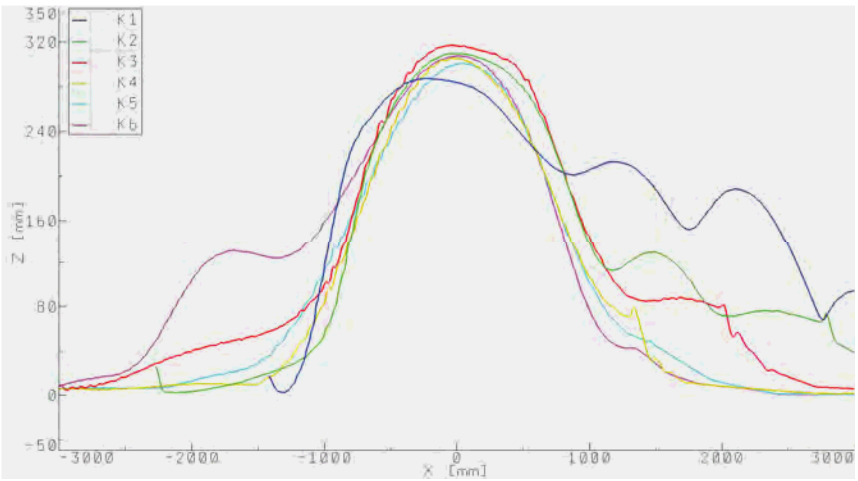

Fig. 10. Movements of all tank wheels (vehicle right side) relative to the vertical axis in the global coordinate system

Below are shown both the results of numerical simulations, and experimental investigations on the proving ground.

The courses were recorded for a vehicle speed equal to 4.66 km/h for the case of overcoming the obstacle by the right tread.

a)

b)

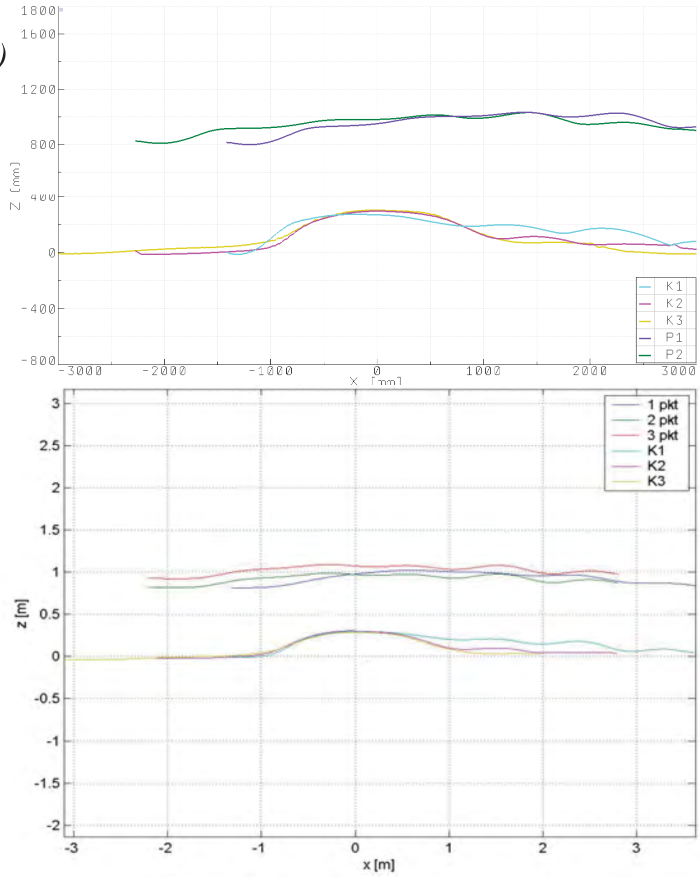

Fig. 11. Displacements of points in the x-z movement plane of the global coordinate system: a) measured on the real object, b) obtained by simulation

a)

b)

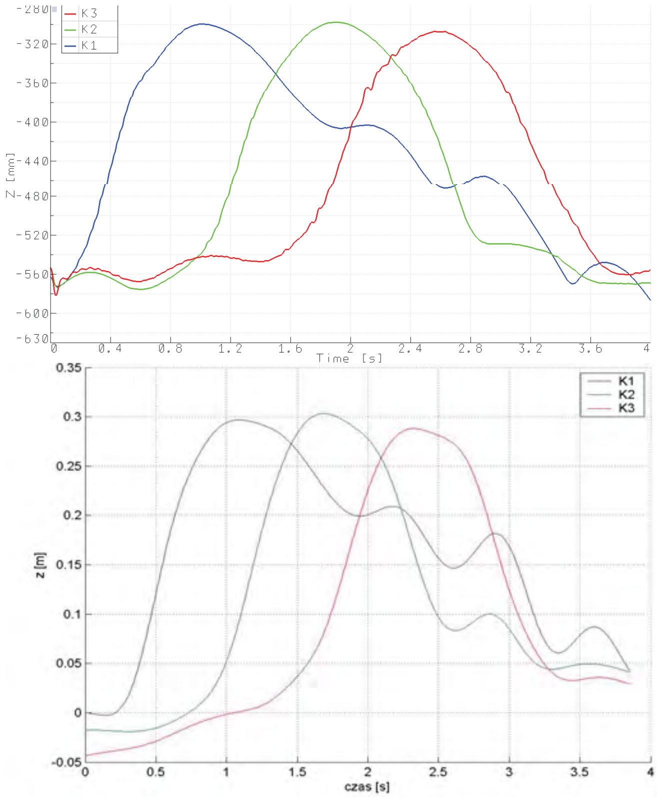

Fig. 12. Movements of tank wheels about a vertical axis in the global coordinate system: a) measured on the real object, b) obtained by simulation

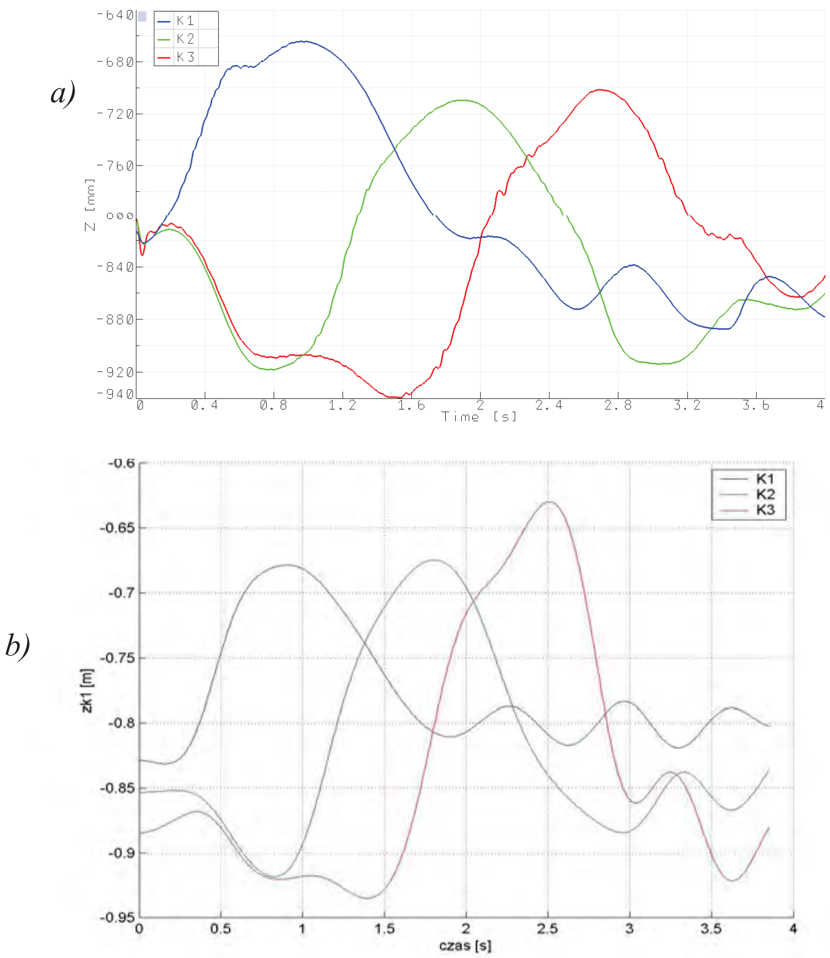

Fig. 13. Movements of the wheels centres about a vertical axis in the coordinate system associated with the tank body: a) measured on the real object, b) obtained by simulation

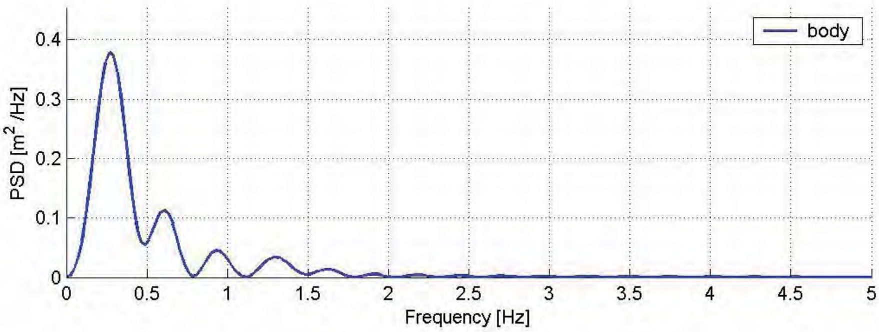

Fig. 14. The power spectral density of the tank body centre of mass displacement signal, measured on the real object

4. Conclusion

The problem of the tracked vehicles dynamics modelling is a complex issue, due to difficulties in the identification of system models.

The basic difficulty is the modelling of track system dynamics.

The authors skipped model of the track in their calculations, reducing its mass to the vehicle body.

The results of measurements on a proving ground were used to tune the vehicle mathematical model.

The results of numerical simulations confirmed the validity of assumptions used in modelling.

The good accordance between recorded and simulated kinematic parameters of the vehicle body and the wheels was received.

The developed model will be used to optimize stiffness and damping characteristics of vehicle suspension.

The studies were performed under the development project R00-O0023/3 funded in years 2007-2010 from MNiSW funds.

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