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Modelling and Investigation of Dynamic Parameters of Tracked Vehicles

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The paper deals with some results of the research dedicated to mechatronic design and machinery dynamics of special vehicles conducted in the Department of Applied Mechanics. The selected important aspects of the tracked vehicles dynamic modelling process related to dynamics of the drive system and the suspension of selected tracked vehicles are presented.

Keywords: Tracked vehicle, magnetorheological damper, shock absorber

1. Introduction

The research and development process dedicated to new generations of military special vehicles continuously tends to employ more and more advanced technologies, including methods of modelling and numerical simulations. These methods become indispensable for the mechatronic engineering procedures, in particular during early phases of the designing practice. The important aspect of the design process in the mechatronic engineering is the use of the synergy effect of research methods in order to achieve the best possible geometrical features and to accomplish the required operational parameters. The example of such an approach is the engineering and design process for a military vehicles aided by results of numeric computations with respect to modelling, analysis of sensitivity and optimization of dynamic properties demonstrated by various systems.

When the initial assumptions for a new design are developed together with the required engineering documentation for a prototype it is necessary to define the target structure of the system and its geometrical attributes. It is the basis that enables to carry out identification of the assumed model and to estimate its parameters. The further phases of the efforts include also optimization of the design with the aim to achieve the desired dynamical properties.

For several years the Department of Applied Mechanics of the Silesian University of Technology has been dealing with the research programs dedicated to mechatronic design and machinery dynamics to various applications, including also special vehicles. In the paper will be presented only some important aspects of the tracked vehicles dynamic modelling process related to dynamics of the drive system and the suspension of selected tracked vehicles.

2. Modelling of a hybrid drive for a tracked vehicle

In practice, the issue of the crucial importance are dynamic loads that result from movements of the vehicle and exert a continuous impact to the vehicle body and members of the squad. It is why intense investigations are carried out with the aim to alleviate the adverse impact of such phenomena. However, the optimum selection of parameters affecting the operation quality of the driveline and suspension needs to develop a dynamic model of a vehicle.

The design of presented tracked platform with a hybrid driving system has been provided by the scientific and industrial consortium that included the Silesian University of Technology, WASKO SA., the AGH University of Technology and the CPW HSW in Stalowa Wola.

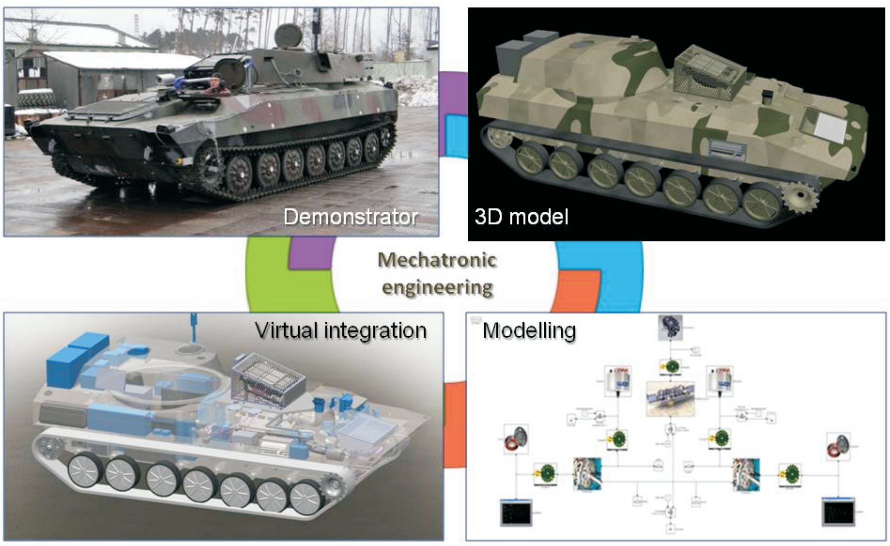

Figure 1 The circle of the mechatronic engineering process of the tracked vehicle with a hybrid driving system

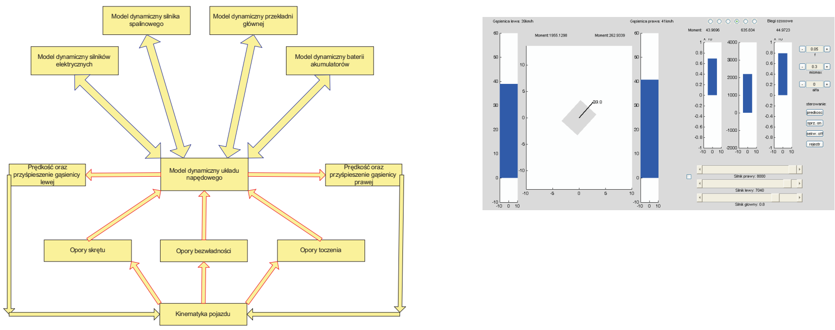

Figure 2 The structure of the dynamic model for the vehicle and the graphic Human–Machine Interface (HMI) with illustration of kinematic parameters for the demonstrative model of the vehicle with a hybrid driving system

The dynamic model of the driving system for the demonstrative model of a tracked vehicle has been developed and implemented with the MATLAB/SIMULINK software environment. This model comprises some modules, such as the model of an internal combustion engine, the models of electric motors, rechargeable batteries and the main transmission gear. The completed simulation experiments enabled to determine traction characteristics of the hybrid driving system and the tracked vehicle as a whole. The graphic interface of the newly developed model that is shown in Fig. 2 enables easy adjustment of dynamic parameters associated with individual components of the driving system.

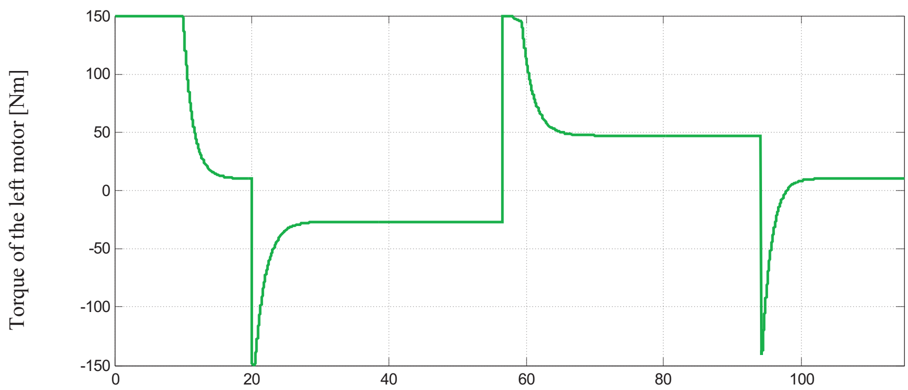

Figure 3 Torque of the left motor

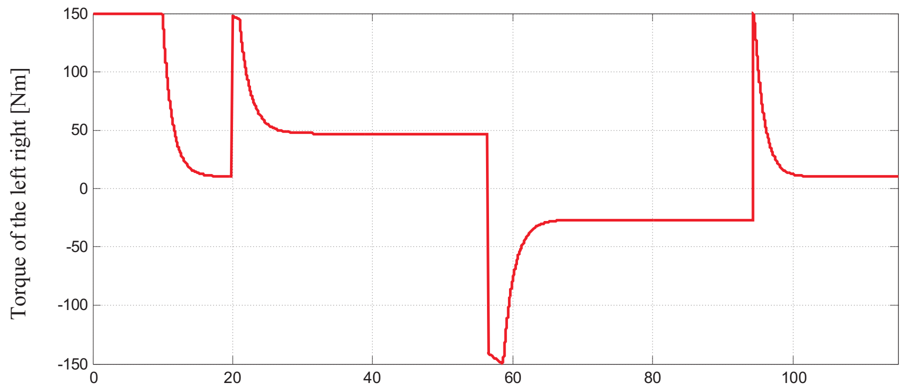

Figure 4 Torque of the right motor

The results of simulation tests provide the evidences that the elaborated hybrid driving system demonstrates high efficiency under real operating conditions and enable the vehicle to turn to infinitely continuous angles at a high speed and the power is recuperated from the inner track. The recovered power is transferred to the outer side and enables mechanical recuperation of power.

3. Suspension systems of special vehicles

The efficiency of suspension systems is the factor that limits increase of the maximum speed available to combat vehicles. In case of military vehicles it is necessary to ensure suspension systems that are capable to adapt to diverse road conditions (a motorway, off–road area, military training grounds, etc.). It is why modern semi–active suspension systems enable carry on with development of innovative military vehicles that feature with high travel speed and increased mobility owing to new generation of machinery and equipment.

The suspension system with a magnetorheological damper is an example of a semi–active suspension system of the mechatronical type. The dynamic model of the system with two degrees of freedom is shown in Fig. 5, where the model defines dynamical features of the vehicle (so called model of a vehicle quarter) and takes account for controlled damping of oscillations by means of the implemented dynamic model for a prototype magnetorheological (MR) damper.The completed computer simulations and experiments on the test workbench (Fig. 6) made it possible to develop a prototype MR damper and to find out its dynamical characteristic curve. The obtained waveforms for the damping force have been determined as functions of the piston rod velocity and the electric current that flows via the solenoid of the MR damper. The identification experiment was carried out with use of the mechanical test machine from MTS (Fig. 7).

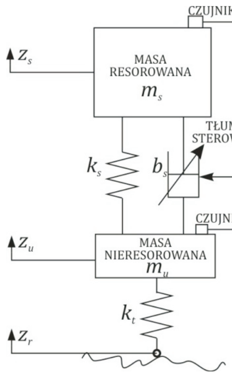

Figure 5 The model of a semi–active suspension system with a MR damper and a control system

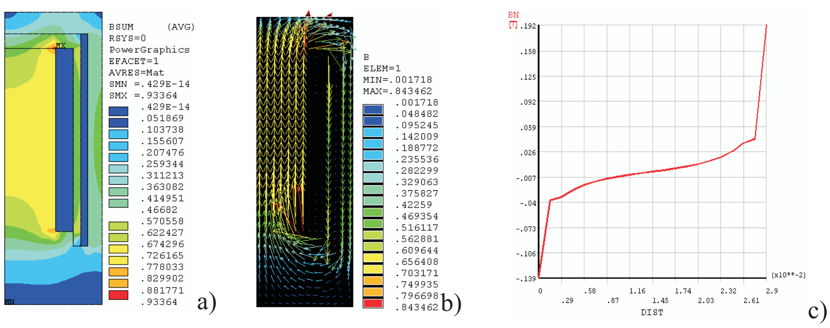

Figure 6 Results from numerical computations for the magnetic contour of the MR damper : a) map of magnetic induction, b) distribution of induction vectors, c) magnetic flux as the function of the gap width (by T. Machoczek)

The program of tests included determination of dynamic characteristic curves for various values of the electric current in the solenoid and for various velocities of the piston rod. The obtained characteristic curves for selected parameters from the adopted intervals are shown in Fig. 8 and 9.

The recorded experimental characteristic curves that reflect the dynamic properties of the prototype MR damper were used to develop the dynamic model of the damper with consideration to its geometrical shape (working stroke of the piston rod), where the Simulink software package was used as the modelling tool. The completed numerical simulations made it possible to draw up waveforms for displacements, velocities, absolute and relative acceleration values of both the cushioned and non–cushioned weight. The input function was assumed as harmonic oscillations. The selected simulation results are shown in the drawings below.

Figure 7 Test workbench for determination of dynamic characteristics parameters demonstrated by the prototype MR damper

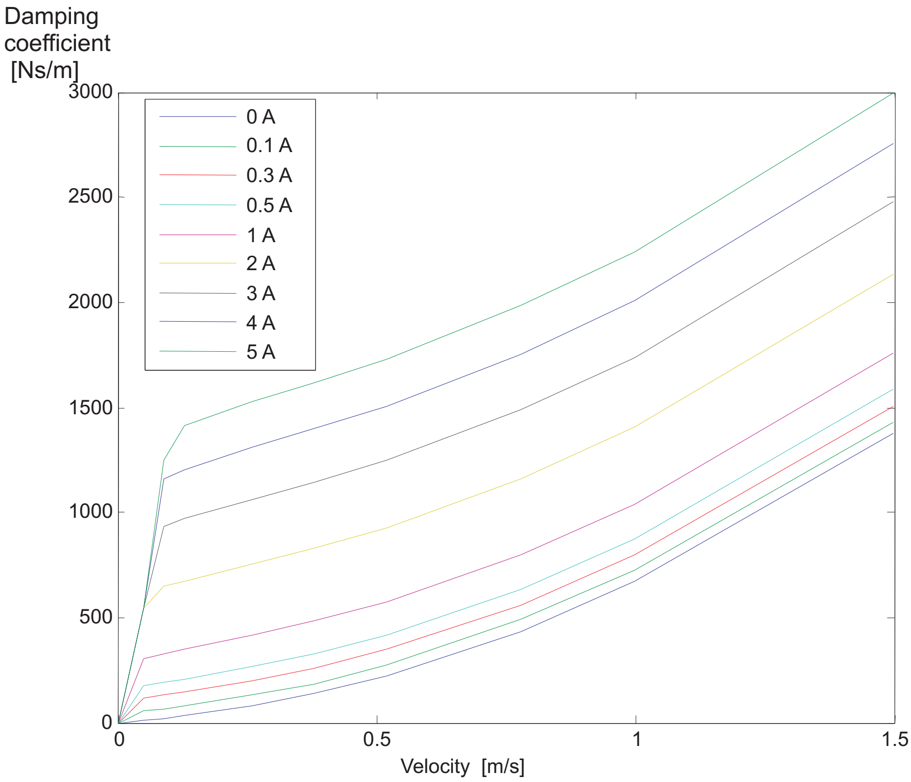

Figure 8 The family of characteristic curves for the damping force as the function of the piston rod velocity and for various values of electric current in the solenoid of the prototype MR damper

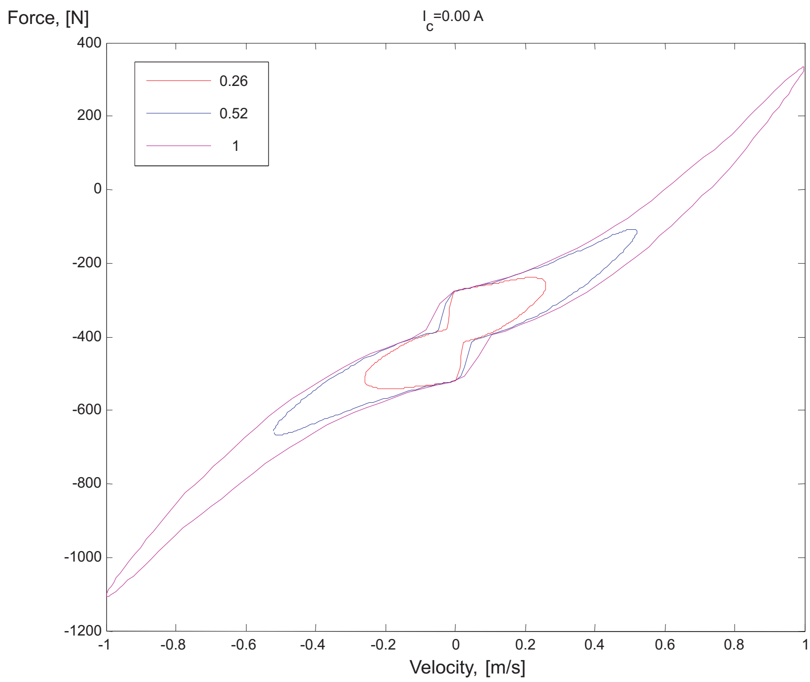

Figure 9 The family of characteristic curves for the damping force as the function of three velocities of the piston rod for the value of electric current in the solenoid equal to 0 A

The Fig. 10 and 11 show how displacements, velocity and acceleration of the cushioned weight differ from one another depending on whether the control system for the magnetorheological (MR) damper is on or off. The resonance frequency is the same for the both cases and equals to 1.25 [Hz].

Simulation of movements performed by tracked vehicles need to develop much more extensive and sophisticated models that take account for the lack of symmetry against the crosswise axis of the vehicle or, under certain conditions, also the lack of symmetry against the longitudinal axis of the vehicle (e.g. rotation of the turret, reconfiguration of the attached equipment, etc.).

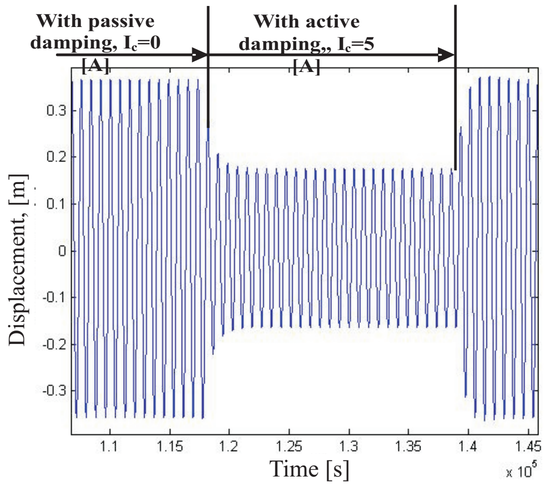

Figure 10 The waveform for absolute displacement of the cushioned weight

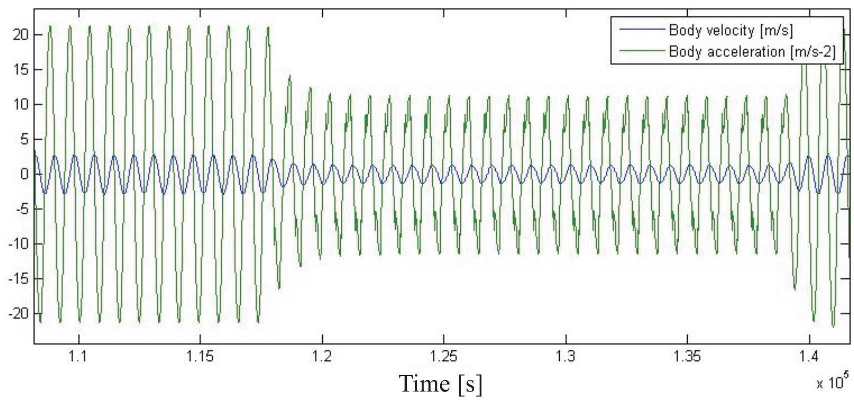

Figure 11 The waveform for absolute velocity and acceleration values for the cushioned weight

4. Modelling of the tracked vehicle suspension system

Suspension systems of tracked military vehicles represent the class of mechanisms that require description of movements exercised by their components with the account that the displacements are sometimes really sizeable and with consideration to imposed constrains and sophisticated interactions with the ambient environment. Such a multibody system can be defined as an arrangement of rigid or flexible bodies, mutually interconnected by kinematic pairs or components that are sources of mechanical forces. The system of equations obtained from the kinematic constrains and driving components can be combined in a consistent manner into a global vector of constraints denoted as Φ and expressed in the form of the following equation:

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

where: $\mathbf{q}$ is a vector of generalized coordinates and t stands for time.

Within the system of bodies subjected to constrains of movements, the bodies are interconnected by means of internal kinematic forces. The reaction forces that act in constraints and are also rated to the forces that constrain movements, are denoted by the $\mathbf{g}^{\Phi}$ vector whilst the sum of active and passive forces, denoted as $\mathbf{g}$, describes all forces that act onto the system and its movements is described by the equation:

$$\mathbf{M}\,\dot{\mathbf{h}}=\mathbf{g}+\mathbf{g}^{\Phi} \tag{2}$$

where: $\mathbf{M}$ is a global matrix of inertia that comprises masses and mass–related moments of inertia for all the bodies within the system, whilst $\dot{\mathbf{h}}$ stands for the vector of accelerations. The $\mathbf{g}$ represents the vector of generalized forces.

For a constrained multimodular system that is made up of n bodies, the equations of motion for a single body can be repeated n times in order to find out equations for the entire system. It leads to equations of motion for a constrained system, where the equations are written in the form:

$$\mathbf{M}\,\dot{\mathbf{h}}-\mathbf{B}^{T}\lambda=\mathbf{g} \tag{3}$$

The equation (3) represents a system of n differential equations with $n+m$unknown variables that correspond to the acceleration vector of $\dot{\mathbf{h}}$ and the Lagrange vector of multipliers. To resolve the system of these equations it is necessary to formulate m additional equations. Preferably, these additional equations should be obtained from the equations for constraints (1) so that to guarantee that both the equations of motion as well as kinematic constraints shall be fulfilled. However, in case of such a procedure, it is really difficult to find a solution for a system of $n+m$ differential and algebraic equations. It is why the second derivatives of the equations for constraints are substitutes with equations for constraints on the level of acceleration:

$$\ddot{\mathbf{\Phi}}=\mathbf{0}\quad\equiv\quad\mathbf{B}\dot{\mathbf{h}}=\gamma^{*} \tag{4}$$

The equations (4) are appended to the equations of motions, thus the equations for the constrained system can be written in the matrix form:

$$\begin{bmatrix}\mathbf{M} & \mathbf{B}^{T}\\ \mathbf{B} & \mathbf{0}\end{bmatrix}\begin{bmatrix}\dot{\mathbf{h}}\\ -\lambda\end{bmatrix}=\begin{bmatrix}\mathbf{g}\\ \gamma^{*}\end{bmatrix} \tag{5}$$

where: $\mathbf{B}$ is the modified matrix of a Jacobian determinant that is expressed by the equation:

$$\mathbf{B}=\left[\mathbf{\Phi}_{\mathbf{r}_1};\tfrac{1}{2}\mathbf{\Phi}_{\mathbf{p}_1}\mathbf{L}_1^{T};\cdots;\mathbf{\Phi}_{\mathbf{r}_{nb}};\tfrac{1}{2}\mathbf{\Phi}_{\mathbf{p}_{nb}}\mathbf{L}_{nb}^{T}\right], \tag{6}$$

whilst γ^{*} can be calculated from the equation (6), which is a modified right–hand side of the equation of acceleration. Now, the total number of equations is the same as the total number of unknown variables that correspond to values of acceleration and Lagrange multipliers.

Nowadays, the modelling phase of the design process usually involves modern tools to create virtual prototypes of machines or vehicles. It is possible to develop both 2D models with concentrated parameters or 3D models that can be resolved with use of dedicated software tools, such as ADAMS or LMS Virtual Lab Motion.

The entire process of simulation studies and reliability of results obtained from numerical computations depend on accurate identification of dynamic models. The key issue is estimation of parameters attributable to models, in particular characteristics of energy dissipation by shock absorbers (dampers) installed in suspension systems. Characteristic parameters of shock absorbers can be acquired by means of experimental measurements or by numerical simulation of phenomena that take place inside the devices. Such simulations are carried out by means of the Finite Elements Method (FEM), where, in case of a hydraulic shock absorber, the liquid chamber is subjected to discretization. The investigations of discrete models provide general operational characteristics of shock absorbers, which makes it possible to select their operational parameters in a best possible way. Fig. 12 shows an example of such a numerical simulation for flow of liquid through a valve of a linear damper. Such studies are time–consuming and need application of dedicate software, but they enable to optimize the design of a shock absorber (damper) and to adapt its characteristic parameters to the dynamic behaviour of the vehicle suspension.

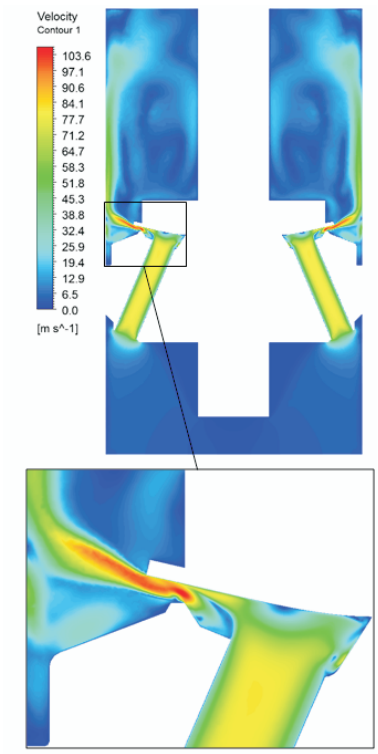

Figure 12 The FEM analysis for flow of liquid via a valve of a linear damper (by J. Gniłka)

5. Simulation and experimental testing of of the tank suspension system

Investigations of dynamic parameters demonstrated by a tracked vehicle were carried with use of the LMS Virtual Lab Motion software and covered a range of vehicles, including the PT–91 tank. The suspension system of the PT–91 tank is made up of 12 torsion shafts combined with rockers as well as 6 hydraulic shock absorbers (dampers) of the vane type disposed at the first, second and sixth row of driving wheels, on the both sides of the tank. The modelling process assumed that the vehicle is fully loaded and the total weight of the vehicle is 46, 220 [kg], of that the track chain weight is 2,400 [kg].

The Virtual Lab software environment that was used for the modelling process enables both kinematic and static simulation of mechanical systems as early as during the design phase, before a physical prototype is made. Fig. 13 shows superposition of components included into the developed model of the tank.

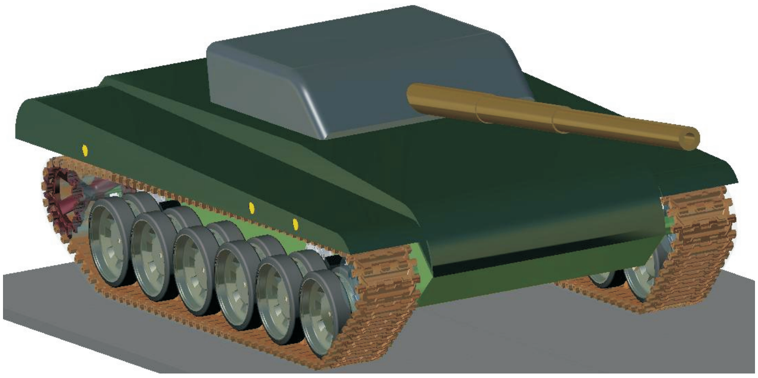

Figure 13 Superposition of all the CAD models for tank components with the suspension system

The modelling process took into account constraints that occur between individual components of the suspension system as well as for characteristic parameters of torsion bars that have been modelled as spring (deformable) and damping components with the torsion rigidity already determined on the basis of experimental measurements and theoretically estimated coefficient of material damping. Fig. 14 shows the graph of the characteristic curve of the spring force demonstrated by the torsion shaft as a function of compression measured down the direction normal (perpendicular) to the vehicle travel.

The applied damping factor was calculated from the linearized characteristic curve for torsional damping, obtained from experimental investigations. The suspension system for a traction wheel together with a rotational damper makes up a sophisticated kinematic scheme with non–linear characteristics of rigidity and damping efficiency. The model of a tracked vehicle comprises traction wheels

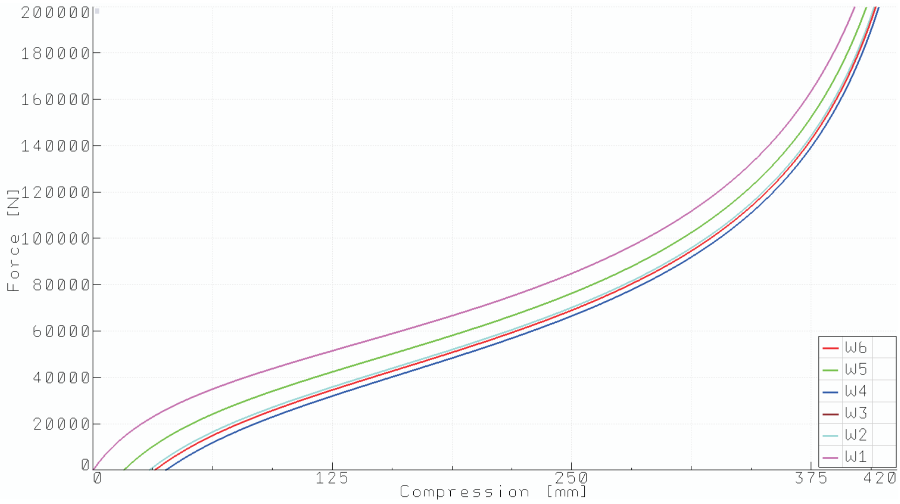

Figure 14 The characteristic curve for the spring force of torsion shafts, the letters w1 – w6 correspond to numbers of torsion shafts

and tracks as well. The scope of experiments assumed that for each wheel the substrate profile shall be defined and the wheel shall travel thereon. For the simulation it was adopted that six traction wheels on the left–hand side of the tank travel on a flat plane whilst the right–hand side encounters an obstacle with the height of about 30 cm, which corresponds to the circumstances when the right–hand track of the tank has to run over an obstacle under field conditions.

The model of the tank that runs over the obstacle is shown in Fig. 15.

Figure 15 Simulation on travelling over an obstacle

One of the simplest techniques that enables to find out kinematic parameters for selected components of the vehicle consists is the photogrametric approach. The method assumes that the kinematic variables are measured with use of digital recording CCTV cameras, where the motion of the examined facilities is initially recorded and then the images are subjected to the digital frame–by–frame analysis.

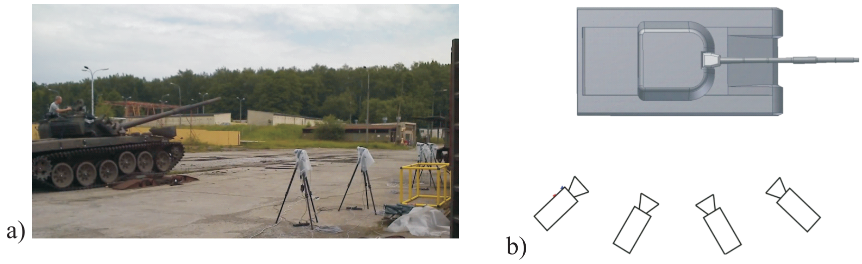

Figure 16 Deployment of CCTV cameras down the path of measurements

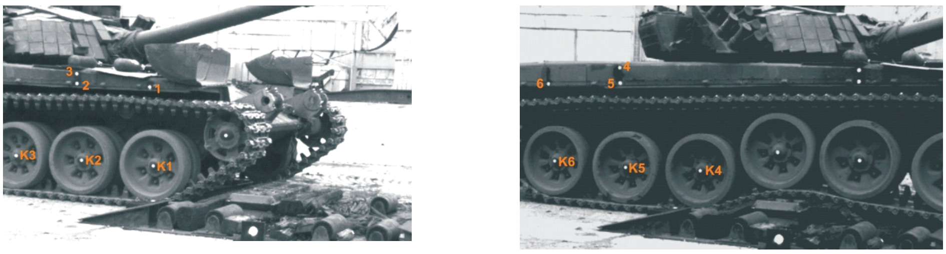

Figure 17 Deployment of markers on the tank

The analysis of recorded images makes it possible to determine coordinates of the markers already deployed on the tank. The coordinates, in turn, enable calculation of absolute and relative displacements, velocities and accelerations for selected components of the vehicle. Owing to the MATLAB software environment it is possible to find out the desired kinematic parameters, to plot graphs and carry on with further processing and analysis of the obtained results. The Fig. 18 presents results from both numerical simulations and field tests.

Another way that allows verifying assumptions adopted for virtual modelling of dynamic behaviour demonstrated by a tracked vehicle consists in measurement of acceleration values for specific points on the tank body. The measurements are taken with use of a multi–channel recorder, e.g. LMS SCADAS III and an advanced system for measurements of vibrations.

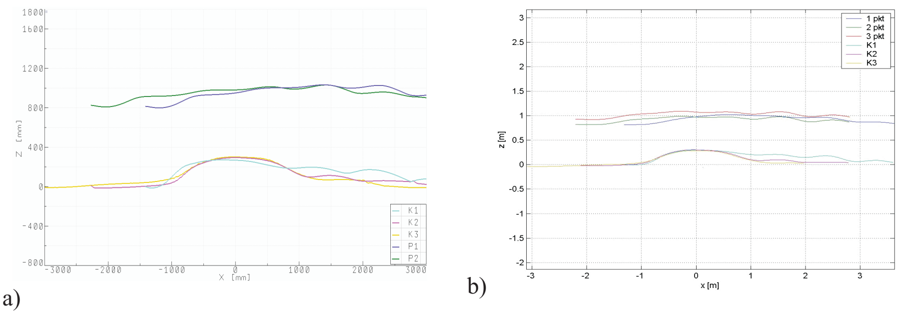

Figure 18 Displacements of the selected points within the XZ plane of the global coordinate system: a) measured for a real vehicle; b) obtained from the numerical simulation

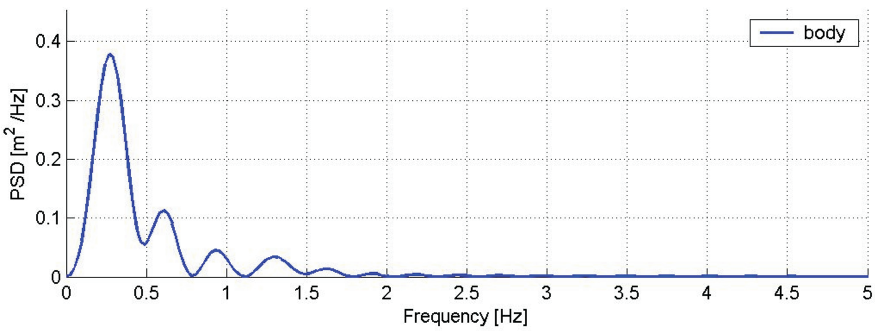

Figure 19 Spectral density of the signal power recorded for displacement of the tank hull gravity centre and measured for a real object

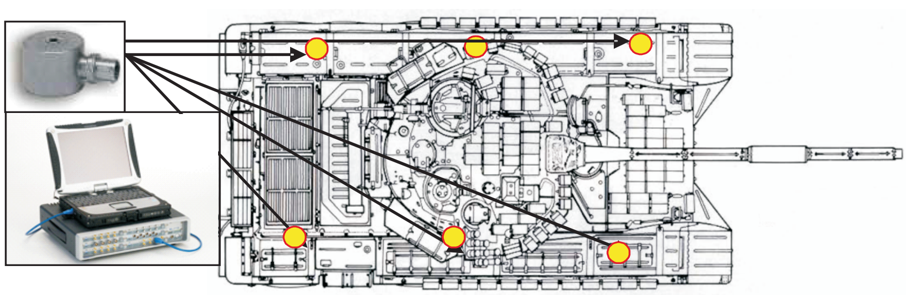

Figure 20 Deployment of acceleration sensors on the vehicle hull

The system enables accurate and repeatable measurements of selected signals while the tests are carried out under field conditions (Fig. 21). Another advantage of the system is high speed of data acquisition. The method makes it possible to record acceleration values when the vehicle travels over terrain obstacles, which serves as the basis for verification whether the results of virtual modeling are correct as the values of forces that act onto the tank hull can be measured and compared.

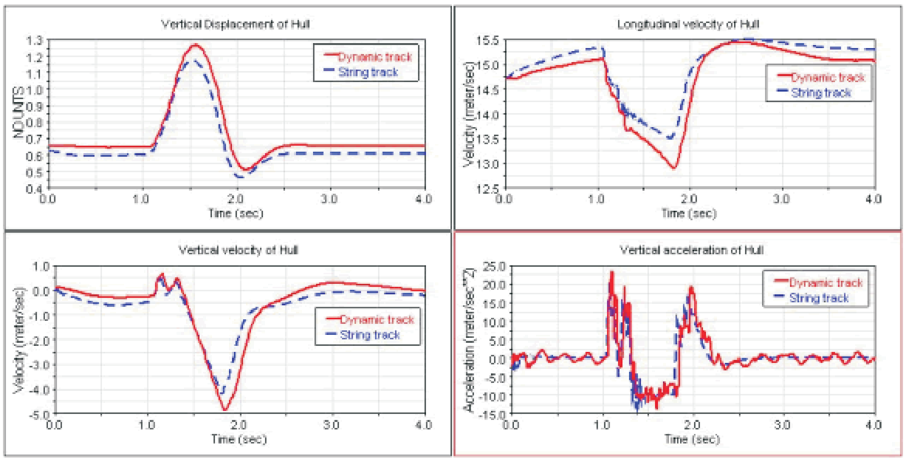

Figure 21 Examples of results for the values of displacement, longitudinal and vertical velocities and acceleration for the investigated model of the tank hull

6. Conclusions

Creation of numerical models, designing and optimization of subassemblies and individual units proved possible owing to modern methods of computer engineering and sophisticated software. However, the complexity degree of the problem that the engineering practice has to deal with as well as the need to apply advances computation techniques imposes the necessity of very close collaboration between the scientific centres and industrial companies, both in the area of fundamental research studies and in implementation of new technologies to the manufacturing practice. The presented paper is just a result of such close collaboration between the Department of Applied Mechanics and companies of the military industry as well as civil and military universities.

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