ANALYSIS OF THE VEHICLE UNDERCARRIAGE

Eng. Neumann V. Ph.D.
Faculty of military technologies –University of Defence, the Czech Republic

vlastimil.neumann@unob.cz

Abstract: *Parameters of the vehicle undercarriage affect the vehicle movement and its behaviour on the road or on the terrain. Basic geometry of the vehicle undercarriage can be expressed by mathematical methods, yet undercarriage analysis of the moving vehicle on the terrain is very complex procedure. We have to take into account wheels vertical movement and deflection (which influences the suspension system). Utilization of the simulating technologies can be very beneficial for solution of this issue. The paper is focus on mathematic model of the vehicle undercarriage and its application for analysis of the vehicle undercarriage.*

Keywords: SIMULATIONS, TRUCK, STEERING, SUSPENSION

1. Introduction

Lethality, protection, mobility and communication are the general capabilities of the combat vehicle. Nowadays conflicts and military missions require different types of vehicles. Improving of the lethality and protection is current trend. Both these capabilities affect vehicle weight and its mobility. Finding balance between mobility, protection and fire power can be very difficult process and very often must be chosen a compromise, between fire power and protection on the one side and mobility and weight on the other side. Simulating technologies are suitable manner for analysis of the vehicle mobility and its change during the process of vehicle improvement.

Simulating technologies are very efficient manner in construction of the mechanical systems and nowadays they are utilized in design and development stage of life cycle of vehicles. Mathematic simulations can be used for assessing, evaluating and comparing of basic capabilities amongst vehicles. The main advantages are:

- possibility of comparing different vehicles in the same operational environment (e.g. on the same obstacles)
- simulation of various critical operational states and in-service behaviour of vehicle (e.g. destruction of wheel)
- evaluation of different modifications (modernization) influence on vehicle mobility
- development of capabilities – mobility, survivability and reliability

Simulating technologies have some disadvantages, too. The main key point is input data – either insufficient or lack of them.

The purpose of this paper is to demonstrate utilization of simulating technologies (mathematic model development) for the analysis of the vehicle undercarriage.

2. Description of the mathematic model

For vehicle analysis and mathematic model development we use Multibody Dynamics software ADAMS (Automatic Dynamic Analysis Mechanical Systems) of MSC software company. Adams improves engineering efficiency and reduces product development costs by enabling early system-level design validation.

Adams is optimized for large-scale problems, taking advantage of high performance computing environments. For dynamics vehicle development and testing MSC offers module ADAMS/Car. With Adams/Car engineering teams can build and test functional virtual prototypes of complete vehicles and vehicle subsystems. Working in the Adams vehicle environment, automotive engineering teams can exercise their vehicle designs under various road conditions, performing the same tests they normally run in a test lab or on a test track, but in a fraction of time.

Czech military vehicle T-810 has been selected for application of simulating technologies. The vehicle T-810 is medium off-road truck which meets requirements of high terrain throughput, robustness, endurance, airway, seaway and railway transportability. Picture of this vehicle is shown in the Fig. 1.

Fig. 1 *The medium off-road truck T-810 [7].*

Basic tactical-technical parameters of the truck T-810 are presented in the table 1.

Table 1: *Basic tactical-technical parameters of the vehicle T-810 [3, 9].*

Parameter	Value
Engine	water cooled diesel, in-line 6 cylinders
Engine power	177 kW
Engine torque	920 Nm
Max. speed	106 km/h
Min. speed	3,2 km/h
Tank range	800 km
Max. weight	13 000 kg
Standby weight	8 500 kg
Length	7 490 mm
Width	2 550 mm
Height	3 280 mm
Gauge of the front axle	2 020 mm
Gauge of the rear axles	2 100 mm
Wheelbase	3 150 + 1 200 mm
Step obstacle	600 mm
Ditch obstacle	900 mm
Ride height	460 mm
Wheels	362/80 R20
Front suspension	wound springs
Rear suspension	leaf springs
Steering gear ration	18,1 – 21,4

The objective of the work is to create complex model of the vehicle T-810 which can be used for general simulations (e.g. turning, gear shifting, vehicle manoeuvres). The general model (assembly) has to include minimally next subsystems: front/rear suspension; steering; powertrain; transmission; brakes; front/rear wheel; body. Chassis of the vehicle T-810 is shown in the Fig. 2.

Fig. 2 *Chassis of vehicle T-810 [6].*

Fig. 3 *Front axle of the vehicle T-810 [9].*

In the paper will be presented assembly of the front axle. Created model of front suspension system with the steering system is shown in the Fig. 4. Assembly of the front axle consists of the two subsystems – front suspension subsystem and steering subsystem. Main parts of the suspension subsystem are: tube, leading rods, V rod, propelling shafts, hook joints, steering pins, springs, dumpers, bumpstops and bushings. The steering subsystem consists of: steering wheel, shafts, hook joints, steering box, steering bars and levers. Testing workbenches and tires are the main parts of this assembly, too. This assembly enables analysis of the steering systems in dependency on wheels vertical position or their vibrations. Characteristics of the wheel can be changed as like as other parameters of the main parts.

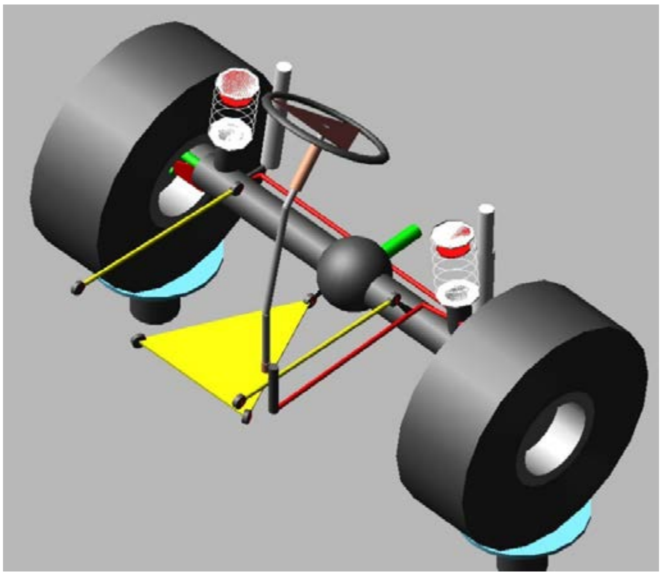

Fig. 4 *Created assembly of the front suspension system with steering.*

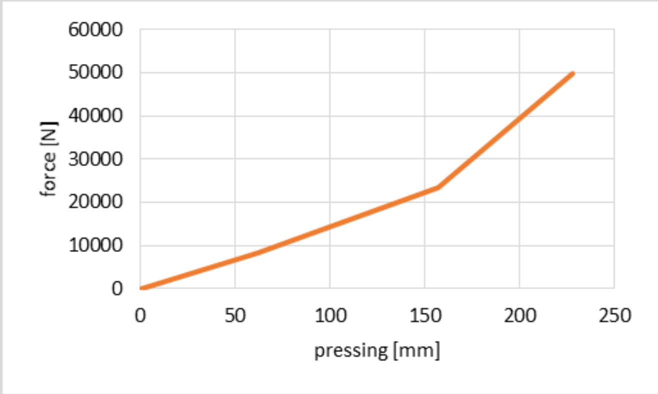

Fig. 5 *Characteristics of the front springs.*

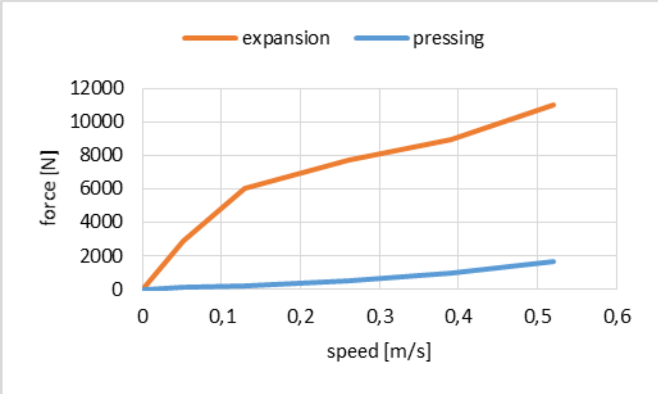

Fig. 6 *Characteristics of the front dumpers.*

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vehicle/test bench
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tyre
ASPECT_RATIO              = 0.8
RIM_RADIUS                = 0.216
RIM_WIDTH                 = 0.355
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DREFF                     = 0.5
FREFF                     = -0.005

FNOMIN                    = 35000
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[LONG_SLIP_RANGE]
KPUMIN                    = -1.5
KPUMAX                    = 1.5
$-------------------------------------
[SLIP_ANGLE_RANGE]
ALPMIN                    = -1.5708
ALPMAX                    = 1.5708
$-------------------------------------
[INCLINATION_ANGLE_RANGE]
CAMMIN                    = -0.26181

CAMMAX                    = 0.26181

$-------------------------------------
[VERTICAL_FORCE_RANGE]
FZMIN                     = 1750

FZMAX                     = 78750
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Fig. 7 *Selected parameters of the model tires – part 1.*

```
longitudinal force
PDX1                        = 0.77751
Fznom
PDX2                        = -0.24431
load
PDX3                        = -0.00015908
camber
PEX1                        = 0.46659
Fznom
PEX2                        = 0.393
load
PEX3                        = 0.076024
load squared
PEX4                        = 2.6509e-006
driving
PKX1                        = 14.848
Kfx/Fz at Fznom
PKX2                        = -9.8161
Kfx/Fz with load
PKX3                        = 0.15818
Kfx/Fz with load
PHX1                        = -0.00088873

PHX2                        = -0.00067818

PUX1                        = -5.5714e-007

PUX2                        = 6.2972e-006
load
RBX1                        = 11.13
Fx reduction
RBX2                        = -12.494
with kappa
RCX1                        = 0.97505
Fx reduction
REX1                        = -0.37196

REX2                        = 0.0017379
with load
RHX1                        = 0.0045181
Fx reduction
PTX1                        = 1.5
Fznom
PTX2                        = 1.4
load
PTX3                        = 1
exponent of load
```

```
[LATERAL_COEFFICIENTS]
PCY1                        = 1.5874
forces
PDY1                        = 0.73957
PDY2                        = -0.075004
load
PDY3                        = -8.0362
squared camber
PEY1                        = 0.37562

PEY2                        = -0.069325
load
PEY3                        = 0.29168
curvature Efy
PEY4                        = 11.559
camber
PKY1                        = -10.289
Kfy/Fznom
PKY2                        = 3.3343
maximum value
PKY3                        = -0.25732
camber
PHY1                        = 0.0056509

PHY2                        = -0.0020257

PHY3                        = -0.038716
camber
PVY1                        = 0.015216
Fznom
PVY2                        = -0.010365
load
PVY3                        = -0.31373
camber
PVY4                        = -0.055766
camber and load
RBY1                        = 13.271
reduction
RBY2                        = 5.2405
with alpha
RBY3                        = 1.1547e-005
reduction
RCY1                        = 1.01
reduction
REY1                        = 0.010513

REY2                        = 5.9816e-005
with load
```

Fig. 8 *Selected parameters of the model tires – part 2.*

3. Simulations

Pasted simulations were focus on wheel swerve in different situations: flat road, raised one wheel and bumpy terrain (profile of the left and right traces are shown in the next graphs – Fig. 9 and Fig. 10). The length of the simulations was 10s. Rotation of the steering wheel was defined by the next graph (Fig. 11).

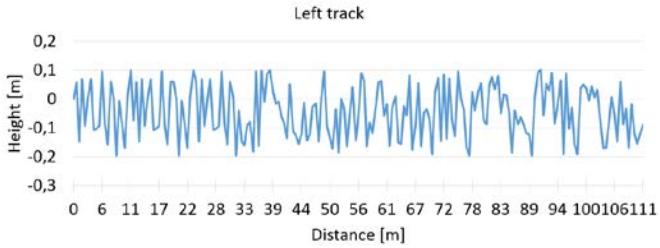

Fig. 9 *Profile of the left wheel track – simulation of bumpy road.*

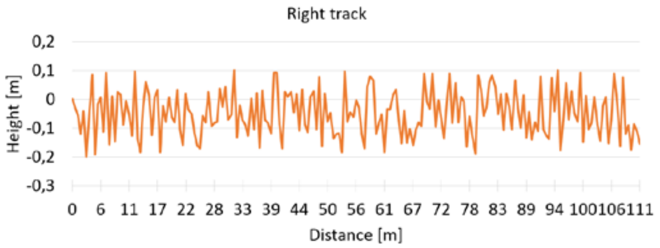

Fig. 10 *Profile of the right wheel track – simulation of bumpy road.*

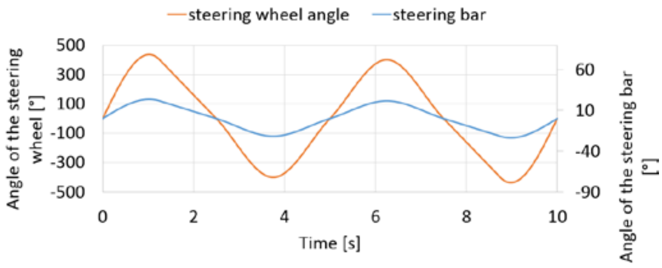

Fig. 11 *Position of the steering wheel and steering bar during the simulations.*

In the Fig. 11 there is also presented rotation of the steering bar. Steering angle of the wheel depends on steering wheel position, geometry of the steering mechanism and vertical position of the wheels. Simulations were focus on the analysis of the change of the steering wheel angle during the vertical wheel movement. The basic simulation was done with no wheels movement – these angles (wheel rotational movements) were reference. 2nd simulation was passed with left wheel up (150 mm) and right wheel down (150 mm). 3rd simulation was inverted left wheel down, right wheel up. 4th simulation was passed with wheels movement defined in the Fig. 9 and Fig. 10 – model speed during the simulation was set up to 40 km/h. Outcomes are presented in the next graphs (Fig. 12 – Fig. 16). In the left side there are the wheel steering angles of the left and right wheel. In the right side there are presented differentials of the each simulation (dif.1 = outcomes of sim.1 – outcomes of sim.2; dif.2 = outcomes of sim.1 – outcomes of sim.3; dif.3 = outcomes of sim.1 – outcomes of sim.4).

Steering force was the next analysed parameter, because vehicle movement produces forces which are transferred thought the steering mechanism and burden the driver. Courses of the steering wheel input torque are presented in the next graphs (Figure 16).

4. Outcomes

Fig. 12 *Wheels angles on the flat road.*

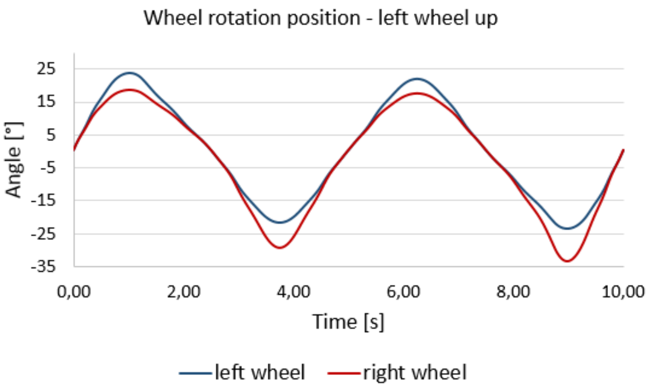

Fig. 13 *Wheels angles on the flat road – left wheel up.*

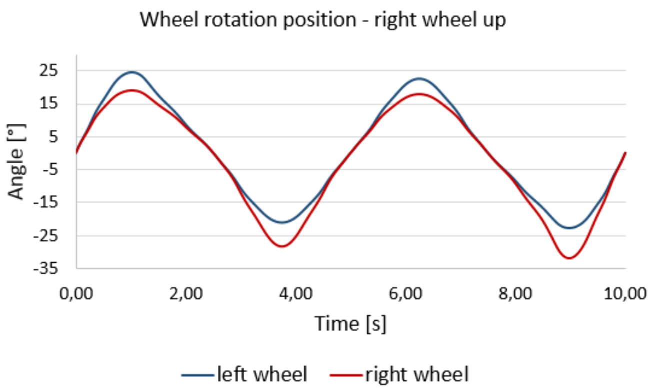

Fig. 14 *Wheels angles on the flat road – right wheel up.*

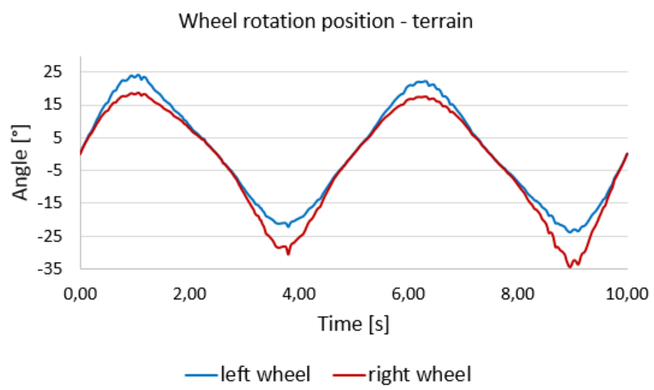

Fig. 13 *Wheels angles on the bumpy terrain.*

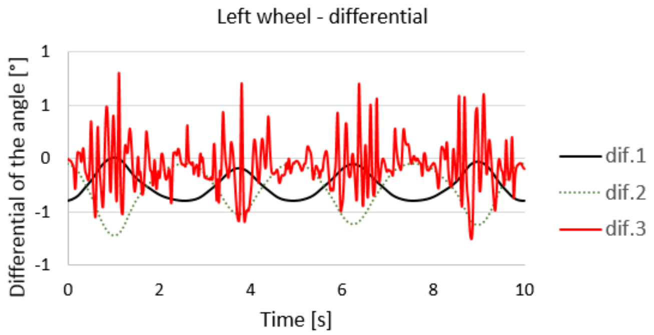

Fig. 14 *Differentials of the left wheel.*

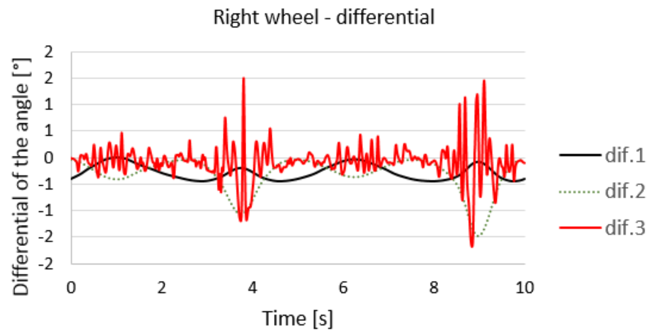

Fig. 15 *Differentials of the right wheel.*

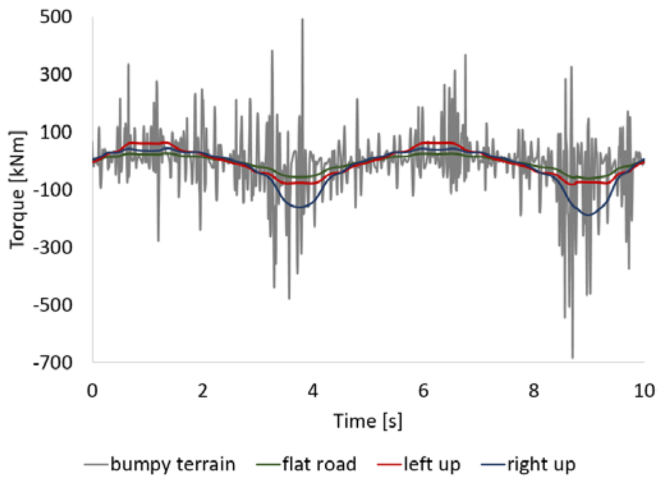

Fig. 16 *Magnitude of the operating torque on the steering wheel.*

4. Cocnclusion

Demonstration of development and utilization of the mathematic model of the truck was the purpose of this paper, because simulating technologies offer a lot of advantages – e.g. evaluation of various vehicle modifications and simulation of various critical operational states of vehicle.

Outcomes of the developed mathematic model show the behaviour of the front suspension system with the steering mechanism. Wheel steering angle depends on the geometry of the steering mechanism and vertical position of the wheels. Steering input torque was the next analysed value. From the simulation we can find that magnitude of this torque depends on vertical movement of the wheels and turning side. In the graph (Figure 16) we can see that (in case with the right wheel ascended) the torque during the right turning is higher than during the left turning (blue line). Course of this toque has highly dynamic character and the highest values are during the ride on the bumpy road (grey line).

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