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8×8 wheeled vehicle modeling in a multibody dynamics simulation software

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Abstract

The article describes a three-dimensional non-linear dynamic model of a 8×8 wheeled vehicle. The authors used the "Universal Mechanism" MBS software to build the model. The article also presents a tire – rigid terrain interaction model built in Matlab/Simulink. The authors tested the process of linking a Matlab/Simulink DLL to the vehicle MBS model. The authors used the developed model to analyze the wheeled vehicle dynamic behavior at different operation conditions. The article contains the results of the lane change test simulation. The simulation results confirmed the validity of the model. The developed model allows the estimation of the wheeled vehicle dynamic behavior at various operation conditions at the design stage.

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Keywords: vehicle dynamics; simulation; tire model; multibody dynamics simulation software.

Introduction

Reliable estimation of the wheeled vehicles main operation characteristics at the design stage needs development of three-dimensional dynamical models describing suspension and steering gear kinematics. The number of bodies in the models of multi-axle wheeled vehicles can amount to several dozen which makes the analytical deriving of equations of motions extremely complicated. The multibody dynamics simulation (MBS) software is an effective tool for solution of this sort of problems [1, 2, 3, 4, 5]. In the MBS software, the user describes the mechanical system as a set of rigid bodies, joints, and force interactions from the library of standard elements, and the software generates the equations of motion automatically and provides built-in means for their numerical solution. Besides, most of the MBS software packages allow linking user models of the force interactions as DLLs.

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This article deals with the problem of building the dynamical model for calculation of the main operation characteristics of a multi-axle wheeled vehicle in the MBS software package “Universal Mechanism” and linking the Matlab/Simulink tire – rigid terrain interaction model developed for this purpose to the MBS model.

1. Wheeled Vehicle Model

The subject of the research is a 8x8 wheeled vehicle having mass 36 t, two steerable axles and a hydropneumatic double wishbone independent suspension for all wheels. The vehicle model consists of the subsystems shown in Fig. 1. Models of the braking system and power train system are included into the model of the assembled vehicle.

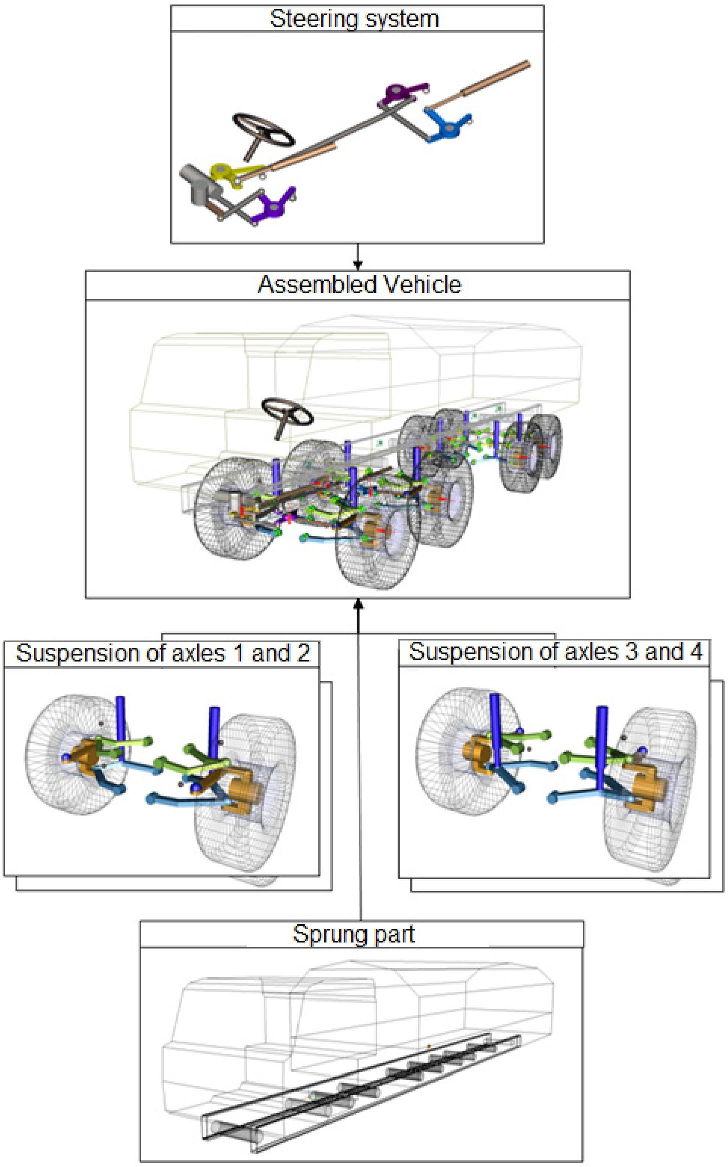

Fig. 1. Structure of the 8×8 wheeled vehicle model.

The sprung mass model consists of a massless geometry model and a mass-inertial model presented by the point mass and tensor of inertia.

Suspension and steering linkages are modeled as a set of rigid bodies connected by ideal joints and force elements. Masses of the suspension and steering system bodies are input as the model parameters and inertia are calculated from the geometry models.

The hydropneumatic spring is modeled as a system of spring and damper coupled in parallel and having tabular characteristics. There are no hydraulic connections between the hydraulic cylinders of the suspension.

The braking system model distributes the total braking torque generated according to the control signal of the braking system between the wheels of the vehicle by the selected scheme. The power train model distributes the engine output torque generated according to the control signal of the power train system equally between the wheels of the vehicle, which corresponds to the drive train with open differentials for all axles. For simulations of different tests we used two different power train models. For the tests which require maintaining constant vehicle velocity we created the variant with the engine having a constant output power and a constant gear ratio of the transmission. For analysis of the dynamics of the vehicle acceleration we built the powertrain model with analytical description of the external characteristic of the internal combustion engine and a variable gear ratio of the power train.

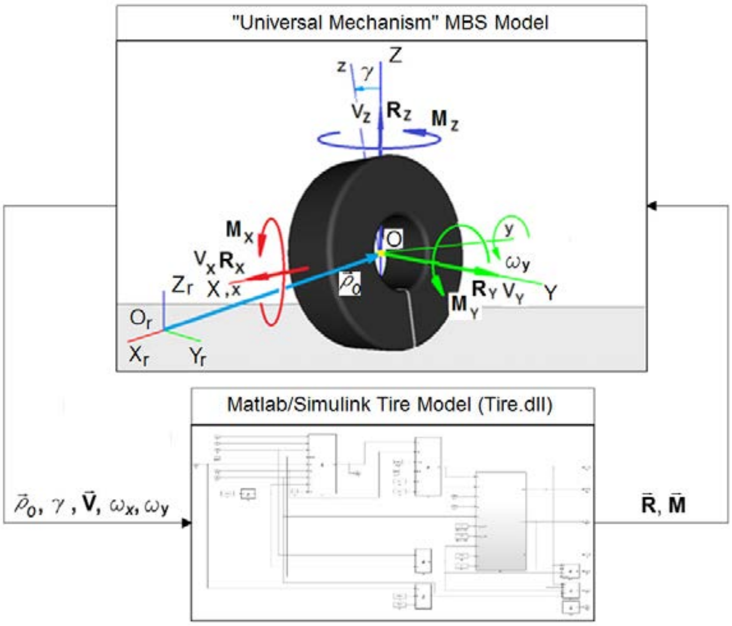

Fig. 2. Interaction of the “Universal Mechanism” MBS model and the Matlab/Simulink tire model.

2. Tire-Road Interaction Model

Road reaction forces and moments acting on the wheel are calculated by the tire model created in Matlab/Simulink and compiled as the DLL module Tire.dll. The schematic diagram of the interaction between the tire model and the “Universal Mechanism” model of the wheel dynamics is shown in Fig. 2.

The MBS model of the wheel dynamics transmits kinematic parameters of the wheel to the tire model. On the base of these parameters the tire model calculates the forces and moments acting on wheel and send them to the wheel dynamics model. The models use the following coordinate systems (see Fig. 2):

wheel stability coordinate system (SCS) $OXYZ$ – a movable coordinate system whose origin is at the wheel center, Z axis is perpendicular to the road plane, X axis is perpendicular to the wheel rotation axis;

road fixed coordinate system (FCS) $O_r X_r Y_r Z_r$ – an orthogonal coordinate system fixed on the road.

The transmitted parameters include the following: projections of the wheel center radius vector $\vec{\rho}_O$ onto the axes of the FSC, projections V_X, V_Y, V_Z of the wheel center velocity onto the wheel SCS axes, projections ω_x, ω_y of the wheel rotation velocity onto the wheel coordinate system axes, and wheel camber angle γ.

Schematic diagram of the tire – rigid terrain interaction model is shown in Fig. 3.

The vertical reaction is calculated by the visco-elastic model:

$$R_Z = P_{z\_st} \cdot \left(\frac{h_z}{h_{z\_st}} \right)^{1.5} - b_z \cdot V_Z , \tag{1}$$

here P_{z_st} – static wheel load, h_{z_st} – tire static deflection; b_z – vertical visco-elastic resistance force coefficient ; h_z – tire normal deflection:

$$h_z = \min(0, r_0 \cdot \cos(\gamma) - r_l) , \tag{2}$$

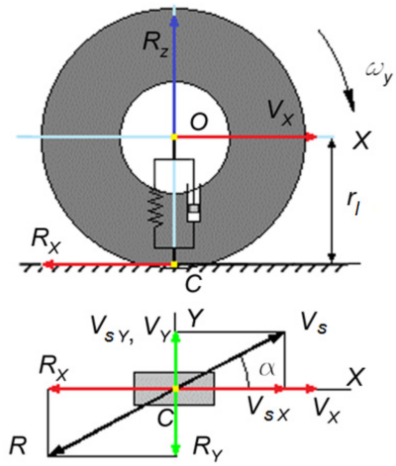

Fig. 3. Tire – road interaction model.

here r_0 – unloaded tire radius; r_l – loaded tire radius which is the projection of the wheel center radius vector onto the vertical axis of the road FSC.

The reaction in the road plane can be calculated by the following equation [6, 7]:

$$R = \mu_s(S_k) \cdot R_Z , \tag{3}$$

here S_k – tire slip; $\mu_s(S_k)$ – tire – terrain interaction coefficient.

For the cohesive soils the following equation for $\mu_s(S_k)$ can be applied [8, 9]:

$$\mu_s(S_K) = \mu_{s\alpha\,\max} \cdot \left(1 - e^{-\frac{S_K}{S_0}} \right) \cdot \left(1 + e^{-\frac{S_K}{S_1}} \right), \tag{4}$$

here S_0, S_1 – constant parameters of the curve shape; $\mu_{s\alpha\,\max}$ – coefficient of the tire – terrain interaction at complete slip:

$$\mu_{s\alpha\max} = \frac{\mu_{sx\max} \cdot \mu_{sy\max}}{\sqrt{\mu_{sx\max}^2 \cdot \sin^2\alpha + \mu_{sy\max}^2 \cdot \cos^2\alpha}}, \tag{5}$$

here $\mu_{sx\max}$, $\mu_{sy\max}$– friction ellipse parameters (see Fig. 4).

Coefficient S_k is calculated as:

$$S_k = \frac{V_s}{\omega_y \cdot r_e}, \tag{6}$$

here r_e – free rolling tire radius which can be approximately calculated by the following equation [10]:

$$r_e = \frac{3 \cdot r_l}{1 + \frac{2 \cdot r_l}{r}}; \tag{7}$$

V_s – sliding velocity defined as:

$$\begin{aligned} V_s &= \sqrt{V_{sX}^2 + V_{sY}^2} \\ V_{sY} &= V_Y - \omega_x \cdot r_l\ ; \\ V_{sX} &= V_X - \omega_y \cdot r_e. \end{aligned} \tag{8}$$

The tire – terrain interaction force vector $\vec{R}$ is opposite to the slip velocity vector $\vec{V}_{ск}$. Angle α between the slide velocity vector and the wheel SSC X axis can be found from the following equations:

$$\begin{aligned} \sin\alpha &= \frac{V_{скY}}{V_{ск}}; \\ \cos\alpha &= \frac{V_{скX}}{V_{ск}}; \end{aligned} \tag{9}$$

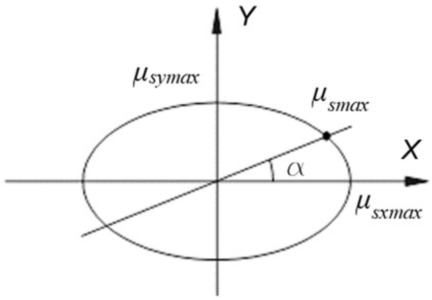

Fig. 4. Friction ellipse.

The projections of the tire – terrain interaction force in the road plane are calculated in the following way:

$$R_X = -R \cdot \cos\alpha; \qquad R_Y = -R \cdot \sin\alpha. \tag{10}$$

The moments acting on the wheel:

$$\begin{aligned} M_X &= R_Y \cdot r_l - R_Z \cdot r_l \cdot \frac{\sin(\gamma)}{\cos(\gamma)}; \\ M_Y &= -R_X \cdot r_e + M_f; \\ M_Z &= 0; \end{aligned} \tag{11}$$

here M_f – tire rolling resistance torque:

$$M_f = -R_Z \cdot r_e \cdot f \cdot sign\left(\omega_y\right), \tag{12}$$

here f – tire rolling resistance coefficient:

$$f = f_0 + k_f \cdot \left(V_X\right)^2, \tag{13}$$

here f_0 – tire rolling resistance at low speed (about 5 km/h), k_f – tire rolling resistance growth factor describing increase in the rolling resistance with the growth of the forward velocity.

3. Vehicle model testing

The authors tested the model by performing simulations of the vehicle motion at different operation conditions.

Figures 5a –5b show the time histories of the vehicle motion parameters obtained during simulation of the lane change test performed within 20 m interval according to the standard [11] at the speed 40 km/h, the maximum attainable speed at which there is no wheel lift-off. Fig. 5 shows snap shots of the characteristic positions of the vehicle.

The maximum velocity is rather low for a vehicle of this class. This can be explained by the fact that the vehicle is not equipped with antiroll bars and there are no hydraulic links between the cylinders of the hydropneumatic suspension. Considerable roll of the sprung mass during the maneuver (see Fig. 6a) implicitly confirms this conclusion.

The obtained results of the lane change test simulation confirm validity of the model.

Conclusion

The “Universal mechanism” MBS software allowed to build a spatial non-linear dynamical model of the multi-axle wheeled vehicle and to link it to the Matlab/Simulink tire – terrain interaction model compiled into a DLL module.

The developed model can provide estimation of the vehicle handling and stability as well as the vehicle dynamics at acceleration and braking at the early design stage before the production of the first prototypes.

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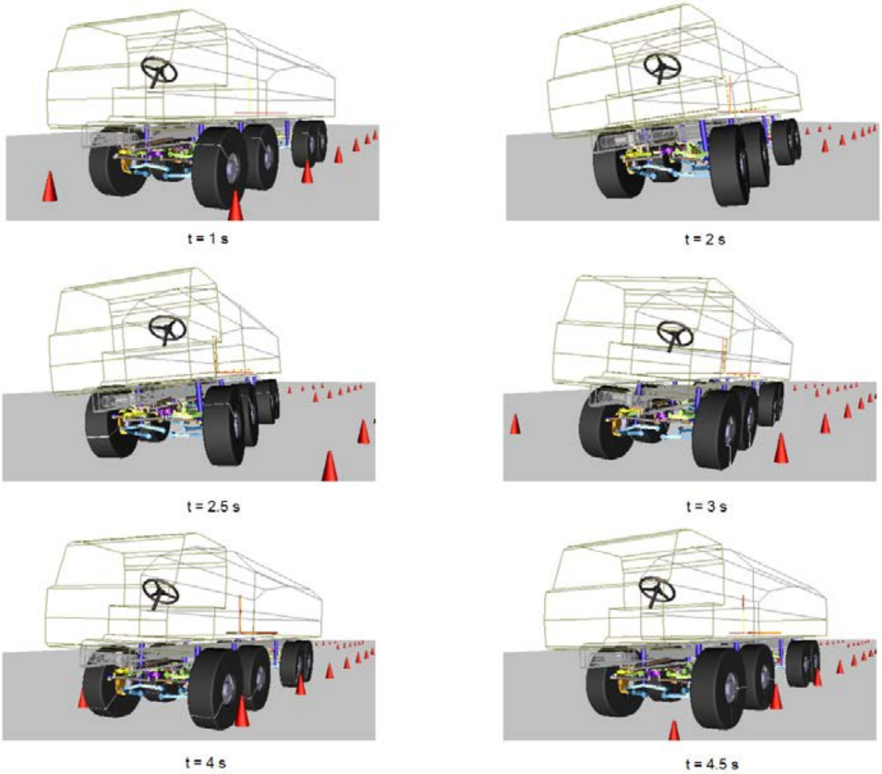

Fig. 5. Snap shots of the vehicle motion obtained during simulation of the lane change within the 20 m interval at speed 40 km/h.

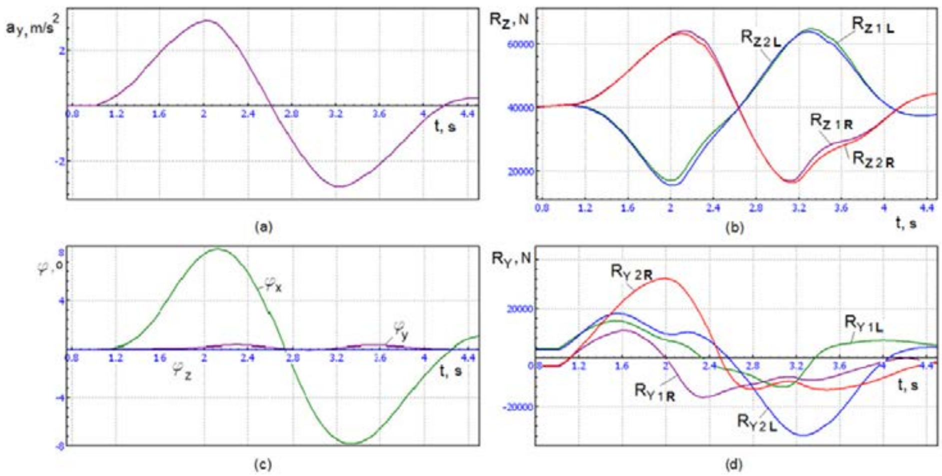

Fig. 6. Time histories of the parameters of the vehicle motion obtained during simulation of the lane change within the 20 m interval at speed 40 km/h:
(a) lateral acceleration of the sprung mass center of gravity; (b) vertical reactions of the tires of the steerable wheels (L –left side wheels; R – right side wheels);(c) roll, yaw and pitch angles of the sprung mass; (d) lateral reactions of the tires of the steerable wheels (L –left side wheels; R – right side wheels).

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