National Conference with International Participation
ENGINEERING MECHANICS 2007
Svratka, Czech Republic, May 14 – 17, 2007

ADVANCED MULTIBODY MODEL OF THE ŠKODA LOW-FLOOR TROLLEYBUS

P. Polach*, M. Hajžman*

Summary: *The structured parametric multibody model of the empty ŠKODA 21 Tr low-floor trolleybus is created using SIMPACK simulation tool. The basic SIMPACK Kinematics & Dynamics module and the SIMPACK Automotive+ module are used to create the ŠKODA 21 Tr trolleybus multibody model. In comparison with the multibody model created in* ***alaska 2.3*** *simulation tool the multibody model is extended by the model of a steering mechanism and by the model of a driving mechanism. The trolleybus multibody model is supposed to be utilized for the simulations of driving manoeuvres (driving along a predefined path, e.g. a severe lane-change manoeuvre in compliance with ISO 3888-1), braking, slow front impact against a concrete wall, running over a large road unevenness and driving along a defined uneven road surface. The aim of the simulations is the calculation of time histories or frequency responses of kinematic and dynamic quantities describing the vehicle examined properties in the chosen operational situation.*

1. Introduction

Computer softwares intended for investigating kinematic and dynamic properties of the mechanical systems are indispensable and standard tool for developing and improving properties of vehicles and also for improving comfort and passive safety of a driver and passengers – e.g. Blundell & Harty (2004), Kepka & Polach (2005), etc.

The structured parametric multibody model of the empty (i.e. of curb weight) ŠKODA 21 Tr low-floor trolleybus was created using SIMPACK simulation tool. This trolleybus type was produced in ŠKODA OSTROV s.r.o. company from 1996 till 2004.

The basic SIMPACK Kinematics & Dynamics module and the SIMPACK Automotive+ module are used to create the ŠKODA 21 Tr trolleybus multibody model. The multibody model is derived from the multibody model “with more precise kinematics of the axles suspension” (Polach, 2003b) created in **alaska 2.3** simulation tool (Maißer et al., 1998). In contrast to that model the advanced multibody model is extended by the model of a steering assembly and by the partly simplified model of a drive line. The trolleybus multibody model is supposed to be utilized for the simulations of driving manoeuvres (driving along a

---

* Dr. Ing. Pavel Polach, Ing. Michal Hajžman, Ph.D.: ŠKODA VÝZKUM s.r.o.; Tylova 1/57; 316 00 Plzeň; tel.: +420.378 182 246, fax: +420.378 182 217; e-mail: {pavel.polach,michal.hajzman}@skodavyzkum.cz

predefined path, e.g. a severe lane-change manoeuvre in compliance with ISO 3888-1), braking, slow front impact against a concrete wall, running over a large road unevenness and driving along a defined uneven road surface. The aim of the simulations is the calculation of time histories or frequency responses of kinematic and dynamic quantities describing the vehicle examined properties in the chosen operational situation.

2. Briefly about SIMPACK simulation tool

SIMPACK simulation tool (INTEC, 2006) is being developed in INTEC GmbH, Weßling, Germany. Similarly as other MBS softwares it is intended for investigating kinematic and dynamic properties of a nonlinear three-dimensional coupled mechanical system consisted of many bodies. The approach to solving the tasks in the field of mechanics using computer models, which is based on the systems of bodies, enables to solve substantially more general problems than the approach based on the finite element method because it is not dependent on the continual model of the investigated system. As a consequence of greater generality of this approach and of the character of studied mechanical systems demands for the computing time of the solution of the nonlinear equations system are growing. When creating a multibody model it is necessary to pay attention to choosing the number of bodies, the number of kinematic pairs and especially the total number of degrees of freedom in kinematic pairs of a mechanical system, i.e. to optimally interpret the physical substance of the solved problem. The total number of degrees of freedom in kinematic pairs determines the number of constructed nonlinear equations of motion, solution of which should be within a real period of time.

Multibody models are created by a finite number of bodies connected by kinematic pairs and massless force elements, which enable to model spring-damper structural parts. With respect to the multibody models creating methodology and automatic generating of the differential equations in SIMPACK simulation tool kinematic pairs are classified into two types (two separate groups within the framework of modelling in SIMPACK simulation tool) – *joints* and *constraints*. Exactly one joint with a given number of degrees of freedom belongs to each body, which enables a body motion considering the previous body in a kinematic chain. Constraints are utilized for the closing of kinematic chains, i.e. for creating kinematic loops, and constraining the relevant degree of freedom. Bodies can move in space in the framework of joints, constraints, force elements, the way of coupling to the reference frame and boundary conditions. Each body is defined by inertial properties (mass, centre of mass coordinates and moments of inertia). It is possible to bind different *markers* to the bodies. A marker is a point, in which a local coordinate system is defined. Markers can be used to locate reference frames, to define the centre of mass. Through the markers it is possible to couple bodies by joints, constraints and force elements, it is possible to act on bodies by applied forces and torques, etc. After creating a multibody model it is possible to simulate the modelled system motion. In simulating motion with multibody models in the MBS softwares nonlinear equations of motion are generated. The equations are solved by means of numerical time integration. Generally, displacements, velocities and accelerations of the individual bodies, forces and torques acting in kinematic pairs and force elements are the monitored quantities. It is possible to obtain results in the form of time series, in the form of graphs or in the form of multibody model visualisation (static or with animation). In outputs in the form of graphs it is possible to compare e.g. influences of changes of various parameters of the multibody model on the simulations results, it means operatively evaluate influences of permitted design adjustment to the desired kinematic and dynamic properties of the real structure.

Besides the basic SIMPACK Kinematics & Dynamics module it is possible to buy additional SIMPACK simulation tool modules and data interfaces with other software. In ŠKODA VÝZKUM s.r.o. there are at disposal the SIMPACK Automotive+ module (support of road vehicles modelling including tire models), the SIMPACK Wheel/Rail module (support of rail vehicles modelling including wheel-rail contact models) and the SIMPACK Contact module (support of contacts between bodies modelling).

3. Topology of the multibody model

Source for the creation of the multibody model of the empty (of mass approx.10 900 kg) ŠKODA 21 Tr low-floor trolleybus were especially research report Polach (2003b), in which multibody models of this trolleybus created in **alaska 2.3** simulation tool (Maißer et al., 1998) are described, and documentation provided by ŠKODA OSTROV s.r.o. (numerical data and technical documentation – see Polach, 2003b or Polach & Hajžman, 2006).

Fig.1 The ŠKODA 21 Tr low-floor trolleybus – the real vehicle and the multibody model visualization in SIMPACK simulation tool.

Multibody model of the ŠKODA 21 Tr low-floor trolleybus is formed by rigid bodies mutually coupled with joints, constraints and force elements. The rigid bodies correspond to the trolleybus individual structural parts or to "dummy" bodies, which are used due to the division of the multibody model into substructures of trolleybus body, front half axles, rear axle, traction motor, trolley collectors, roof unit, steering assembly, drive line, front bumper and tires. Introducing "dummy" bodies follows from approach to the multibody models creation in SIMPACK simulation tool. Air springs, hydraulic shock absorbers and bushings are modelled by connecting the corresponding bodies by force elements. Tires are modelled using Pacejka Similarity method included in the SIMPACK Automotive+ module.

In order to approximate dynamic behaviour of the vehicle more precisely the ŠKODA 21 Tr trolleybus body is divided into the front and the rear part, which are coupled by a spherical joint, in the multibody model. Using appropriately chosen torsional stiffnesses in the joint the body model enables to "tune" the values of natural frequencies corresponding to its first bending vibration modes (vertical and lateral) and to its first torsional vibration mode to the natural frequencies of the FEM model of the trolleybus body (Jankovec, 2001) created in the

COSMOS/M software (SRAC, 1999). On the basis of the similar approach a collectors model is created, too.

Multibody model of the ŠKODA 21 Tr low-floor trolleybus (“Main Model of Complex Vehicle” – see Fig.2) is created by the coupling of substructures, which correspond to the individual structural units of the trolleybus. This approach was chosen because of a good arrangement of the trolleybus multibody model, easier error identification at multibody model creating and the possibility of operational modelling in case of the potential structural modification. Trolleybus substructures of body, front half axles, rear axle, traction motor, trolley collectors, roof unit, steering assembly (see Fig.3), drive line (see Fig.4), front bumper and tires are coupled utilizing “dummy” bodies in the trolleybus multibody model.

Number of bodies corresponding to the trolleybus individual structural parts, number of joints, number of constraints and total number of degrees of freedom in joints are given in Tab.1. Tab.2 contains the list of substructures, bodies corresponding to the trolleybus individual structural units and parts, joints and constraints.

Tab.1 Number of bodies, joints, constraints and degrees of freedom of the multibody model.

Number of bodies corresponding to the trolleybus individual structural parts	45
Number of joints (without joints with “dummy” bodies)	47
Number of constraints	10
Total number of degrees of freedom in joints	92

Tab.2 Substructures, bodies, joints and constraints in the multibody model.

Substructures, bodies corresponding to structural parts, joints and constraints			
Sub-structure	**Body**	**Joint**	**Constraint**
		(axes of the coordinate system considered according to Fig.1)	
	“track_joint_19” *)	unconstrained (with respect to ground)	-
trolleybus body	front part of the trolleybus body	rigid (with respect to “track_joint_19”)	-
	rear part of the trolleybus body	spherical (with respect to the front part of the trolleybus body)	-
left of right front half axle	left front half axle	unconstrained (with respect to the front part of the trolleybus body)	-
	front suspension left lower radius arm	spherical (with respect to the left front half axle)	-
	front suspension left upper radius arm	spherical (with respect to the left front half axle)	-
	left front wheel carrier	revolute (with respect to the left front half axle, around the "**z**" axis)	connection with link (with the left steering arm)
	left front wheel	revolute (with respect to the left front wheel carrier, around the "**y**" axis)	-
	right front half axle	unconstrained (with respect to the front part of the trolleybus body)	-

Substructures, bodies corresponding to structural parts, joints and constraints			
Sub-structure	**Body**	**Joint**	**Constraint**
		(axes of the coordinate system considered according to Fig.1)	
left of right front half axle	front suspension right lower radius arm	spherical (with respect to the right front half axle)	-
	front suspension right upper radius arm	spherical (with respect to the right front half axle)	-
	right front wheel carrier	revolute (with respect to the right front half axle, around the "**z**" axis)	connection with link (with the right steering arm)
	right front wheel	revolute (with respect to the right front wheel carrier, around the "**y**" axis)	-
rear axle	rear axle	unconstrained (with respect to the rear part of the trolleybus body)	-
	rear axle left lower radius rod	spherical (with respect to the rear axle)	-
	rear axle left upper radius rod	spherical (with respect to the rear axle)	-
	left rear inside wheel	revolute (with respect to the rear axle, around the "**y**" axis)	constant transmission (with respect to the rear left wheels drive shaft)
	left rear outside wheel	rigid (with respect to the left rear inside wheel)	-
	rear axle right lower radius rod	spherical (with respect to the rear axle)	-
	rear axle right upper radius rod	spherical (with respect to the rear axle)	-
	right rear inside wheel	revolute (with respect to the rear axle, around the "**y**" axis)	constant transmission (with respect to the rear right wheels drive shaft)
	right rear outside wheel	rigid (with respect to the right rear inside wheel)	-
steering assembly (see Fig.3)	steering gear housing	rigid (with respect to the front part of the trolleybus body)	-
	steering gear housing (in lower mounting position to the chassis frame)	rigid (with respect to the front part of the trolleybus body)	-
	steering gear housing (in upper mounting position of the chassis frame)	rigid (with respect to the front part of the trolleybus body)	-

Substructures, bodies corresponding to structural parts, joints and constraints

Sub-structure	Body	Joint	Constraint
		(axes of the coordinate system considered according to Fig.1)	
steering assembly (see Fig.3)	steering wheel	revolute (with respect to the steering gear housing, around the "**z**" axis)	-
	steering gear arm	revolute (with respect to the steering gear housing, around the "**y**" axis)	constant transmission (with respect to the steering wheel angle), connection with link (with the left steering arm)
	left steering arm	revolute (with respect to the front part of the trolleybus body, around the "**z**" axis)	-
	right steering arm	revolute (with respect to the front part of the trolleybus body, around the "**z**" axis)	connection with link (with the left steering arm)
roof unit	roof unit	prismatic (with respect to the rear part of the trolleybus body, in the "**z**" axis direction)	-
traction motor	traction motor	user defined – with one prismatic degree of freedom (with respect to the rear part of the trolleybus body, in the "**z**" axis direction) and two revolute degrees of freedom (with respect to the rear part of the trolleybus body, around the "**x**" and "**y**" axes)	-
front bumper	front bumper	prismatic (with respect to the front part of the trolleybus body, in the "**x**" axis direction)	-
drive line (see Fig.4)	differential input shaft	revolute (with respect to the rear axle, around the "**x**" axis)	-
	left differential output shaft	revolute (with respect to the rear axle, around the "**y**" axis)	differential (with respect to the differential input shaft)
	rear left wheels drive shaft	revolute (with respect to the rear axle, around the "**y**" axis)	constant transmission (with respect to the left differential output shaft)
	right differential output shaft	revolute (with respect to the rear axle, around the "**y**" axis)	differential (with respect to the differential input shaft)
	rear right wheels drive shaft	revolute (with respect to the rear axle, around the "**y**" axis)	constant transmission (with respect to the right differential output shaft)

Substructures, bodies corresponding to structural parts, joints and constraints			
Sub-structure	**Body**	**Joint**	**Constraint**
		(axes of the coordinate system considered according to Fig.1)	
trolley collectors	collector base	prismatic (with respect to the rear part of the trolleybus body, in the "**z**" axis direction)	-
	first part of the left collector	universal (with respect to the collector base, around the "**y**" and "**z**" axes)	-
	second to fifth part of the left collector	universal (with respect to the previous part of the left collector, around the "**y**" and "**z**" axes)	-
	first part of the right collector	universal (with respect to the collector base, around the "**y**" and "**z**" axes)	-
	second to fifth part of the right collector	universal (with respect to the previous part of the right collector, around the "**y**" and "**z**" axes)	-
*) "track_joint_19" is an element of the SIMPACK Automotive+ module, that enables to join movement of the vehicle sprung mass with predefined track in space and to describe vehicle location on the basis of the track arc length and the vehicle speed.			

Kinematic scheme of the ŠKODA 21 Tr trolleybus substructured multibody model is in Fig.2, kinematic schemes of the steering assembly and the drive line substructures are in Figs 3 and 4.

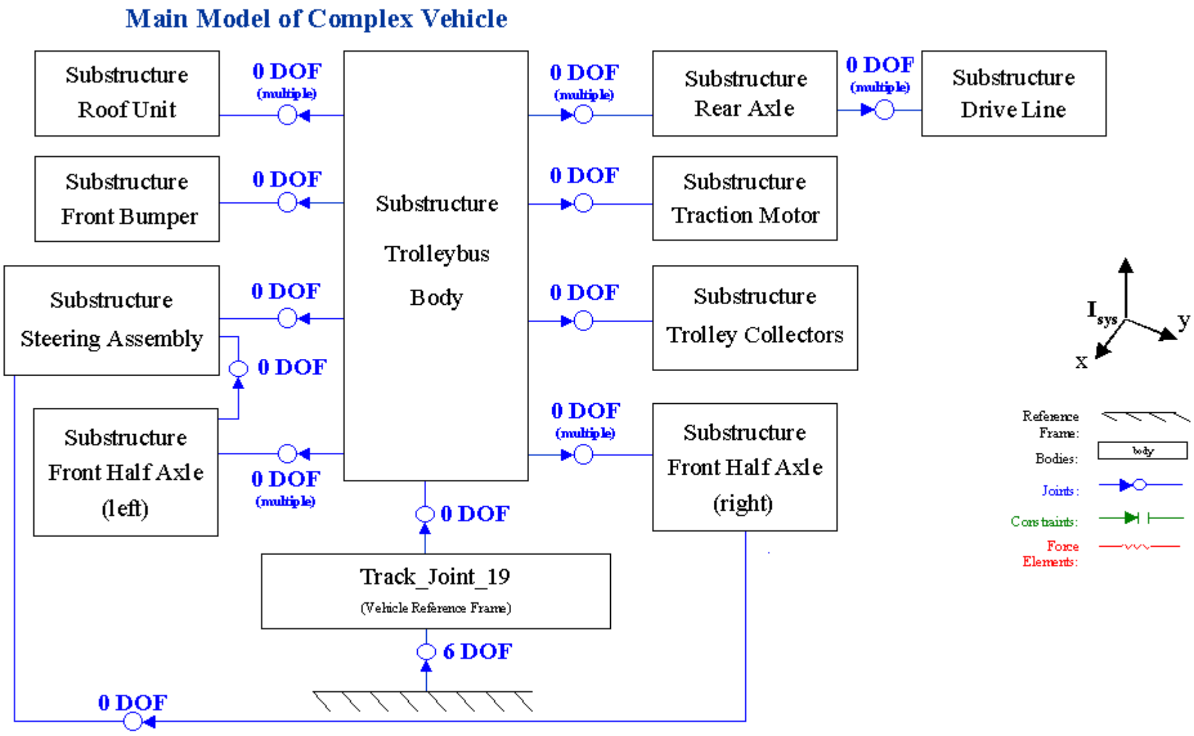

Fig.2 Kinematic scheme of the trolleybus substructured multibody model.

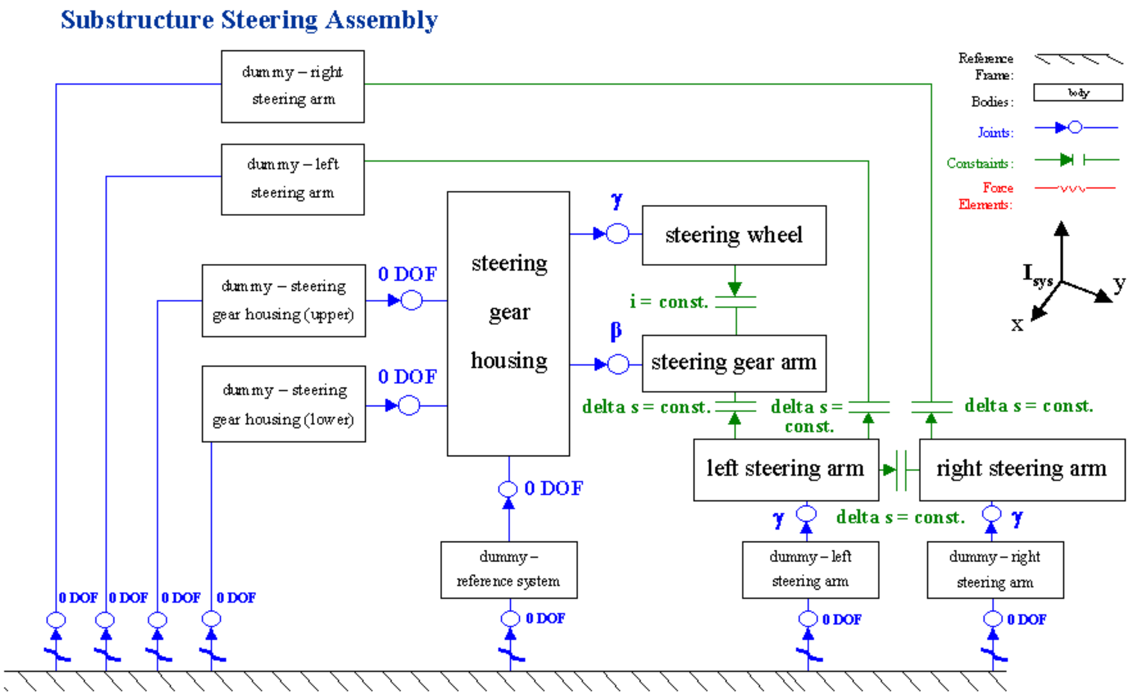

Fig.3 Kinematic scheme of the steering assembly substructure.

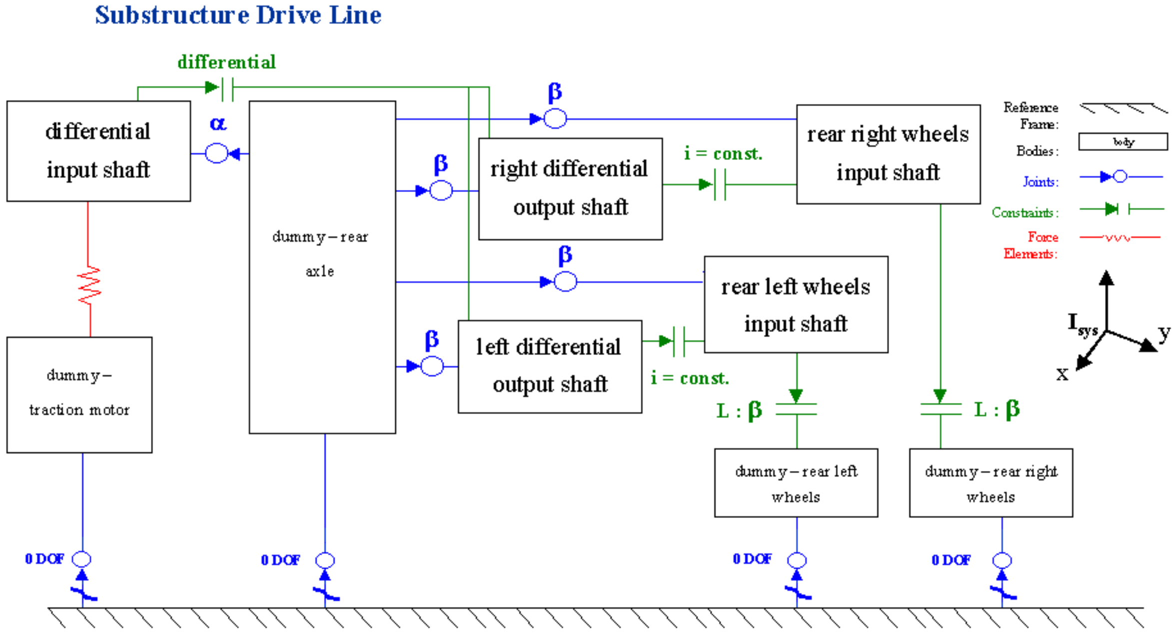

Fig.4 Kinematic scheme of the drive line substructure.

4. Force elements in the multibody model

The ŠKODA 21 Tr trolleybus structural parts modelled in the multibody model using force elements are given in Tab.3. The force elements are used for modelling e.g. air springs, hydraulic shock absorbers and bushings in positions of mounting structural parts in the trolleybus multibody model.

Tab.3 Force elements.

Structural part (force element between bodies)
division of the trolleybus body (rear part of the trolleybus body – front part of the trolleybus body)
front axle air springs (front half axles – front part of the trolleybus body)
rear axle air spring (rear axle – rear part of the trolleybus body)
front axle shock absorbers (front half axles – front part of the trolleybus body)
rear axle shock absorbers (rear axle – rear part of the trolleybus body)
bushing in positions of mounting of the front half axles to the chassis (front half axles – front part of the trolleybus body)
bushing in positions of mounting of the front suspension radius arms to the chassis frame (front suspension radius arms – front part of the trolleybus body)
bushing in positions of mounting of the front suspension radius arms to the front half axles (front suspension radius arms – front half axles)
bushing in positions of mounting of the rear axle radius rods to the chassis frame (rear axle radius rods – rear part of the trolleybus body)
bushing in positions of mounting of the rear axle radius rods to the rear axle (rear axle radius rods – rear axle)
joint rear inside wheels – rear axle (rear inside wheels – rear axle)
bushings in positions of the traction motor mounting (traction motor – rear part of the trolleybus body)
motor clutch and the differential input shaft (differential input shaft – traction motor)
positions of mounting of the front bumper (front bumper – front part of the trolleybus body)
rubber elements of the front bumper (front bumper)
coil springs of the collectors (first part of the collectors – collector base)
division of the collectors (first to fifth parts of the collectors)
contact of trolley shoe and traction line wire in vertical direction (“track_joint_19” – fifth part of collectors)
contacts of trolley shoe and traction line wire in horizontal plane (“track_joint_19” – fifth part of collectors)
contact of wheels and road (wheels – ground)

Dynamic properties of road vehicles are influenced most by suspension springs, hydraulic shock absorbers and tires (e.g. Vlk, 2000). In order that vehicle virtual computer model should reliably approximate kinematic and dynamic properties of the real vehicle knowledge of the above mentioned crucial spring-damper structural elements′ characteristics is the important presumption.

The air springs characteristics (force in dependence on deflection) of the ŠKODA 21 Tr trolleybus were determined on the basis of the Test Reports of ŠKODA OSTROV s.r.o. (front axle air springs) and the Hydrodynamic Laboratory of the Faculty of Mechanical Engineering, TU of Liberec (rear axle air springs) (Polach, 2003b).

From the point of view of multibody simulations at hydraulic shock absorbers it is necessary to know the force acting in the shock absorber in dependence on the mutual relative movement of points of a shock absorber mounting to the chassis frame and to the vehicle axle. Functions of the shock absorbers, their structure and mathematical models of shock absorbers used in virtual models of vehicles are described e.g. in Blundell & Harty (2004) and in Hajžman & Polach (2004).

In the multibody model of the ŠKODA 21 Tr trolleybus dependence of damping force on the relative velocity of compression and rebound of the shock absorber is used as the shock absorbers characteristics. The characteristics were measured on the premises of BRANO a.s., the trolleybus producer, in the Testing Laboratory of Telescopic Shock Absorbers on the Schenck testing device, working part of which is formed by crank mechanism exciting harmonically the tested shock absorber. The measured velocity characteristics of the shock absorbers show higher or lower rate of hysteresis caused especially by the compressibility of the shock absorber filling liquid. In the multibody model application the hysteresis curve values were averaged so that the resulting characteristics might be a simple curve without a hysteresis loop (Hajžman & Polach, 2004).

In order to define the ŠKODA 21 Tr trolleybus multibody model more precisely force-velocity characteristics of the shock absorbers used in the vehicle structure up to the velocities of compression and rebound higher than ± 0.8 m/s (front shock absorber in the velocity range ± 1.5 m/s, rear shock absorber due to the failure in the testing device only in the velocity range from – 1 m/s up to + 0.8 m/s) were measured in the BRANO a.s. Testing Laboratory of Telescopic Shock Absorbers in September 2004 (Polach & Hajžman, 2005a).

Rubber bushings used in the points of mounting the hydraulic shock absorbers to the chassis frame and the axles of the trolleybus are not included in the multibody model. On the basis of previous experience consideration of deformation characteristics of these bushings has only a negligible influence on the results of the simulations of the anticipated operational situations (Polach & Hajžman, 2005a).

Due to the fact that tires are modelled using Pacejka Similarity method included in the SIMPACK Automotive+ module, it is possible to consider only linear stiffnesses and linear damping coefficients of tires. It was not possible to use radial stiffness and damping properties of tires measured experimentally (Polach & Hajžman, 2007) or computed using their FEM model (e.g. Krmela, 2005). The linear radial stiffness of a standard tire was chosen 985 000 N/m (the value was provided by the trolleybus producer ŠKODA OSTROV s.r.o.), the other stiffnesses were derived from the radial stiffness. The linear radial damping coefficient of standard tire was chosen 1000 N·s/m (Polach & Hajžman, 2007), the other radial damping coefficients were derived from the radial damping coefficient.

Torsional stiffnesses of the front suspension radius arms bushings were taken from the technical documentation of ŠKODA OSTROV s.r.o. (see e.g. Polach, 2003b). Stiffnesses of the bushings in the assembly eyes for connecting rear axle radius rods and chassis frame were taken from the technical documentation of the Lemförder Metallwaren and Autófelszerelési Vállalat Sopron companies (see e.g. Polach, 2003b).

In the multibody model of the ŠKODA 21 Tr low-floor trolleybus a bumper which is a product of ŠKODA OSTROV s.r.o. and belongs to the standard equipment of the ŠKODA 14 Tr trolleybus is considered. The bumper consists of a steel part and two identical rubber elements. The steel part is firmly fixed to the body frame. Both rubber elements are symmetrically attached in front of the steel part (during the front impact the rubber elements are the first to come in contact with the obstacle). The static loading characteristic of the bumper steel part was determined on the basis of the result of the COSMOS/M FEM software calculation (SRAC, 1999), in which the half of the bumper steel part model was loaded in the point of the rubber element fixing (Zámečník, 2000). Deformation characteristics of two force elements modelling elastic properties of rubber elements of the bumper were determined on the basis static loading characteristics of rubber elements (Bártík et al., 1999), measured experimentally in the Accredited Dynamic Testing Laboratory of ŠKODA VÝZKUM s.r.o. on the SCHENCK 400 kN hydraulic loading machine.

The traction characteristics of the ŠKODA 21 Tr trolleybus were provided by ŠKODA OSTROV s.r.o. The specification of dependence of a motor driving force transmitted to the rear wheels on the trolleybus running speed is used in the multibody model (in Fig.5 designated FT9). In the trolleybus multibody model this characteristic is converted to driving torque transmitted from the traction motor to the differential gear (with wheel radius 460 mm and with total axle drive reduction ratio 5.427). The driving torque acts between the traction motor clutch and the differential input shaft (in the joint coupling the bodies of a differential input shaft and a traction motor – see Tab.2 and Fig.4). Transmission of driving torque to the differential input shaft is controlled by the demand on the trolleybus instantaneous running speed (see INTEC, 2006). This driving torque is transmitted by the differential gear up to the constraints coupling the rear inside wheels and the rear wheels drive shafts – see Tab.2 and Fig.4.

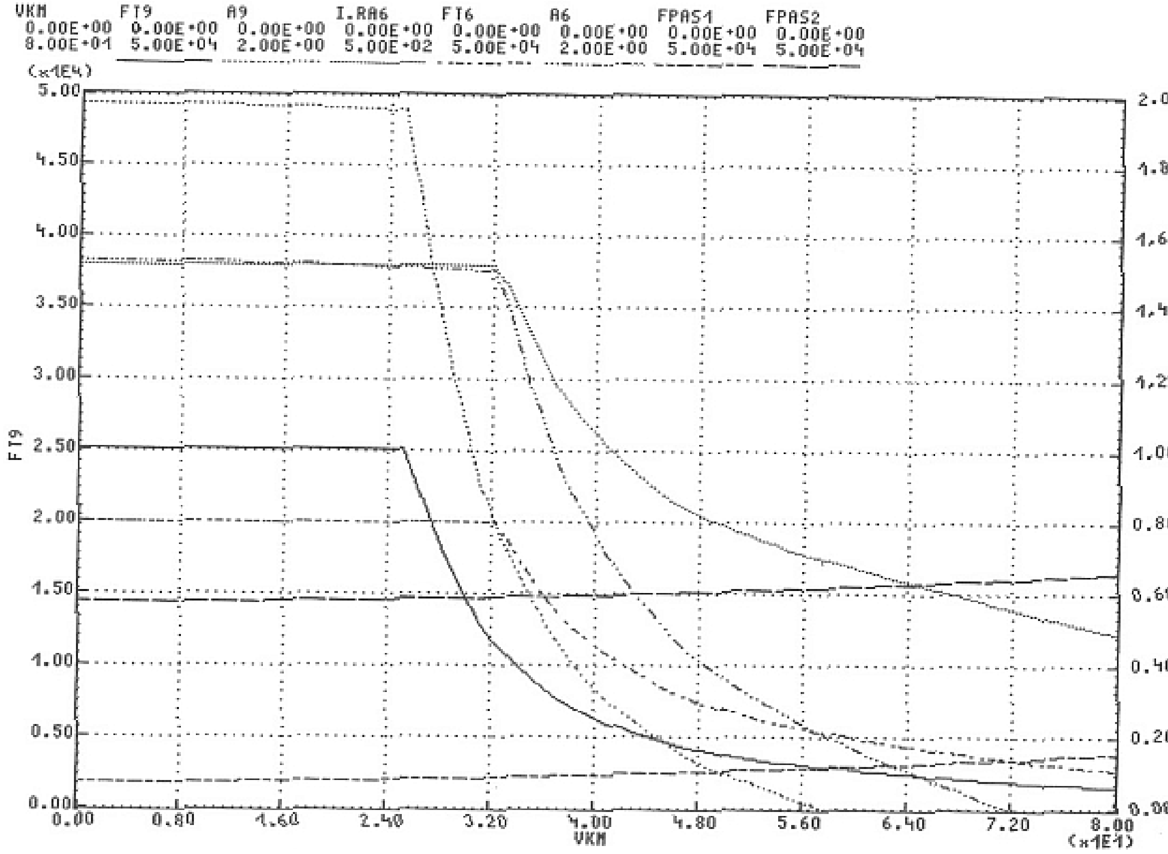

Fig.5 Traction characteristics of the ŠKODA 21 Tr low-floor trolleybus.

5. The trolley collector model

The model of the ESKO trolley collector consists of five (the same as in Polach, 2003a) rigid bodies mutually coupled by universal joints. Using appropriately chosen torsional stiffnesses in the kinematic joint the values of three lowest natural frequencies corresponding to the bending vibration modes of the collector are “tuned” to the values determined at the experimental measurement (Polach, 2003a and Tab.4).

Tab.4 Natural frequencies and natural modes of the trolley collector.

Vibration mode	Natural frequency			
	Measurement			**Multibody model**
	Free collector	Collector on the traction line		Collector on the traction line
	Vertical and horizontal vibration modes	Vertical vibration mode	Horizontal vibration mode	Vertical and horizontal vibration modes
1st bending	4.5 Hz	4.75 Hz	5.5 Hz	4.96 Hz
2nd bending	11.75 Hz	12 Hz	11.25 Hz	12.49 Hz
3rd bending	18 Hz	20.25 Hz	19 Hz	20.37 Hz

The wire of the traction line, which is in contact with the collector, is modelled using *ximpact* penalty functions (Maißer et al., 1998) („barriers“ in the directions „upward“, „to the right“ and „to the left“). *Ximpact* function is of the form:

$$ximpact\left(x, x_p, x_1, c, d, e, k\right) = \begin{cases} -c \cdot \left(x_1 - x\right)^e + step\left(x, x_1 - d, k, x_1, 0\right) \cdot x_p & , \quad x < x_1 \\ 0 & , \quad x_1 \leq x \end{cases} \tag{1}$$

where c, d, e, k are coefficients characterizing the spring-damper properties of the “barrier” (wire), x is an independent variable, x_p is derivation x with respect to the time and x_1 is the distance of the “barrier” (wire).

Step quasi-step function approximates a step function (modified Heaviside step function) by evaluating a cubic polynomial:

$$step\left(x, x_0, h_0, x_1, h_1\right) = \begin{cases} h_0 & , \quad x \leq x_0 \\ h_0 + \left(h_1 - h_o\right) \cdot \left(\dfrac{x - x_0}{x_1 - x_0}\right)^2 \cdot \left(3 - 2 \cdot \dfrac{x - x_0}{x_1 - x_0}\right) & , \quad x_0 < x < x_1 \\ h_1 & , \quad x_1 \leq x \end{cases} \tag{2}$$

where x is an independent variable, x_0 is the point, up to the functional value of the *step* function is h_0, and x_1 is the point, from to the functional value of the *step* function is h_1.

According to the data provided by ŠKODA OSTROV s.r.o. the static downforce of the trolley shoe to the traction line wire is approximately 60 N (Polach, 2003a). In the ŠKODA 21 Tr multibody model this value of downforce is achieved by choice of the rate of the coil springs which vertically press the collector to the traction line (in the multibody model there are not used real rates of the coil springs).

6. Conclusions

The paper deals with the structured parametric multibody model of the empty ŠKODA 21 Tr low-floor trolleybus created using SIMPACK simulation tool (INTEC, 2006). The multibody model is derived from the multibody model “with more precise kinematics of the axles suspension” (Polach, 2003b) created in **alaska 2.3** simulation tool (Maißer et al., 1998). In contrast to that model the advanced multibody model is extended by the model of the steering assembly and by the partly simplified model of the drive line.

It is supposed that due to extending the multibody model by the steering assembly model and the drive line model, time histories of the monitored quantities during the simulation of the operational situations will be determined more precisely. Especially improving in the correspondence of the results of the simulations of driving along the test track consisting of artificial vertical obstacles with the results of the experimental measurement performed with the empty real trolleybus in the Hradec Králové Public City Transit Co. Inc. depot in October 2004 (e.g. Polach & Hajžman, 2005b, Polach & Hajžman, 2007) was the motivation for the advanced multibody model creation. When simulating test drives with the so far used virtual models, it has been necessary, due to the software limitations, to consider the constant speed of the vehicle. The drive line model in the advanced multibody model enables to keep the prescribed instantaneous speed of the trolleybus. Driving manoeuvres, for the simulations of which the steering assembly model creation is useful, cannot be compared with the experimental measurements at the ŠKODA 21 Tr low-floor trolleybus. Simulation of driving manoeuvres will be only of the character of the verification with the other virtual models. But the steering assembly model created on the basis of the same approach is supposed to be implemented e.g. into the multibody model of the ŠKODA 22 Tr low-floor articulated trolleybus, with which the operational tests focused on the investigation of driving stability were performed and documented (e.g. Polach, 2007).

7. Acknowledgement

The paper has originated in the framework of solving the Research Plan of the Ministry of Education, Youth and Sports of the Czech Republic MSM4771868401.

8. References

Bártík, J., Frémund, J. & Kotas, M. (1999) *Test of a Bumper Stiffness of the 14 Tr-SF Trolleybus*. Research Report ŠKODA VÝZKUM s.r.o. VYZ 0294/99, Plzeň. (in Czech)

Blundell, M. & Harty, D. (2004) *The Multibody Systems Approach to Vehicle Dynamics*. Elsevier Butterworth-Heinemann, Oxford.

Hajžman, M. & Polach, P. (2004) Hydraulic Shock Absorbers Modelling in Trolleybus Multibody Simulations of Running over a Large Road Unevenness, in: *Proc. of National Conference with International Participation Engineering Mechanics 2004* (I. Zolotarev & A. Poživilová eds), Institute of Thermomechanics AS CR, Svratka, CD-ROM. (in Czech)

Jankovec, J. (2001) *FEM Model of the Škoda Tr 21 Trolleybus for Plastic Program*. Research Report ŠKODA VÝZKUM s.r.o. VYZ 0548/2001, Plzeň. (in Czech)

Kepka, M. & Polach, P. (2005) Testing and Computing of Vehicles in SKODA VYZKUM. in: *Proc. of Combined Conference on Heavy Vehicles XXXVI. Meeting of Bus and Coach Experts and Congress on Commercial Vehicles* (A. Voith & É. Szatmári eds), Hungarian Scientific Society of Mechanical Engineers, Budapest, CD-ROM.

Krmela, J. (2005) Computational Simulations of Long-fibre Elastomer Matrix Composites with Steel Reinforcement, in: *Proc. of the III. International Conference Dynamics of Rigid* *and Deformable Bodies 2005* (B. Skočilasová ed.), ÚTŘV UJEP in Ústí nad Labem, Ústí nad Labem, pp. 71-78. (in Czech)

Maißer, P., Wolf, C.-D., Keil, A., Hendel, K., Jungnickel, U., Hermsdorf, H., Tuan, P.A., Kielau, G., Enge, O., Parsche, U., Härtel, T. & Freudenberg, H. (1998) ***alaska**, User Manual, Version 2.3*. Institute of Mechatronics, Chemnitz.

Pacejka, H.B. (2002) *Tyre and Vehicle Dynamics*. Butterworth-Heinemann, Oxford.

Polach, P. (2003a) Approaches to the Creation of the ŠKODA Low-Floor Trolleybus Multibody Models with a Divided Front Axle, in: *Proc. of the 19th Conference with International Participation Computational Mechanics 2003* (J. Vimmr ed.), Department of Mechanics FAS UWB in Pilsen, II, pp. 367-374. (in Czech)

Polach, P. (2003b) *Multibody Models of the ŠKODA 21 Tr Low-Floor Trolleybus – Modification with a Divided Front Axle*. Research Report ŠKODA VÝZKUM s.r.o. VYZ 0651/2003, Plzeň. (in Czech)

Polach, P. (2007) Design and Verification of the Stabilizer Bar of the Articulated Trolleybus, in: *Proc. of ECCOMAS Thematic Conference Multibody Dynamics 2007* (C.L. Bottasso, P. Masarati & L. Trainelli eds), Politecnico di Milano, Milano, CD-ROM.

Polach, P. & Hajžman, M. (2005a) Influence of the Hydraulic Shock Absorbers Model in Trolleybus Multibody Simulations on the Suspension Deformations and Comparison with the Experimental Results, in: *Proc. of National Conference with International Participation Engineering Mechanics 2005* (V. Fuis, P. Krejčí & T. Návrat eds), Institute of Thermomechanics AS CR, Svratka, CD-ROM.

Polach, P. & Hajžman, M. (2005b) Various Approaches to the Low-floor Trolleybus Multibody Models Generating and Evaluation of Their Influence on the Simulation Results, in: *Proc. of ECCOMAS Thematic Conference Multibody Dynamics 2005 on Advances in Computational Multibody Dynamics* (J.M. Goicolea, J. Cuadrado & J.C. García Orden eds), Universidad Politécnica de Madrid, Madrid, CD-ROM.

Polach, P. & Hajžman, M. (2006) *Multibody Model of the ŠKODA 21 Tr Low-floor Trolleybus in SIMPACK software*. Research Report ŠKODA VÝZKUM s.r.o. VYZ 0890/06, Plzeň. (in Czech)

Polach, P. & Hajžman, M. (2007) Multibody simulations of trolleybus vertical dynamics and influences of tire radial characteristics, in: *Proc. of The 12th World Congress in Mechanism and Machine Science*, The French IFToMM Committee, Besançon, CD-ROM.

Vlk, F. (2000) *Dynamics of Motor Vehicles*. VLK Publishing House, Brno. (in Czech)

Zámečník, Š. (2000) *Calculation of the Trolleybus Bumper Stiffness with Respect to the Material Non-Linearity*. Research Report ŠKODA VÝZKUM s.r.o. VYZ 0412/2000, Plzeň. (in Czech)

INTEC (2006) *SIMPACK 8.800, User Manual*. INTEC GmbH, Weßling.

SRAC (1999) *COSMOS/M, Finite Element Analysis System, User Guide, Version 2.5*. SRAC, Los Angeles.