TMEASY FOR RELIABLE VEHICLE DYNAMICS SIMULATION

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ABSTRACT

The mission of this paper is to provide some brief information about the contribution of the tyre model, the TMeasy, to reliable and accurate full-vehicle dynamic simulations, which are carried out in the MSC.ADAMS environment. The vehicle dynamic simulations become an efficient method for choosing the right concepts at early developmental stages, but the process of virtual model development requires accurate properties of the tyre models, based on reliable input data. As one pragmatic approach, the lean semi-physical TMeasy tyre model may help to overcome the lack of reliable tyre input data, because the set of tyre model parameters is manageable and has got a clear, physical meaning. To strengthen the trust in the results from simulations, ~~a~~ special attention was paid to the accurate modelling of springs, dampers, bushes and the flexible crossbeam, and the results obtained were verified using real measurements from testing manoeuvres. The semi-physical tyre mode, the TMeasy, combines the advantages of the closed algebraic approaches with a lean parameter set of physical meaning and has good runtime behaviour as well as real time capability.

*Keywords**:** vehicle, tyre, modeling, dynamics simulation.*

INTRODUCTION

The growing importance of simulations in the early stages of the automotive vehicle development process is evident thanks to great effort done to refine applied methods with respect to reliability and the accuracy of the results obtained. In the field of vehicle dynamics simulation, multibody systems (MBS) have established themselves as the essential method for modelling. Although a high quality level of complex full vehicle models could be achieved, there is still a remarkable lack of certain input data to feed the parameters of the models. Particularly in the important field of tyre modeling, an apparent lack of certain testing data has to be mentioned. It is a well known fact that the process of data acquisition is much more time consuming than the work of modelling and simulation itself. For the dynamic simulation of automotive vehicles, the “tyre/road” model element is of special importance, given its direct influence on the achievable results. It can be said that the adequate description of the interactions between tyre and road is one of the most important tasks of vehicle modelling, because all the other components of the chassis influence the vehicle dynamics properties via the tyre’s contact forces and torques. Therefore, in the interest of balanced modelling, the precision of the complete vehicle model should stand in reasonable relation to the performance of the applied tyre model, [1]. The present paper focuses on the aspect of the entire process of tyre modelling from the testing data source, via parameter identification, to MBS implementation.

TESTING DATA FOR TYRE MODELS

As already mentioned in [1], in engineering practice there often exists the problem of data availability for a special type of tyre for the examined vehicle. A considerable amount of experimental data for car tyres has been published or can be obtained from the tyre manufacturers. If one cannot find data for a special tyre, its characteristics can be estimated at least by an engineer's interpolation of similar tyre types. In the field of truck tyres there is still an undesirable backlog in data provision. These circumstances were taken into account in conceiving a user-friendly tyre model. In order to overcome the practical restrictions of incomplete and uncertain tyre measurement data, it is recommended to apply semi-physical tyre modelling. Semi-physical tyre models aim to combine the advantages of the above mentioned modelling techniques, namely to apply closed algebraic approaches with a lean parameter set of physical meaning and to have good runtime behaviour as well as real time capability. Detailed information about the availability and sensitivity of the main parameters, and the procedure for parameter identification and estimation respectively, based on a passenger car application, are introduced in [6]. What follows is the process of building up a full vehicle model of an Opel Combo 1.6 CNG is shown, where the vehicle is modelled in the MBS-system MSC.Adams [7], using the application of the semi-physical tyre model, the TMeasy [1].

MODEL OF A PASSENGER CAR

For the investigations, a light commercial vehicle, the Opel Combo 1.6 CNG ecoFLEX 69KW/94PS was available. This vehicle is equipped with front axle with independent McPherson struts with gas pre-loaded dampers and an anti roll bar. At the rear, a twist beam axle with coil springs and gas-filled shock absorbers serve as wheel suspension. The empty mass of the vehicle is 1210 kg. With the maximum allowed payload of 520 kg the gross vehicle mass yields 1730 kg. The vehicle is equipped with radial tyres Continental 185/60 R15 inflated to 2.6 bar.

The full vehicle model is build up as a multibody system (MBS) in the MSC.Adams/Car simulation software [6], see Fig. 1. The model consists of about 50 parts with 37 degrees of freedom (DOF) and it is briefly described here.

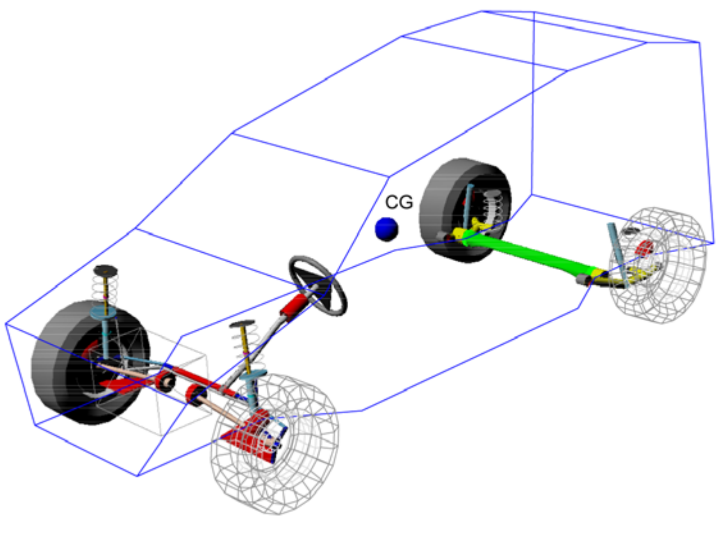

Fig. 1 MBS full model of the vehicle.

The vehicle body is built up as rigid part. The engine, gearbox and differential are modelled as a single power unit part that is fixed to the chassis part, such that the relative movement between them is suppressed according to the interesting low frequency domain of vehicle dynamics. The brake system is taken into account as a controlled torque which is applied on each wheel counter-clockwise to the spin velocity of the wheel.

The McPherson front suspension represents a standard design feature, with rigid lower control arms. The wheel carriers include the wheel bearings and connect the lower control arms with the struts and the tie rods which control the steering motion of the wheel. All these parts are connected together with joints as geometrical constraints. Springs and dampers are represented as nonlinear force elements, using ADAMS/Solver SFORCE routines, which interpolate a force versus deflection table for the spring, and force versus velocity table for the damper. The front anti roll bar consists of two rigid bars, where a torsion spring acts between them. The bars are mounted by revolute joints to the chassis and by side bars with spherical joints to the upper parts of each of the struts.

The rack and pinion steering gear translates the rotational motion of the steering wheel into the linear movement of the rack which is transmitted to the wheel carriers via rigid steering rods. The steer angle of the vehicle is controlled either trajectory based, (closed loop mode) or based on a certain input signal (open loop mode). The steering wheel can be driven by angle and/or torque input.

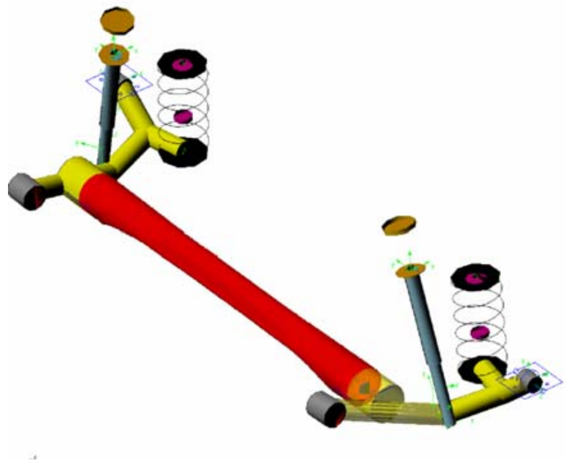

Fig.2 Rear suspension model.

The twist beam suspension in Fig.2 represents a widely used rear wheel axle. The axle sub-model consists of two rigid trailing arms, one at each side, connected by a flexible FEM crossbeam. The 3D model of the crossbeam was built and meshed by shell elements in a CATIA environment. The MNF file was generated in MSC.NASTRAN and imported into the MSC.ADAMS/Flex module. The flexible body has got 16562 nodes and, according to modal superposition, the first $M = 32$ of modal coordinates q_i, and corresponding mode shapes represent the dynamic modal properties of the flexible body in order to simulate the elastic behaviour of the structure.

$$u = \sum_{i=1}^{M} \Phi_i q_i \qquad (1)$$

As one important element in the correct side slip behaviour of the vehicle while turning, the spatial stiffness of the bushes which connect the twist beam axles with the vehicle's body have to be modelled. Similarly to the front axle, the rear springs and dampers are represented as force elements, using the ADAMS/Solver SFORCE routines. The splines of nonlinear dependency between deformation and forces on spring and bump-stops form were adjusted and verified by vertical force comparison of the measurement and simulation of vertical wheel travel.

Because the dependency between the force and velocity on the damper was not known, a linear damper approach was used. From vehicle quarter model theory it is possible to derive linear damping coefficient d_0 under consideration of damping ratio $D = 0.4$ for transport vehicles,

$$d_0 \approx 2\, D \sqrt{c_s\, m_{ch}}\,; \qquad c_t >> c_s \tag{2}$$

where c_t, resp. c_s is tyre, resp. suspension stiffness and m_{ch} is chassis mass equal to one wheel. Assuming a flat road, and therefore small damper velocities is acceptable.

All four wheels are built up as rigid rims and hubs with the overall mass and inertia properties of the wheel. The tyre is modelled as an external force element, described by the semi-physical tyre model, TMeasy, which acts between the road and the rim.

SEMI-PHYSICAL TYRE MODEL

Contact Geometry

Within TMeasy it is assumed that the contact patch is sufficiently flat. Four road points Q_1 to Q_4 located in the front, in the rear, to the left and to the right of the tyre patch are used to define the normal vector e_N of a local track plane, and to calculate the geometric contact point P on rough roads, Fig. 3. As in real tyre-road contact, sharp bends and discontinuities, which will occur at step- or ramp-sized obstacles, are smoothed by this approach.

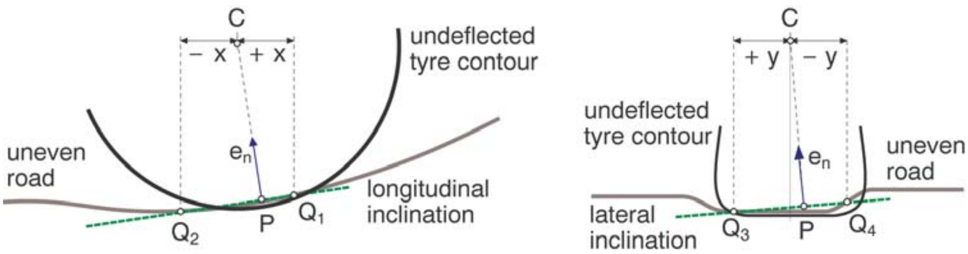

Fig. 3 Track normal and geometric contact point on uneven roads.

The directions of the longitudinal and lateral forces as well as the tyre camber angle are then derived from the direction of the wheel rotation axis and the normal track. By taking into account that the tyre deformation consists of the belt and flank deformation, a realistic approximation of the length L of the contact patch is possible. The dynamic rolling radius r_D of the tyre which is needed for average transport velocity of tread particles is calculated by a weighted sum of the undeflected and the static tyre radius.

Steady State Tyre Forces

The normal force or wheel load is separated into a static and a dynamic part:

$$F_z = F_z^S + F_z^D \tag{3}$$

The static part F_z^S is described as a nonlinear function of the tyre deflection Δz and the dynamic part F_z^D is roughly approximated by a damping force proportional to the time derivative $\Delta \dot{z}$ of the tyre deflection. Because the tyre can only apply pressure forces to the road, the normal force is restricted to $F_z \geq 0$. The longitudinal force as a function of the longitudinal slip $F_x = F_x(s_x)$ and the lateral force depending on the lateral slip $F_y = F_y(s_y)$ are defined by characteristic parameters: the initial inclination dF_x^0, dF_y^0, the location s_x^M, s_y^M and the magnitude of the maximum F_x^M, F_y^M as well as the sliding limit s_x^S, s_y^S and the sliding force F_x^S, F_y^S. By combining the longitudinal and lateral slip to a generalised slip s the combined force characteristics $F = F(s)$ can be automatically generated by the characteristic tyre parameters in longitudinal and lateral direction, [1]. In general driving situations, e.g. acceleration or deceleration in curves, the longitudinal and lateral tyre forces are the given by the projection of the generalised force characteristic into the longitudinal and lateral direction

$$F_x = F\cos\varphi = F\frac{s_x^N}{s} = \frac{F}{s}s_x^N \quad \text{and} \quad F_y = F\sin\varphi = F\frac{s_y^N}{s} = \frac{F}{s}s_y^N \tag{4}$$

where the normalised slip quantities s_x^N, s_y^N are used to improve the combined force modelling.

The self-aligning torque is approximated via the pneumatic trail which again is described by characteristic parameters. The influence of the camber angle to the lateral tyre force, and the self-aligning torque is modelled by an equivalent lateral slip and by a bore torque which is generated by the component of the wheel rotation around an axis perpendicular to the local track plane.

The mostly degressive influence of the wheel load on the steady state tyre forces and torques is taken into account by defining all characteristic tyre parameters for the payload of the tyre and its double value.

First Order Tyre Dynamics

Measurements show that the dynamic reaction of the tyre forces and torques to disturbances can be approximated quite well by first order systems, [2]. Taking the tyre deformation into account, the TMeasy approach for steady state tyre forces can easily be extended to dynamic tyre forces. The tyre forces F_x and F_y acting on the contact patch, deflect the tyre in a longitudinal and a lateral direction, Fig.4.

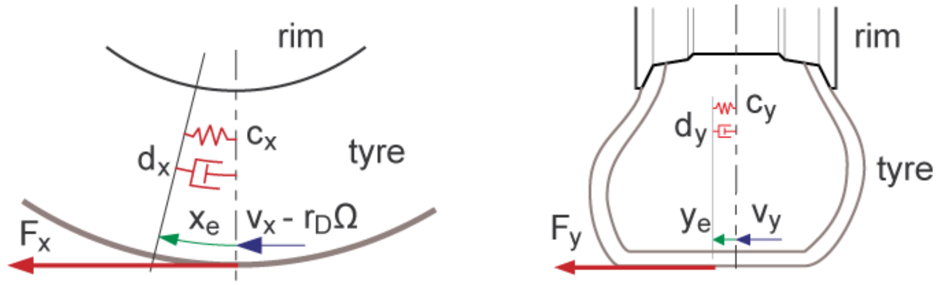

Fig. 4 Tyre deflection in longitudinal and lateral direction.

In a first order approximation, the dynamic tyre force F_x^D in the longitudinal direction yields

$$F_x^D = F_x(v_x + \dot{x}_e) \approx F_x(v_x) + \frac{\partial F_x}{\partial v_x}\dot{x}_e \approx F_x^S + \frac{\partial F_x}{\partial v_x}\dot{x}_e \tag{5}$$

where x_e names the elastical longitudinal tyre deflection and the steady state tyre force F_x^S in longitudinal direction is provided by Eq. (4). On the other hand, the dynamic tyre force can be derived from

$$F_x^D = c_x\, x_e + d_x\, \dot{x}_e \tag{6}$$

where c_x and d_x denote linear stiffness and damping properties of the tyre in the longitudinal direction. Combining eqs. (5) and (6) finally results in a first order differential equation for the longitudinal tyre deflection

$$\left(d_x\, r_D\, |\,\Omega\,|\, s_x + \frac{F}{s} \right) \dot{x}_e = -\frac{F}{s}(v_x - r_D\,\Omega) - c_x\, x_e\, r_D\, |\,\Omega\,|\, \hat{s}_x \tag{7}$$

where r_D names the dynamic rolling radius of the tyre, Ω is the angular velocity of the wheel and $\hat{s}_x$ denotes the normalisation factor of the longitudinal slip.

The modelling of the lateral tyre force dynamics is straight forward. A dynamic model of the bore torque which is needed for simulating the parking effort is described in [4].

This principally simple but effective extension to first order dynamic tyre forces and torques allows a smooth transition from normal driving situations to stand still and keeps the dynamics of the system in a finite state.

IMPLEMENTATION AND APPLICATION

The implementation of TMeasy into Adams is done via the Standard Tyre Interface (STI), [3]. This interface, currently in version 1.4, is supported today by most of the commercial simulation systems and allows the link of any STI-compatible tyre model, as far as they represent specific vehicle dynamic models with an idealised contact point.

The simulation program processes the wheel motion values in the sequence of the vehicle's wheels W_i , i = 1, 2 ... n_W at every time step to STI, which are here transformed into the internal motion values of the applied tyre model and then passed to it. As output, STI delivers the actual vectors of the tyre forces F_i and torques M_i in the specified form back to the simulation program. As necessary, from there they can be passed on to the appropriate post processor, as well as to additional tyre variables for any control purpose.

A complete set of parameters for a vehicle model thus consists of at least one road file and one tyre file for each group of identical vehicle tyres, therefore, at least of one tyre file. The correct assignment of the tyre to its related model body wheel is defined in the model file and again this is directed by STI. The two times five parameters, which can easily be changed on demand “by hand” due to their physical meaning, are depicted in Fig.5.

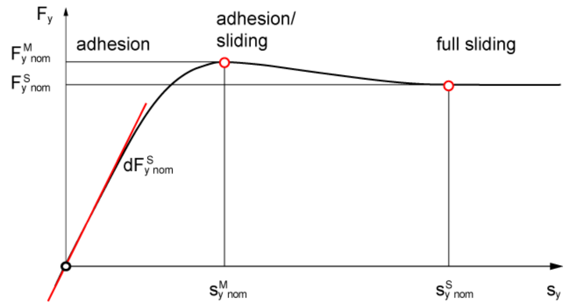

Fig. 5 Lateral force function with five parameters for $F_{z\,nom}$.

SIMULATION AND MEASUREMENT RESULTS

In the following, two selected Adams/Car - TMeasy applications are shown. An example of the comparison of results for suspension geometry characteristics (camber and toe angles) obtained from measurements on the test bench and simulation during parallel wheel travel is in Fig.6.

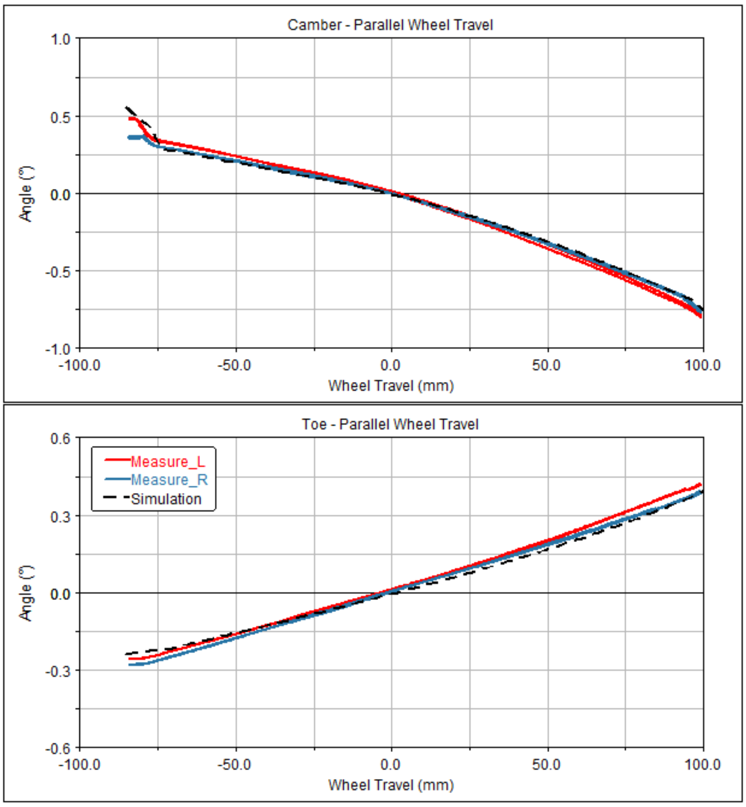

Fig. 6 Camber and toe angles during parallel wheel travel from simulations and measurements.

The testing manoeuvre (multiple) lane change is well suited to validating the vehicle model and, due to its transient nature, also to checking the dynamics capabilities of the tyre model. In the present example, the time series of the measured steering angle is applied to the vehicle model by an input driver. As it can be shown in Fig. 7, the vehicle's responses to these disturbances correlate quite well with the measured signals.

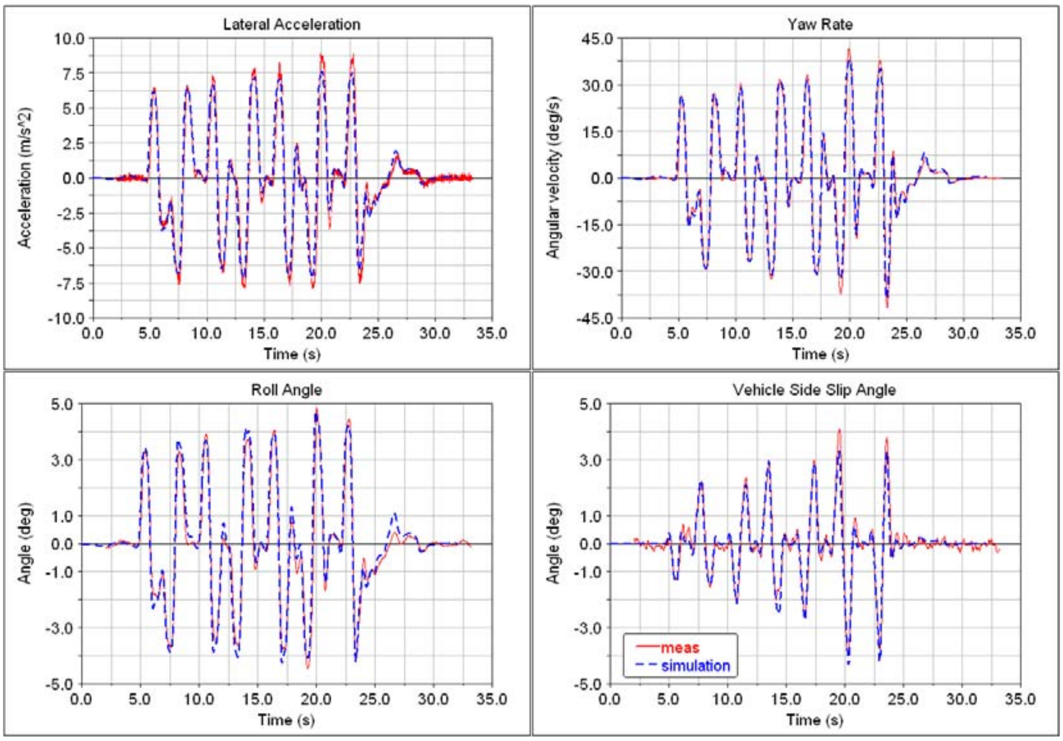

Fig. 7 Comparison simulation-measurement: Multiple lane change.

CONCLUSIONS

As was mentioned in the introduction, this paper focuses on the entire process of tyre modelling from the testing data source asessment, the parameter identification and the MBS implementation. The process of virtual model development requires accurate properties of tyre models based on reliable input data. Some experimental data for car tyres can be found in literature or can be obtained from the tyre manufacturers. If one cannot obtain data for a specific tyre, its characteristics can be estimated at least by an engineer's interpolation of similar tyre types. Evaluation of the measurement data from different sources (i.e. measuring techniques) for a special type of tyre using different measuring techniques result in widely spread results. The need for the user to use their experience to assemble a "probably best" set of data, as a basis for the tyre model from these sets of data, and to verify it eventually with their own experimental results, can be reduced by the physical meaning of the parameters of the applied tyre model.

It was shown that the lean semi-physical TMeasy tyre model may help to overcome the lack of reliable tyre input data, because the set of tyre model parameters is manageable and has got clear physical meaning.

The developed tyre model, the TMeasy, with a good correlation between simulation and experiment, was proved successful in meeting the practical requirements in use. This procedure was shown by means of a full vehicle model of a light commercial vehicle, which is modelled and verified in the multibody system MSC.Adams environment. The comparison of simulation results with measurements from testing manoeuvres allows rectification of the chosen assumptions. The acceptable correlation of results from measurements and simulations presented, confirms that the vehicle dynamic simulations have become reliable and efficient methods for choosing the suitable concepts at early development stages. The semi-physical TMeasy tyre model combines the advantages of the above mentioned modelling techniques, namely a closed algebraic approach, with small set of parameters of physical meaning and provides good runtime behaviour as well as real time capability.

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