Calculation of Cross Sectional Forces Representing a Handling Track

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1. Motivation

In order to estimate the life cycle of vehicle bodies Magna Steyr Engineering Graz uses rough road courses as well as handling tracks. The simulation of multi-body systems (MBS) is a very good method for determining the input values needed to estimate the life cycles by means of the FEM-method.
This paper describes how the cross sectional forces acting on the vehicle body can be determined by means of driving on a virtual handling track.

2. Modelling

For the simulation of cross sectional forces we usually apply forces and moments to the cut-free MBS-model deriving from wheel force transducers. Thus we receive the load histories needed for estimating the loads acting on the components [1]. A disadvantage of this method is the fact that the vehicle has to be stabilized in relation to ground. The consequence of this is that additional forces and torques (stabilization forces) are applied to the system in the area of stabilization. We usually stabilize the body. Low frequency excitations with big amplitudes, which occur on handling tracks, lead to enormous displacements thus causing a strong increase of the stabilization forces. As a result the life cycle estimations are of limited value.

We can solve this problem if we ride on a virtual road instead of stabilizing the vehicle. In this case we do not only need a vehicle model but also corresponding models for the driver, the tires and the road.

2.1. Vehicle model

The MBS-model has been designed in ADAMS/Car. Except for the stabilizer in the front axle, which is integrated as a flexible structure this model consists of rigid bodies.

The drive unit is elastically mounted to the body with ADAMS/standard bushings. The different parts of the drive train are linked with each other by means of rigid joints. The properties of the masses are taken into consideration but the drive train does not contain any torsional stiffnesses. The longitudinal displacements at the cardan shaft leading to the rear axle, has been modelled as a translational joint, which is combined with a friction element. This element prevents the translational displacement during the acceleration of the vehicle and remains open in the case of a braking maneuver, which is of special importance for a correct simulation of the forces acting on the engine and transmission mounts. In this case the vehicle body has been modelled as a rigid part, therefore equivalent stiffnesses resulting from a FEM-analysis have been applied to the damper domes. Without these equivalent stiffnesses the results of the cross sectional forces in this area would be much higher than at the real body.

The MBS-vehicle model (see illustration no. 1) has a total of 210 degrees of freedom.

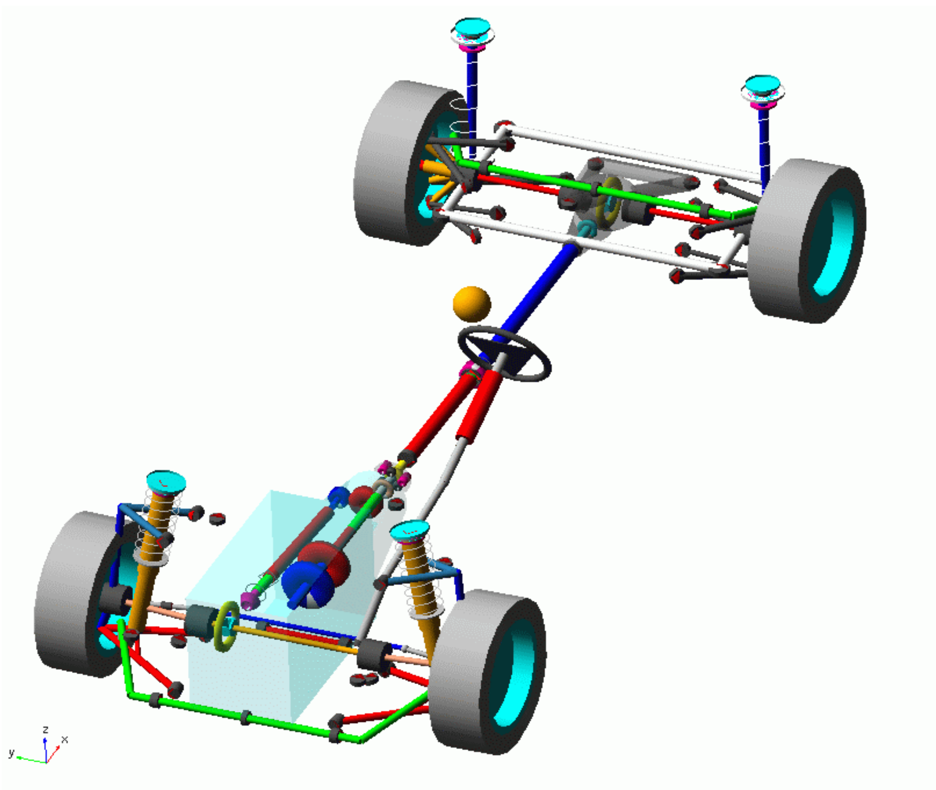

Illustration no. 1: Vehicle model

2.2 Driver model

In order to ride over a handling track the speed and steering-wheel angle histories have to be determined by means of measurements carried out on a similar vehicle. With the help of the SDI control device of ADAMS/Car the driving and braking torques can be regulated. As only torques of driving are significant for the loads acting on the components, the gear box is not modelled. The engine torque is applied to the transmission output and has to be adapted in order to achieve the measured speed and acceleration values. For braking maneuvers torques, which are supported by the wheel carriers and regulated by a control device, are applied to the wheels.

The steering wheel angle history, which is derived from measurements carried out on a similar vehicle, is applied to the model with a motion element. With this method the handling course is not followed in a precise way but this is not an essential condition for calculating the cross sectional forces. In order to determine the loads on the components it is, however, important to reach the required longitudinal and lateral accelerations.

2.3 Tire model

The tire model we use here has been designed by Magna Steyr Engineering Graz. It is a mathematical model based on the Magic Formula [2], [3]. The advantage of such a mathematical model is the fact that compared to physical tire models it takes considerably less computing time. The weak point of this tire model is the simulation of the non-stationary behaviour and the lift-off of the tire. In more recent projects we have therefore examined other tire models (RMOD-K, F-Tire).

2.4 Road model

For the simulation of the handling track we use an even surface because the vertical excitations are of minor importance. As the coefficient of friction between tires and road we have assumed the constant value $\mu = 1.2$. This value is estimated from measurements.

2.5 Verification of the model

The basic idea of the MBS department at Magna Steyr Engineering Graz is to use one vehicle model consisting of different subsystems for all MBS calculations. This makes it easier to update the model and ensures that the latest version of the model is available to all the members of the MBS team. As a consequence it becomes easier to compare and interpret the simulation results. Simulations of the subsystems (body, axles, drive train etc.) in all their different fields of application are being calculated, and therefore they are checked and tested. Thus we can guarantee a comprehensive verification of the model. By analysing the kinematic properties of the axle and the eigenfrequencies we gain first information about incorrect input data or possible insufficiencies of the model. By simulating certain types of static loads (pothole, braking, cornering) we gain information about the loads acting on the different components and about the cross sectional forces to be expected. Clearly defined driving maneuvers (e.g. steady-state cornering, lane change etc.) enable to check the driving behaviour of the MBS-model.

3. Simulation

The calculation was carried out in ADAMS, version 10.1, using the GSTIFF Solver. The stability of the integrator is significantly influenced by the friction element and its parameter. The SI2 Integrator offers certain advantages but it cannot be used in this case because the displacement-motion element is not supported. Using a Silicon Graphics Octane 2 workstation 117 seconds of CPU time are needed for 1 second of real time.

4. Results

In order to evaluate the simulation results in an efficient way we presently use Diadem, a commercial software, and FEMSITE, which is a product of our company. With a special routine, all the load channels and comparative values such as component acceleration values, wheel travel etc., which are needed for the evaluation, are read from the ADAMS output file. With the help of the evaluation software they can then be further processed. Each of the load channels is then evaluated on the basis of the damage equivalent load amplitude, which is a parameter for evaluating the life cycle of components.
The verification of the simulation is carried out in several steps.

4.1. Speed of the vehicle:

The first step in order to verify the calculation is to compare the measured speed histories to the calculated histories. Illustration no. 2 shows that the results correspond excellently with the measurements. In the simulation the maximum lateral acceleration of the vehicle is slightly smaller than the measured values (Ill. 3). This deviation is caused by the deficiencies of the tire property file.

4.2. Wheel force transducer loads:

For a realistic simulation of the forces and torques acting on the body it is important that the measured and calculated quantities are comparable. Such a comparison has been carried out with the help of wheel force transducer loads as can be seen in illustration no.4. The peak loads caused by low frequency excitations during the driving maneuver can be reproduced quite well. Deviations regarding the lateral forces are caused by the characteristics of the friction between tire and road ($\mu > 1.2$).

4.3. Verification of the cross sectional forces:

The next step in checking the plausibility of the results is to animate the movements of the vehicle. Measured values such as spring travel, acceleration of the wheel carrier forces acting on the steering tie rod etc. are compared with the calculated cross sectional forces. Furthermore the plausibility of the results is checked, by comparing the calculated cross sectional forces and the results of the static load calculations. Apart from statistical analyses of the damage-relevant quantities of the different load channels an analysis of the frequency spectrum is also an integral part of every plausibility check. It is included in the documentation of a cross sectional force calculation.

4.4. Cross sectional forces

For every point of the body where loads are applied a number of load channels are determined which correspond to the respective degrees of freedom. The low frequency parts of the handling track are reproduced very well. In the illustrations no. 5 the simulated forces acting on the engine mounts are compared with the measured values. Regarding the x- and z-directions the maximum values of the simulation reach the same level as the measured values. Due to the tire parameters used in this case the maximum lateral accelerations of the vehicle cannot be reached in the simulation. As a consequence of the differences in the lateral accelerations the maximum forces acting on the mounts in the y-direction are a little lower than the measured quantities.

5. Summary

The results achieved by simulating a handling track in order to determine the cross sectional forces acting on the components can be considered as satisfactory. It is an advantage that there are no components which due to their stabilization forces can either not at all or only in a limited way be evaluated with regard to their stability. These calculations are part of all our projects. They are carried out as a routine at Magna Steyr Engineering and contribute to evaluate the fatigue limit of vehicle bodies.

6. Outlook

At Magna Steyr Engineering Graz we use at present cross sectional force calculations determined by rough road and handling track simulations in order to evaluate the fatigue limit of bodies. The quality of the calculation results could be optimized if the cut-free vehicle model was replaced by the method of riding on a virtual rough road with an appropriate tire model. As a precondition we need, of course, an appropriate road model as well as a rough-road tire model, which generates realistic forces and torques. By using tire parameters, which reproduce the longitudinal and lateral forces more precisely we can also expect better results as regards the simulation of handling tracks. At present analyses of the tire models RMOD-K and F-Tire are being carried out. Our (digitalized) in-house rough road track serves as a road model [4].

7. Literature

[1] Hirschberg W.: Modellbildung und Simulation von Schlechtwegfahrten (Eröffnungssymposium Engineering Zentrum Graz 1998)

[2] Hirschberg W.: Tyre Forces Computation Module DTYRE (Proc. of 1st International Colloquium on Tyre Models for Vehicle Dynamics Analysis, Delft 1993)

[3] Winter G.: Vergleich fahrdynamischer Ergebnisse aus Rechnung und Messung (Diplomarbeit TU-Wien 2001)

[4] Fruhmann G., Reinalter W.: Synthetische Streckengenerierung zur virtuellen Lastkollektivermittlung (MKS-Simulation in der Automobilindustrie, Graz 2001)

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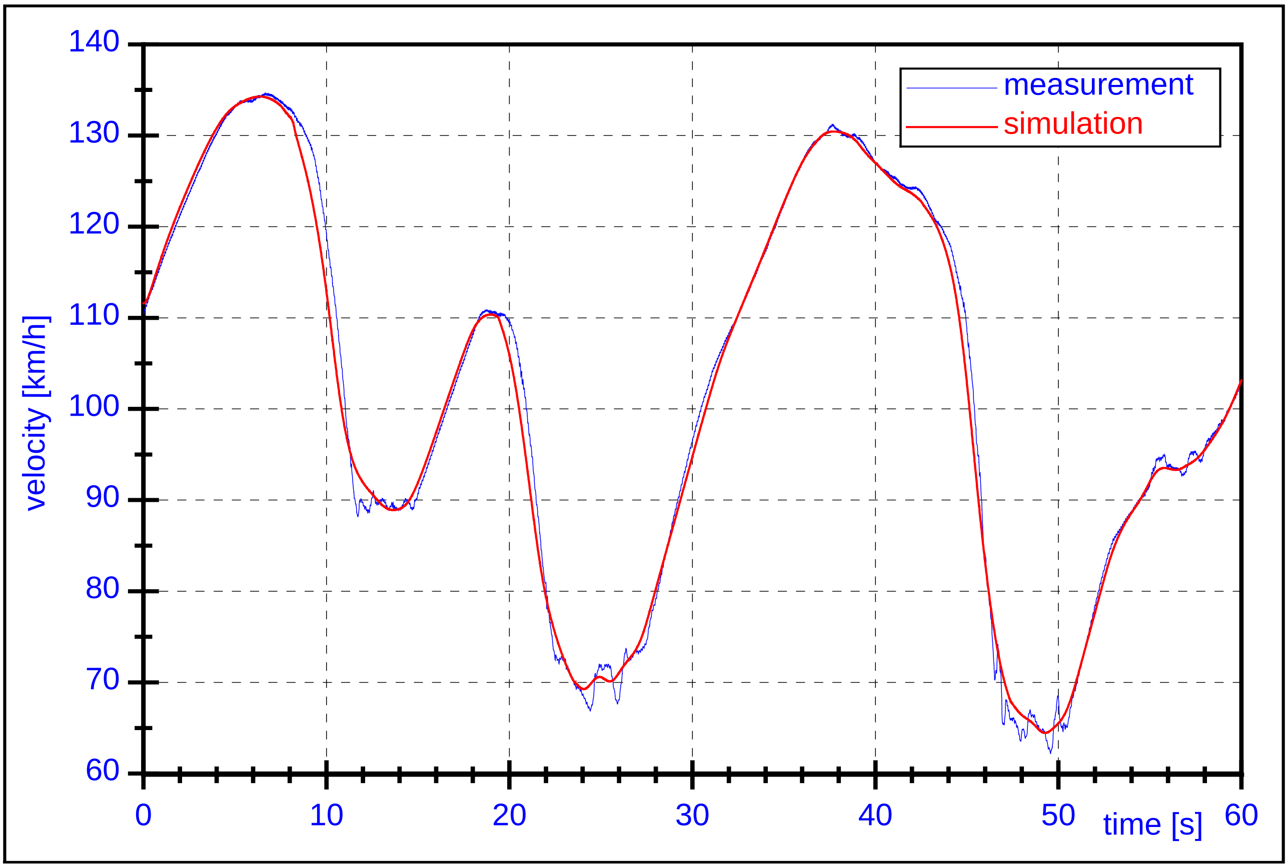

Illustration no. 2: Speed of the vehicle

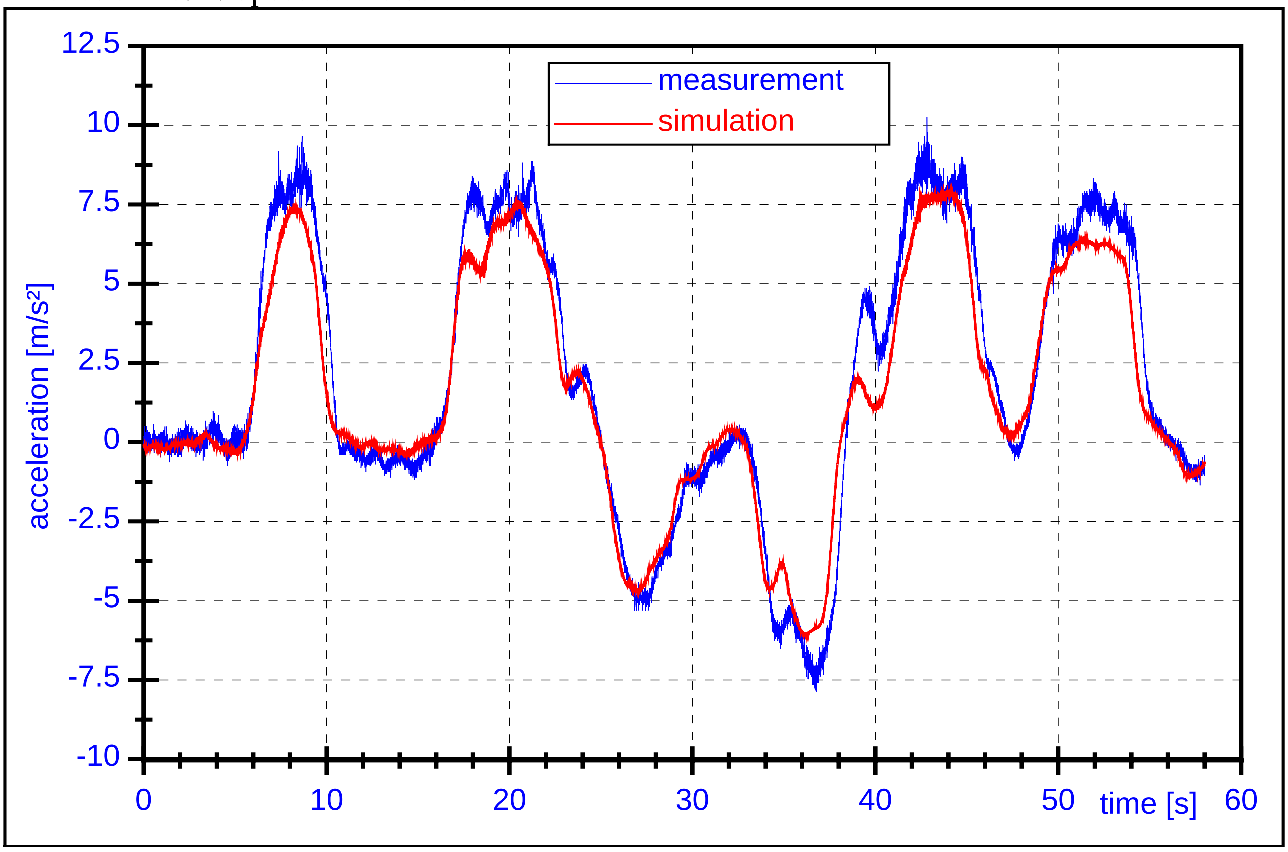

Illustration no. 3: Lateral acceleration of the vehicle

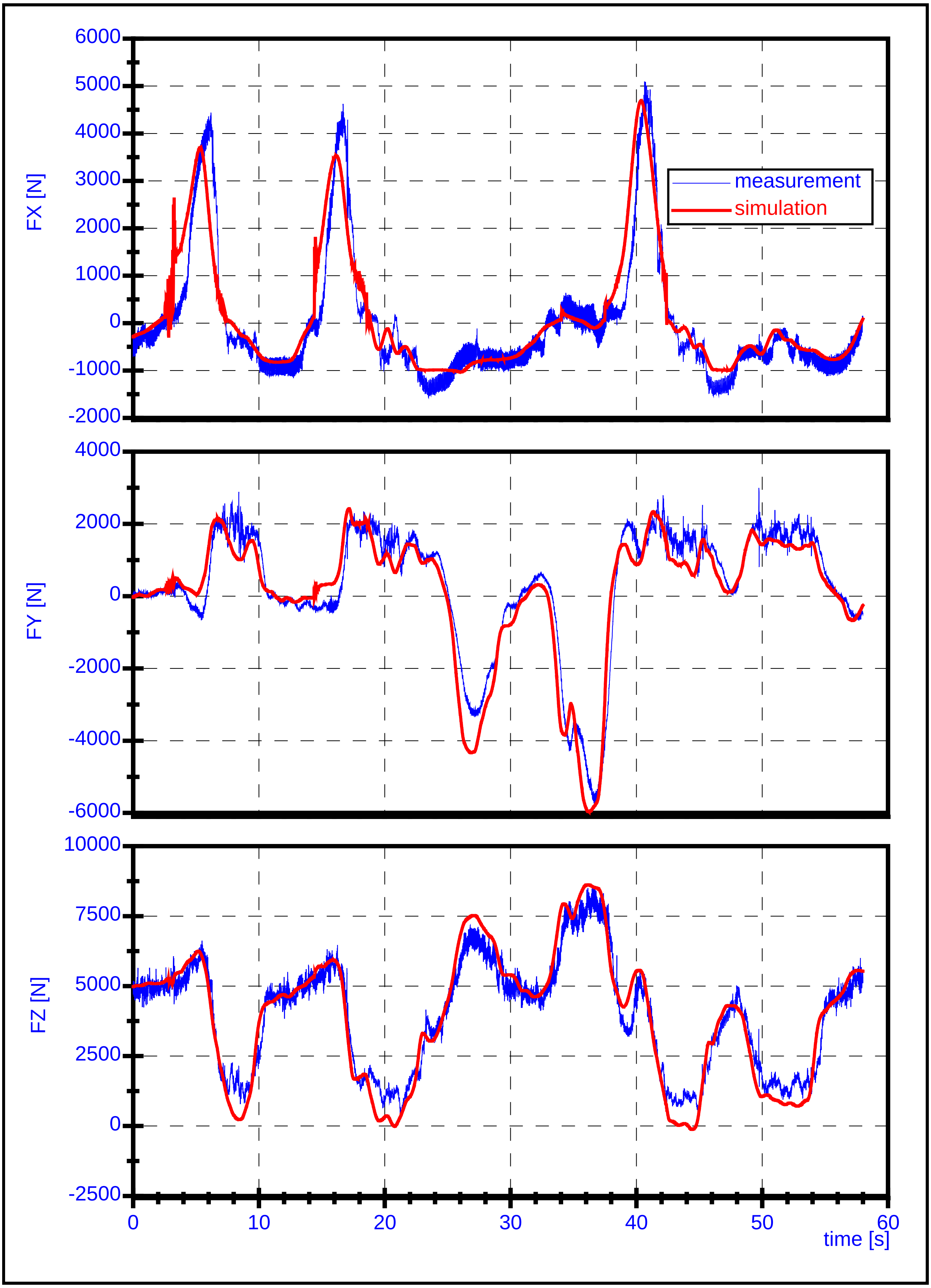

Illustration no. 4: Wheel force transducer loads (wheel front left)

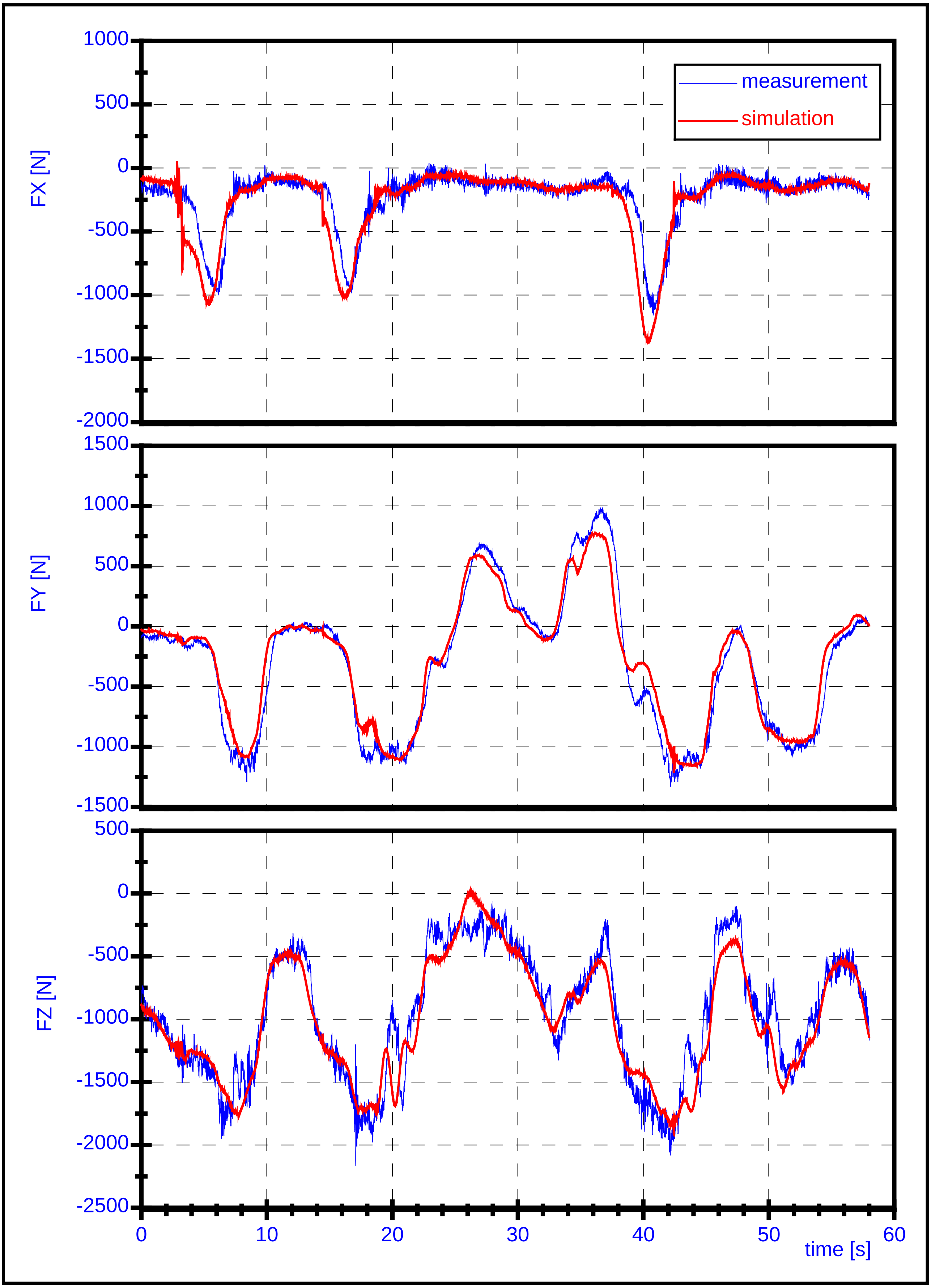

Illustration no. 5: Engine mount loads (wheel front left)