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Dynamic Loads in the Suspension of a Heavy Truck

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Abstract. *Twin rear axle suspensions of heavy trucks typically include assemblies called torque rods and Panhard rods to give the axles a defined position in longitudinal and transversal direction, respectively. The vertical forces are distributed to the rear axles by laminated leaf springs. Even in case of vertical loads and displacements of the axles there are horizontal displacements between leaf spring and axle due to the kinematics of suspension and consequently horizontal forces occur. These forces are loading the torque rods additionally to the external longitudinal tire forces e.g. from accelerating/braking or uneven roads. Using the Multibody Systems approach a model with 46 degrees of freedom was built and simulations of particular maneuvers like overriding a step or braking have been carried out. A high number of alternating stress cycles in the torque rods was found which could explain the failures by fatigue fracture.*

Keywords*: Twin axle, torque rod, Multibody dynamics, leaf springs, friction.*

Introduction

Horizontally oriented struts, so called torque rods hold the axles of heavy trucks in their longitudinal position. These rods are loaded by external forces acting at the wheels e.g. during braking, accelerating, tipping the cargo or driving on inclined roads or over obstacles like steps or potholes. But due to the frictional contact between leaf springs and axles in connection with the kinematics of the suspension there are reasonable forces in the torque rods caused by vertical displacements of the axles. This fact may be the reason for fatigue fractures that could not be explained by only extreme load collectives during life time like unloading tipper trucks or others. It is more likely that these failures are due to dynamic loads under normal travel on normal roads. To study these effects a truck with a twin rear axle was modeled by the methods of Multibody Systems with a total number of 46 degrees of freedom using the program system NEWEUL.

A particular attention was drawn to the modeling of the friction between the laminated leaf springs and the rear axles of a heavy truck like MAN 40.603 or Volvo F16 with an unloaden weight of 13 500 *kg* and a maximal payload of 26 000 *kg*. The dynamic loads in the suspension were studied by simulating several maneuvers like driving over a normal road, driving over steps of different heights with different speeds and braking. Depending on the actual load and the friction between the leaf springs and axles, a high number of alternating stress cycles in the elements of the suspension were found in the simulation results, thus the reason for the failures could be explained.

2. Nomenclature

A = area, m^2
b = width of axles, m
E = Young's modulus, N/ m^2
F = force, N
f = degrees of freedom
H = contact point
h = height, m
I = moment of inertia, m^4
i, j = integer variable
m = mass, kg
p = number of rigid bodies
q =holonomic constraints
s = arc length, m
t = time, s
v = velocity, m/s
w = deflection of beam, m

Subscripts
1 axle 1 hits the step
2 axle 2 hits the step
a axle
c characteristic
c chassis
e equivalent
H contact point
max maximal
n normal
pl payload
r rocker
rel relative
t tangential

Greek Symbols
α = cardan angle, x-axis, rad
ß = cardan angle, y-axis, rad
γ = cardan angle, z-axis, rad
μ = coefficient of friction
σ = smoothing parameter
f = frequency, Hz

3. Modeling

The rear part of the investigated truck is shown in figure 1 with the chassis and the two rear axles, which are coupled to the chassis by two laminated leaf springs on the left and right side. The dampers are not shown. Each laminated leaf spring is mounted on a rocker, which is connected to the chassis by a revolute joint.

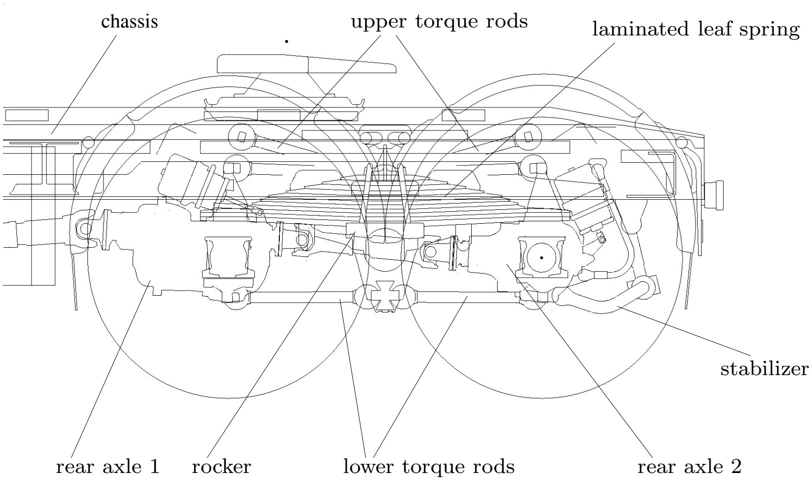

Figure 1. Twin rear axle of heavy truck.

To give the rear axles a defined longitudinal position three torque rods are used per axle. The upper one is centered and the lower ones are placed laterally below the laminated leaf springs. The torque rod itself is a cylindrical tube with spherical bushing elements at each side in the form of bolts embedded by rubber within nuts. Thus the loads are primarily longitudinal. Additionally, a cabin and the front axle are connected to the chassis by springs and dampers. To reduce the roll motion of the chassis a stabilizer is installed.

Since for the investigation of the longitudinal forces in the torque rods the overall motion is of particular interest, effects from structural dynamics are neglected. Thus the bodies are considered as rigid. Hence, the system is modeled as a Multibody System with $p = 13$ bodies, cabin, chassis, three axles, two rockers and 6 torque rods.

With respect to the considered driving maneuvers the description of the lateral motion is not needed. Therefore, the chassis has a prescribed motion in longitudinal direction superposed by a vertical displacement and its roll and pitch angles. For the cabin only a vertical motion is considered. For the front axle a vertical displacement and its roll angle is taken into account, whereas the rear axles are described by a longitudinal and vertical displacements, as well as their roll, pitch and yaw angles around the longitudinal, transversal and vertical axis, respectively. Each rocker has a pitch motion as base of the laminated leaf spring. For the torque rods all motions except the rotation about their longitudinal axis are allowed. The entire system thus has $f = 46$ degrees of freedom.

3.1. Leaf springs

A laminated leaf spring is assembled by several leaf spring elements. Between these elements Coulomb's friction occurs, which results in unsteady characteristics for compliance and damping. The area enclosed by the hysteresis curve can be interpreted as dissipation caused by inner damping and thus it is modeled as a damper element.
In the contact between the laminated leaf springs and the axles friction occurs, too. Due to high normal forces high friction forces may result.

For applying Coulomb's friction law phases of sticking and sliding have to be distinguished. In the latter case the relative motion between the leaf spring and the rear axle has to be calculated.

The deformation of the leaf spring as shown in figure 2 is estimated by applying classical first order beam theory of Bernoulli-Euler. With and y_H denoting the coordinates of the contact point H in the coordinate system of the rocker it follows

$$w(x^r, F_y^r) = \frac{F_y^r}{2EI}\,(x^r)^2\left(x_H - \frac{x^r}{3}\right) \tag{1}$$

and with $w(x_H) = y_H$ the deformation is given by

$$w(x^r) = \frac{3y_H}{2x_H^3}\,(x^r)^2\left(x_H - \frac{x^r}{3}\right) \quad . \tag{2}$$

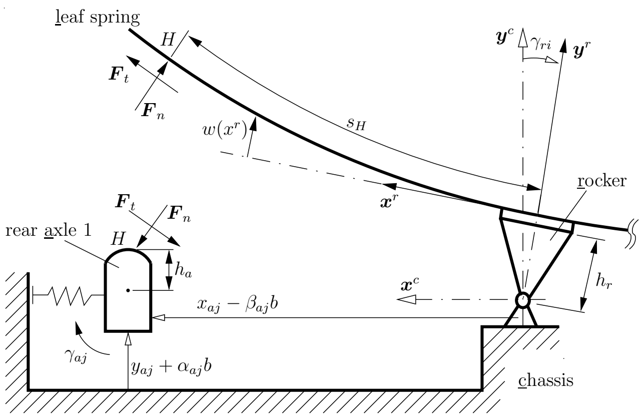

Figure 2. Leaf spring with rocker and rear axle 1 in the deformed configuration.

To avoid numerical problems a smoothened Coulomb's friction law as proposed by Song et al. (2001) is used. The friction force F_t then reads

$$F_t = -\mu \tanh\left(\frac{v_{rel}}{\sigma}\right) F_n \quad . \tag{3}$$

To determinate the movement of the contact point H along the leaf spring, additionally to the rigid body kinematics of rocker and axle the elastic deformation of the spring is taken into account. Therefore the arc length s_H of the deformed beam is calculated by

$$s_H\left(x_H, y_H\right) = \int_0^{x_H} \sqrt{1 + \left(\frac{dw}{dx^r}\right)^2} dx^r \tag{4}$$

and the relative motion can be derived as

$$v_{rel} = \dot{s}_H = \frac{\partial s_H}{\partial x_H}\dot{x}_H + \frac{\partial s_H}{\partial y_H}\dot{y}_H \quad . \tag{5}$$

Finally the force law between the leaf spring and the axle can be written as

$$\boldsymbol{F} = \begin{bmatrix} F_n + F_t\gamma \\ -F_n\gamma + F_t \\ 0 \end{bmatrix} \quad \text{, with} \quad \gamma \approx \frac{dw}{dx^r} \quad . \tag{6}$$

The description for the friction force F_t in Eq. (3) is very useful for dynamical investigations and vibration analysis. Its application to static or quasistatic considerations with stick effects is severely limited, and the parameter σ has to be chosen carefully with respect to the expected velocities of sliding $v_{rel,max}$ in the coupling region. After simulation a verification of the velocities assumed before has to be made.

3.2. Tire model

For the investigation of loads in the suspension rather global vertical and longitudinal tire forces than detailed forces in the contact patch are of interest. Although very precise finite element tire models exist, in order to avoid long computation time a heuristic tire model as proposed by Sui and Hirshey (2001) is used. It is based on radial spring elements continuously distributed along the circumference and allows an approximate description of uneven road profiles as well as enveloping effects like overriding obstacles.

For an assumed proximation of tire to an arbitrary road surface a characteristic area A_c as overlap of undeformed tire and the arbitrary road profile under consideration is defined. Further, an equivalent area A_e as the overlap of the undeformed tire and an even road surface is considered where A_e is equal to A_c. Then, with a known stiffness of the tire, the radial force belonging to the equivalent area is calculated. This radial tire force is assumed to be acting from the center of the considered characteristic area A_c to the center of the wheel.

4. Simulations

Having the simplified model of the considered truck, the belonging mathematical equations have to be formulated. For a holonomic Multibody System with $f = 6\,p - q$ degrees of freedom different kinds of kinematical descriptions are applicable, f generalized coordinates regarding the q holonomic constraints or 6 unconstrained coordinates for each of the p bodies. By applying mechanical principles like Newton´s and Euler´s law, in the first case the governing equations are a set of f ordinary differential equations (ODE) and in the second case there is a set of 6 p differential equations combined with q algebraic constraint equations (DAE). Generally, this descriptor form has an index higher than 1 which has to be reduced by differentiating the constraint equations before numerical time integration is applied (Eich-Soellner and Führer, 1998) and (von Schwerin, 1999). To generate the equations of motion several program packages are available, here the package NEWEUL (Kreuzer and Leister, 1997) was applied and the equations are formulated using generalized coordinates (minimalform) in a symbolical representation (Schiehlen, 1990, 1993). The specific program modules for leaf spring coupling and tires can also be used with other program packages like SIMPACK, ADAMS or others by adapting the modules to the particular program interfaces.

The dynamical behavior can be studied by linearizing the equations and applying the eigenvalues analysis or by numerical time integration of linear or nonlinear equations of motion. The lower natural frequencies of the considered truck with their corresponding damping and dominant motion of the modes are listed in table 1. Here, the linearized model does not contain nonlinear effects like energy dissipation and sticking by friction between leaf springs and axles.

Table 1. Linearized system: Natural frequencies, modal damping and characteristic motion of modes.

frequency [Hz]	modal damping	characteristic motion of mode
0.66	0.36	cabin vertical
1.44	0.56	chassis pitch
1.88	0.43	chassis vertical
9.56	0.40	rear axles vertical inphase
10.39	0.48	front axle
10.60	0.46	rear axles vertical counterphase
20.30	0.04	rear axles horizontal inphase
20.37	0.05	rear axles horizontal counterphase
20.62	0.02	rocker pitch
32.20	0.08	rear axles pitch inphase
32.29	0.08	rear axles pitch counterphase

For time integration of the nonlinear system the maneuver "overriding a step" was chosen and the following considerations are related to this case.

Figure 3 shows the vertical displacements of cabin, chassis and the axles, respectively for the case of fully loaded truck, a driving speed of v = 22 *m/s*, a cargo of m_{pl} = 26 000 *kg*, a step heigth of h = 2 *cm* and a coefficient of friction of μ = 0.1.

After overriding the step with the front axle at time t = 0 s, clearly the lift of the chassis can be seen, the cabin follows with a damped vibration of low frequency of about 0.7 *Hz*. Both rear axles are slightly excited by the chassis and obviously they are hitting the step after a time of approximately t_1 = 170 *ms* and t_2= 228 *ms*, respectively.

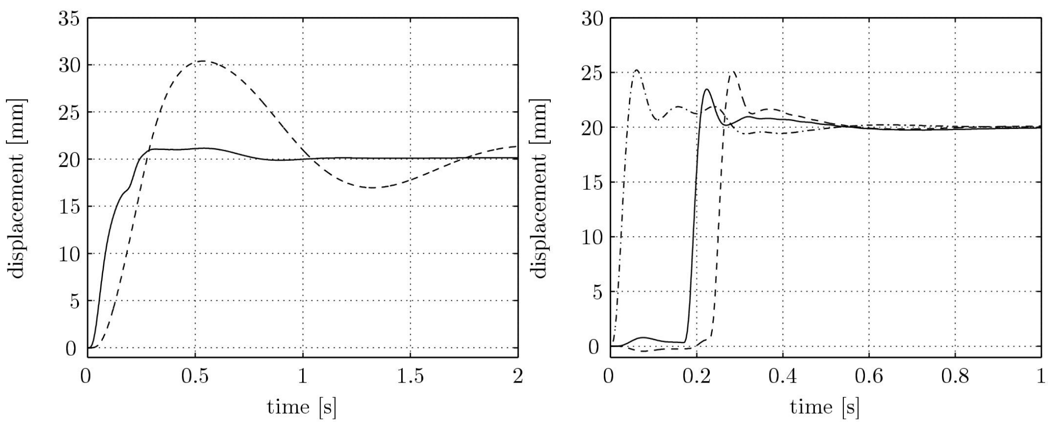

Figure 3. Left: vertical displacements of chassis (solid) and cabin (dashed). Right: vertical displacements of front axle (dashed-dotted), rear axle 1 (solid) and rear axle 2 (dashed). Maneuver: Overriding a step of height h = 2 *cm*, velocity v = 22 *m/s,* cargo m_{pl} = 26 000 *kg,* coefficient of friction μ = 0.1

For the same maneuver the longitudinal forces in the torque rods are plotted in figure 4. At the first instance after hitting the step, the rod of axle 1 is compressed and that of axle 2 is extended. Again the time delay of Δt = 58 *ms* between axle 1 (solid line) and the axle 2 (dashed line) can be seen.

Before and as well as after hitting the step the forces are on a level different from zero indicating an inner prestress due to sticking effects in the frictional coupling between leaf spring and axles. Moreover, there is a coupling of both axles by the rocking leaf springs which transfers the excitation from axle 1 to axle 2 within the time interval $t_1 < t < t_2$. As expected, the amplitude of forces is strongly dependent on the height of step.

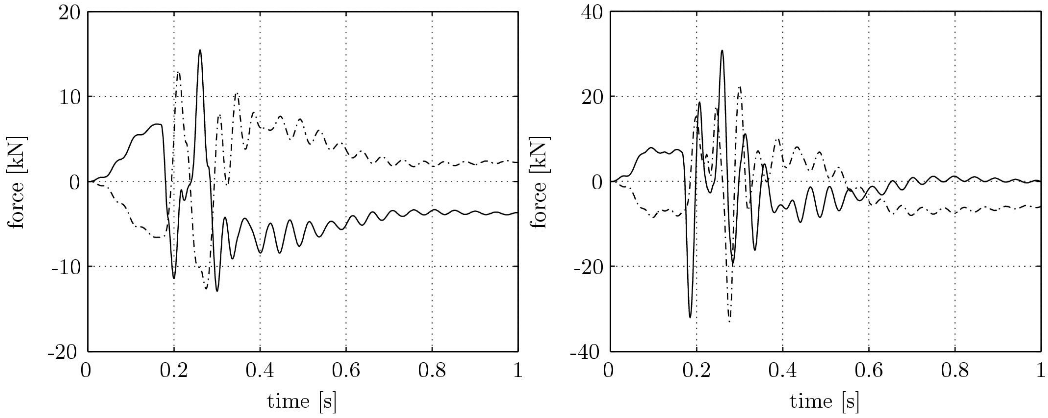

Figure 4. Longitudinal torque rod forces for overriding a step of different heights, h = 2 *cm* (left) and h = 5 *cm* (right). Velocity v = 22 *m/s,* cargo m_{pl} = 26 000 *kg,* coefficient of friction μ = 0.1

An increased payload of the truck increases the force amplitudes in the torque rods as shown in figure 5 for the rear axle 1. Due to the coupling of both axles by the rocking leaf springs, the time instant when the second axle is hitting the step can be seen at axle 1, too.

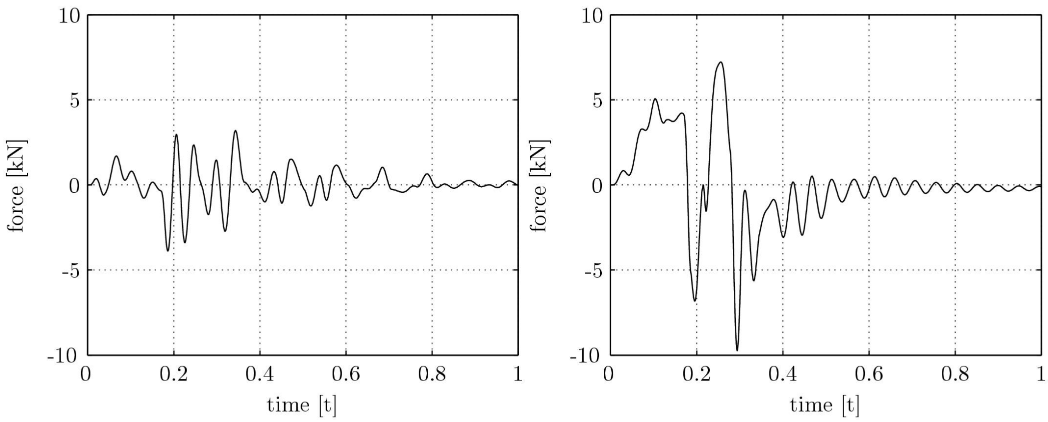

Figure 5. Longitudinal torque rod force for overriding a step with different payloads m_{pl} = 0 *kg* (left) and m_{pl} = 13 000 *kg* (right). Height h = 2 *cm*, velocity v = 22 *m/s,* coefficient of friction μ = 0.1

A higher cargo leads to higher normal and friction forces between leaf spring and axle causing an increased energy dissipation in form of unsteady damping of the longitudinal vibrations of the axles. On the other hand, with a higher friction and more frequent sticking contact phases during vibrations the leaf spring acts more as a kinematical constraint element causing reasonable forces in the torque rods even for pure vertical loads or vertical motions of the axles. Consequently, a high number of alternating load cycles occur which may cause fatigue fraction of the torque rods.

This damping effect with sliding and sticking phases can also be seen by varying the coefficient of friction μ between leaf springs and axles, as illustrated in figure 6.

Looking on the frequency decomposition by means of Fast Fourier Transform, for a low coefficient of friction the influence of vertical vibrations of cabin, chassis and axles can be seen in figure 7 at 0.7 *Hz*, 3.5 *Hz*, 6 *Hz*, 9.5 *Hz*, 13.7 *Hz*, 17.8 *Hz*, 21.1 *Hz*, respectively. This is due to the coupling of vertical motions with horizontal forces in the torque rods. In comparison to the linearized model without effects of friction the natural frequencies are now reasonably lower. Further, for a higher coefficient of friction the spectral components of the modes connected with the vertical motions in the suspension between 4 *Hz* and 10 *Hz* are significantly pronounced. These facts indicate that the dynamical behavior of the suspension is dominated by the damping effect of friction. The natural frequencies found in the spectra are strongly dependent on the specific situation in the contact area. There is an additional kinematical constraint during sticking, and contrary to that, there is no kinematical constraint but high energy dissipation during sliding.

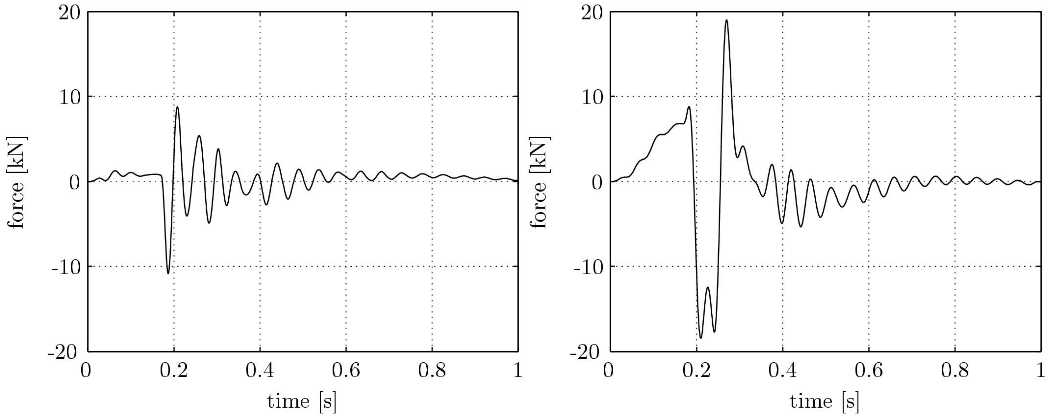

Figure 6. Longitudinal torque rod force for overriding a step with different coefficients of friction $\mu = 0.01$ (left) and $\mu = 0.4$ (right). Height $h = 2$ *cm*, velocity $v = 22$ *m/s,* cargo $m_{pl} = 26\,000$ *kg*

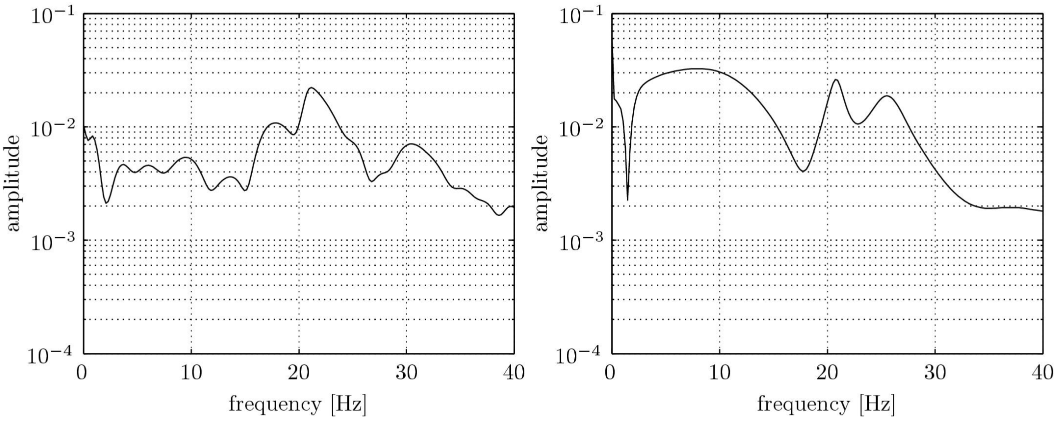

Figure 7. Spectral decomposition of torque rod forces shown in figure 6

In figure 8 the variation of driving speed is shown. Clearly the different levels of forces due to sticking can be seen. Again, at t_1 there is a compression of the rod of axle 1 and at t_2 there is an opposite excitation of the same rod with a similar intensity caused by the second axle. The loads of torque rods for increasing driving speed show only a slight increase at the first instance of excitation. Obviously, the compliance of tire and the inertia of axle are reducing the horizontal impulse and the observed excitation is due to vertical motions of axles.

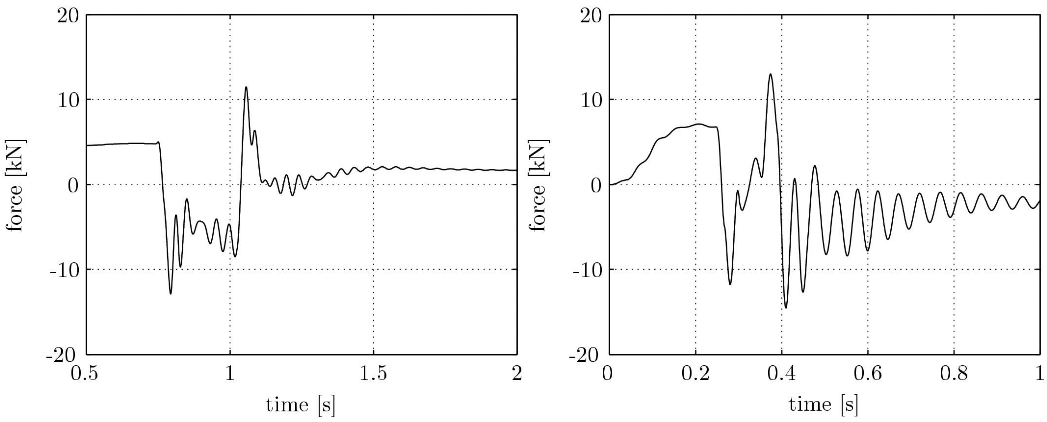

Figure 8. Longitudinal torque rod force for overriding a step with different speeds $v = 5$ *m/s* (left) and $v = 15$ *m/s* (right). Height $h = 2$ *cm,* cargo $m_{pl} = 26\,000$ *kg,* coefficient of friction $\mu = 0.1$

Tire forces are acting at the contact point to the road and the forces in the lower plane of torque rods are higher than those in the upper plane. For that reason there are two rods in the lower plane and one in the upper. Braking is another maneuver to illustrate the vibrations induced in the rear suspension, figure 9. A sudden braking force is applied on all wheels and additionally to the braking force alternating forces from vertical motions of truck components and longitudinal vibrations in the bushing of the torque rods can be seen.

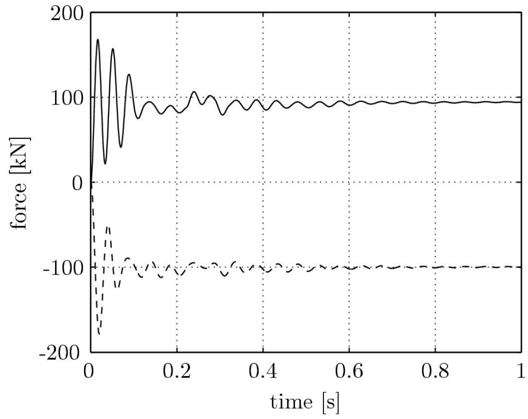

Figure 9. Longitudinal forces in the upper torque rod (solid) and lower torque rod (dashed) due to full braking

5. Conclusion

Fractures in the rear axle suspension of heavy trucks could not be explained by classical load collectives with its static forces and estimated dynamic loads resulting from longitudinal tire forces.

Modeling the truck as a Multibody System allows to regard effects of friction in the system. Such models are capable to investigate dynamic loads in suspension of trucks more precisely and allow to study the influence of road profile, cargo mass, driving speed characteristics of tires and the design of suspension itself. It could be shown that friction between laminated leaf spring and axles cause loads in the longitudinal torque rods even for vertical excitations. These horizontal friction forces increase the damping of the suspension system, but due to the resulting high number of alternating load cycles additional stress is induced which may explain the occurrence of fatigue fracture.

For the determination of loads in the elements of suspension, friction has a central influence and brings reasonable nonlinearities into account and the results are very sensitive with respect to the coefficient friction. The application of friction laws smoothened by introduction of specific functions facilitates the time integration of the nonlinear systems equations. But the parameters of the smoothing functions have to be chosen very carefully to get a realistic approximation of sticking and sliding phases

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