A New Massless Leaf Spring Model for Full Commercial Vehicle Simulations

Leaf springs are widely used suspension components for heavy commercial vehicles because of their robustness and low cost. A leaf spring has two functions: suspension of the vehicle and support of the axle. The deformation of the leaf spring has an influence on the axle kinematics, and therefore, it affects the dynamic behavior of the vehicle.

Model building in an early development phase is complicated because of the lack of parameters for a multi-body system (MBS) model. Most commercial MBS software packages prefer a model approach with elastic beams whose data sets are usually not known in the early phase of vehicle development. In this article, a new leaf spring model is described which offers a capable leaf spring description in both the concept and fine tuning phase.

STATE OF THE ART

In the commercial vehicle industry, there is no documented standard for modeling leaf springs. This means that the modeling process is fault-prone as well as uncertain and the simulations produce hard-to-compare results. The quality of the model depends mainly on the experience of the user, on the applied methodology and, in some cases, on the availability of measurement data.

Because of the large influence on axle kinematics [1], it is important to consider the main deformations of a leaf spring in an early development phase. During operation, the vertical deflection will be excited due to track irregularities and, in the case of a braking process, the so-called S-Shape occurs.

In the analysis of the modal behavior of an FE-model, the first two eigenmodes are equivalent to these two deformations (Fig. 1).

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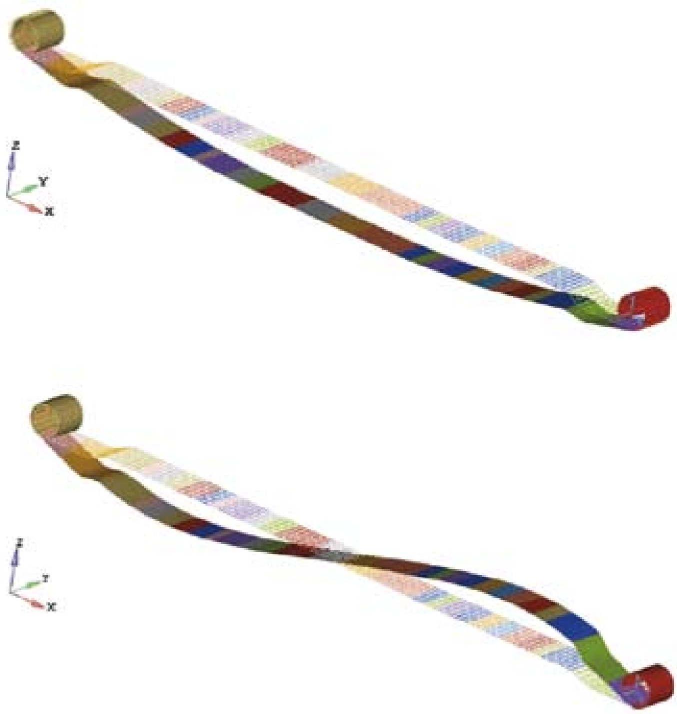

Fig. 1: First (above) and second (bottom) bending mode of a leaf spring Finite Element (FE) model

These functionalities can be described with a very detailed leaf spring model. The main problem is that this detailed modeling is not required in all of the modeling stages. This means that in the concept phase, the geometrical data are not available to build up an elastic body model. The only available value is the required vertical stiffness. To use this vertical stiffness, a simple spring secures high computational efficiency, but cannot describe the geometrical effects from the real leaf spring deformation which can be critical for the dynamic behavior of the vehicle. On the standard model level, the model can be expected to describe the important physical effects (contact with the S-Bump and geometric coupling vertical-longitudinal). One stage higher — the complex model — the dynamic effects, twisting and the description of other parts (e.g., bearing eye) must also be considered. The higher the model complexity, the more time is required for both the modeling and the simulation. Another disadvantage is that these different stages require different models. Switching between them is not possible, and a completely new model is necessary.

In this paper, a new standard modeling approach will be described — the so-called "massless-model". After the theoretical background, the description of the implementation of the massless-model as a SIMPACK User Routine will be presented. The massless-model has been validated with test-rig measurements at MAN Truck & Bus AG in Munich, and some comparison simulations have been carried out using a modal description of the leaf spring as a reference in SIMPACK.

MODELING CONCEPT

The proposed model approach represents one half of a leaf spring with a limited number of three-dimensional massless beam elements. As the name indicates, it neglects the mass and therefore the dynamic effects. The following two important effects are considered within the model: linear beam static deformation and nonlinear geometrical coupling between rotation and displacement — the so-called shortening effect. With the help of the Hermite polyno-

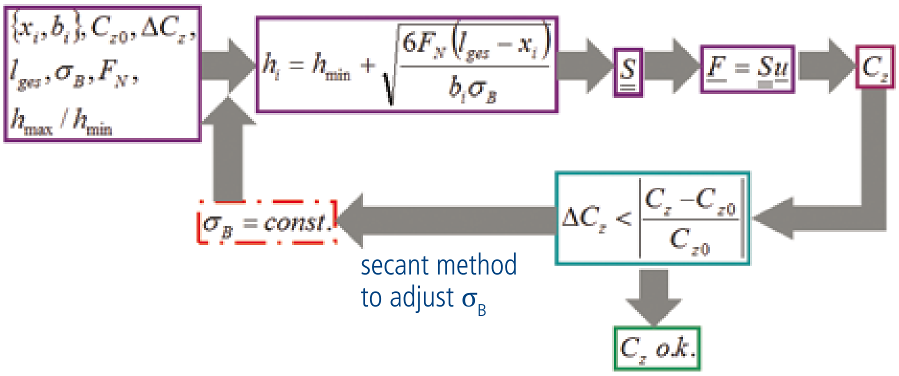

Fig. 2: Determination of the vertical stiffness (Cz) in case of a stiffness based parameterization

mials, the terms of the stiffness matrix and the effective length change of the spring have been determined [3]. The stiffness matrix is constant. It must be determined before the calculation. The internal node positions and the forces at the end nodes are calculated using a static reduction method (Guyan). To consider contact with the S-Bump, a non-equidistant node distribution is also easy to realize.

Two different parameterization methods are offered. The first one is suitable to model a leaf spring with known geometry; in this case, the width and height characteristics are given. Alternatively, a requested vertical

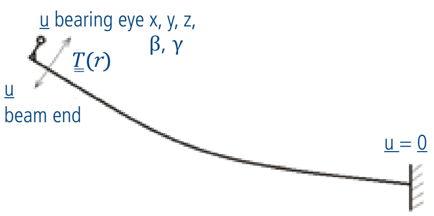

Fig. 3: Mechanical model of one half of a leaf spring

stiffness and a width characteristic are given by the user and a parabolic thickness profile is determined at the initialization process.

FUNCTIONALITY OF THE MASSLESS-MODEL

The first step is to determine the stiffness matrix. The user defines the number of beam elements (up to 10) and the segmentation process begins. If the leaf spring has an S-Bump, a node must be placed coincident with the marker of the S-Bump. The solver finds the segment in which the S-Bump lies and divides the beam accordingly. When the lengths of the elements are fixed, the next step is to determine the heights and widths. There are two parameterization strategies: the first one based on a given geometry, a given width and height characteristic; the second one goes out from a required vertical stiffness. In the case of the geometry based parameterization, the characteristic of the height and width are given as input functions. These must be interpolated according to the coordinates along the longitudinal beam axle. If the leaf spring geometry is not given, which is usually the case in the concept phase, the process to determine the element geometry is more complex. In this case the only known parameters are the required vertical stiffness (C_{z0}) and the width characteristic (b_i). The assumption is that the bending stress is constant along the cantilever beam which corresponds to a parabolic leaf spring. With the help of the maximal bending stress (σ_B) and vertical load (F_n), the height for each element (h_i) can be determined (Fig. 2). To avoid the zero height at the leaf spring eye, the user defines the ratio of the maximal to minimal height which is usually a constant value in the commercial vehicle industry.

"The mechanical model of the leaf spring is a cantilever beam."

The mechanical model of the leaf spring is a cantilever beam. The generalized coordinates lay at the end of the beam, by the axle, and by the leaf spring eye. Considering half of a leaf spring, the end by the axle is modeled as a clamped end while the end by the leaf spring eye has five degrees of freedom (Fig. 3): three translational and the rotation about the lateral (y) and the vertical (z) axis (β and γ respectively). Because the clamped end means zero displacements and rotations the system has only five generalized coordinates. Considering the two rotations, moment free, the only given coordinates are the displacements and the torsional rotation at the leaf spring eye — which come from the From Marker in SIMPACK. To determine the other two rotations, the displacements and rotations of the inner nodes, a Guyan reduction has been applied. The stiffness matrix must be partitioned according to the master nodes (x, y, z, α → $\underline{u}_m$), and the displacements of the inner nodes plus the two rotation angles can be determined as slaves ($\underline{u}_s$):

$$\underline{\underline{S}}\,\underline{u} = \underline{F}$$

$$\begin{bmatrix} \underline{\underline{S}}_{mm} & \underline{\underline{S}}_{ms} \\ \underline{\underline{S}}_{sm} & \underline{\underline{S}}_{ss} \end{bmatrix} \begin{bmatrix} \underline{u}_m \\ \underline{u}_s \end{bmatrix} = \begin{bmatrix} \underline{F}_m \\ \underline{0} \end{bmatrix}$$

$$\underline{\underline{S}}_{sm}\,\underline{u}_m + \underline{\underline{S}}_{ss}\,\underline{u}_s = \underline{0}$$

$$\Rightarrow \underline{u}_s = -\underline{\underline{S}}_{ss}^{-1}\,\underline{\underline{S}}_{sm}\,\underline{u}_m$$

In the case of the geometry based parameterization, the model is ready for the simulation. The stiffness matrix is determined. If the model is parameterized on the basis of the vertical stiffness, a further investigation is needed because the determined structure does not have the same stiffness as the input value. If it is within the given tolerance range, it will be accepted. If not, the bending stress will be adjusted according to the second method (Fig. 2), and the whole process will be carried out to determine a new height characteristic until the vertical stiffness meets the tolerance conditions. If the positions are known, the forces and mo-

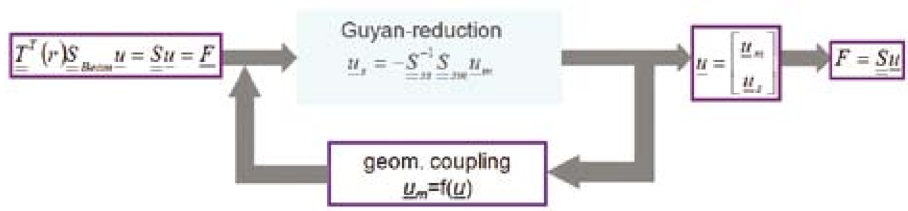

Fig. 4: Modeling approach to describe the static beam deformations

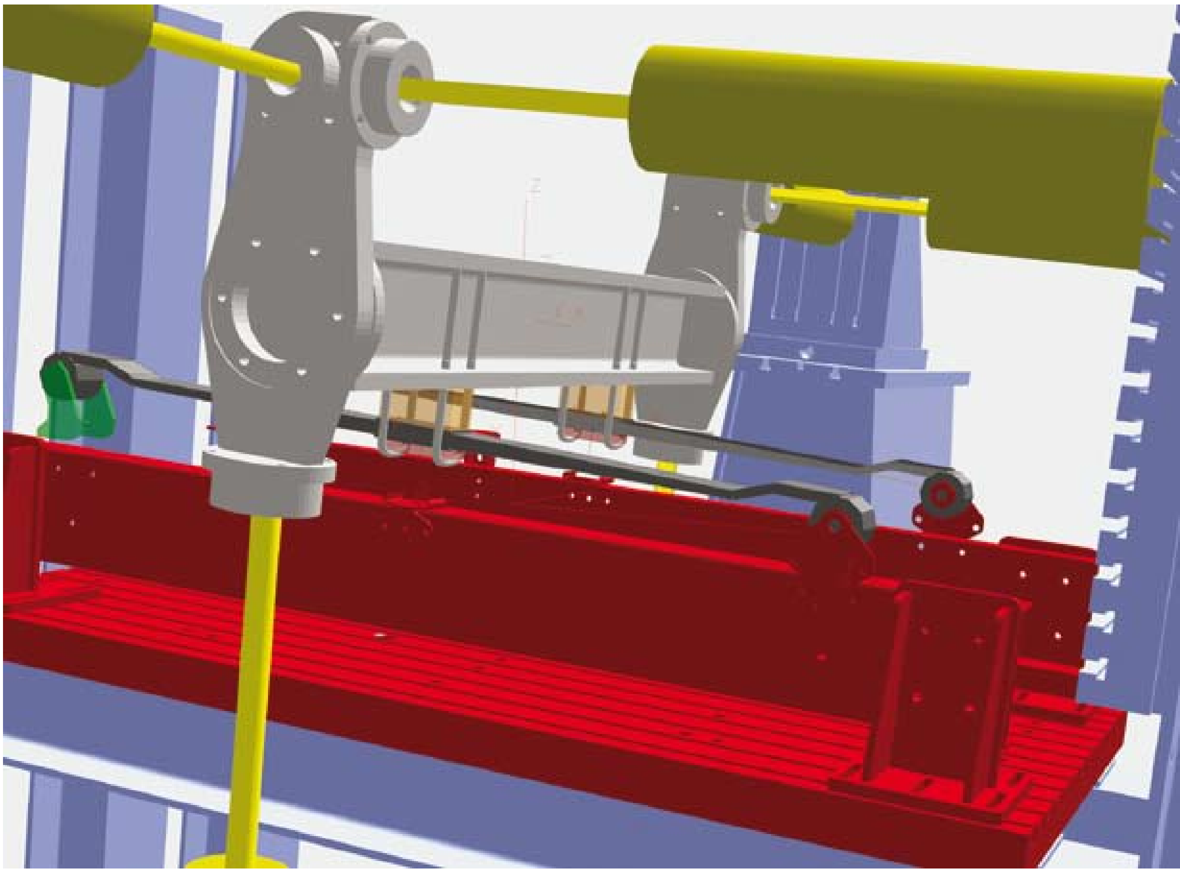

Fig. 5: Virtual test rig in SIMPACK (MAN Truck & Bus AG, Virtual Vehicle)

ments can be determined with help of the linear beam statics. The radius of the leaf spring eye is also handled internally in the massless-model (Fig. 3). The displacements by the leaf spring eye must be transformed to the end of the beam. Not only the geometric coupling-radius, but the radial stiffness of the leaf spring eye is considered. After the modification of the stiffness matrix ($\underline{T}(\underline{r})$) a Guyan reduction is carried out to determine the deformed shape of the leaf spring. It is possible to calculate the shortening from these displacements and rotations but, this step affects the master positions. With the help of a second Guyan reduction, the end state of the deformed leaf spring is determined (Fig. 4).

If an S-bump also exists, a contact must be monitored. In the case of a contact with an S-Bump, the contact force will be calculated with the help of the given S-Bump stiffness — which can be in a form of a constant value or of a stiffness chart — and of the calculated penetration. The stiffness will be interpolated according to the penetration, and this value affects the corresponding terms of the stiffness matrix. The deformed shape is iteratively determined with the Newton-Raphson algorithm within a given required tolerance.

If the deformed shape is known ($\underline{u}$), the moments and forces ($\underline{F}$) can be calculated from the well-known static equation system. Because the deformed shape has been determined from a static equation, the model cannot describe dynamic effects.

No. of test phase	Pre-set for vertical displacement (z)	Pre-set for brake force	Pre-set for lateral force
K1	deflection (+/-)	no	no
K2	deflection (+)	yes	no
K3	0	yes	no
K4	deflection (-)	yes	no
K5	deflection (+/-)	yes (const.)	no
K6	deflection (+)	yes	no
K7	deflection (+/-)	no	yes
K8	0	no	yes

Table 1: Description of the preset forces and displacements during the test rig measurements (MAN Truck & Bus AG)

IMPLEMENTATION IN SIMPACK

The massless-model is implemented in SIMPACK as a User Routine, a force element. The buildup of the leaf spring model is carried out before the simulation process (Task = 2). The stiffness matrix, the inverse of the slave part of the stiffness matrix, the mass, and vertical stiffness are determined. These are saved in the memory, and are imported in every integration step. From the displacements and rotations, a force is determined.

The massless-model needs different input data sets for the different parameterization strategies (geometry- and stiffness-based). To define and to simplify the modeling process, a leaf spring generator has been developed. This is an ASCII-file steered tool which creates the required models in much less time. With the help of this tool, the changes of the model and the transition between the parameterization strategies are more effective and simplified. The leaf spring generator is also able to build an elastic beam model as well as the two massless variants from a quite similar data set.

VALIDATION OF THE MASSLESS-MODEL

The applied mechanical model has been validated via test-rig measurements at the MAN Truck & Bus AG in Munich. During the tests, different load cases (Table 1) have been defined to map of all the possible load cases during operation of the heavy vehicle.

The load is mainly applied in the vertical direction, but a leaf spring must also guide the axle along the direction of motion. The longitudinal displacement due to the shortening effect has a big influence on the steering behavior of the vehicle. Twisting can occur in cornering or by asymmetrical deflections. During the braking process, a moment is developed about the lateral axis and excites the second bending eigenmode (S-shape) of the leaf spring. If the S-bump makes contact with the leaf spring, then there is an influence on the deformed shape of the spring.

On the basis of the MAN leaf spring test rig, a virtual test rig has been built up in SIMPACK (Fig. 5) to carry out the virtual testing of the behavior of the massless leaf spring model. There is good correlation between measurements and simulation by test rig scenarios. In Fig. 6, one can see the first three tests from Table 1 (K1 – K3) which are a combination of vertical load and braking moment. The pictures above show the measured (blue) and the SIMPACK produced (red) vertical forces while on the bottom pictures, the longitudinal displacement of the brake cylinder can be seen.

"The applied mechanical model has been validated via test-rig measurements at the MAN Truck & Bus AG in Munich."

The computing performance of the massless-model allows the user to carry out real time simulation with full vehicle models.

SUMMARY

During this project a new massless leaf spring model has been developed at Virtual Vehicle for SIMPACK applications. The assumption is linear beam statics, i.e., the model neglects the dynamic effects. The deformed shape has been determined with the help of the Guyan-reduction which secures an effective time integration process. Although the model is linear, a nonlinear coupling exists between displacement and rotation to consider the shortening effect. The S-Bump and the bearing eye are considered internally, parameterized with their spring characteristic and radius, respectively. The model has been implemented in SIMPACK as a user force element. There are two possible parameterizations. The first one is based on vertical stiffness. It needs only a stiffness value and the main dimensions to create a leaf spring model. For fine tuning of a given leaf spring, the second one is dedicated. It needs a geometry — thickness and width — to build up the stiffness matrix, which is a constant matrix and its terms are calculated only once before the time simulation. To reduce modeling time, and to simplify the transition between the different parameterization strategies, a leaf spring generator has been developed. It is steered with an ASCII-file with the input data sets.

The massless-model has been validated due to test-rig measurements. The results show very good correlations with the measurements. The comparison simulations with the elastic beam model show good agreement. The competitive CPU-time allows the application in real time full vehicle simulations.

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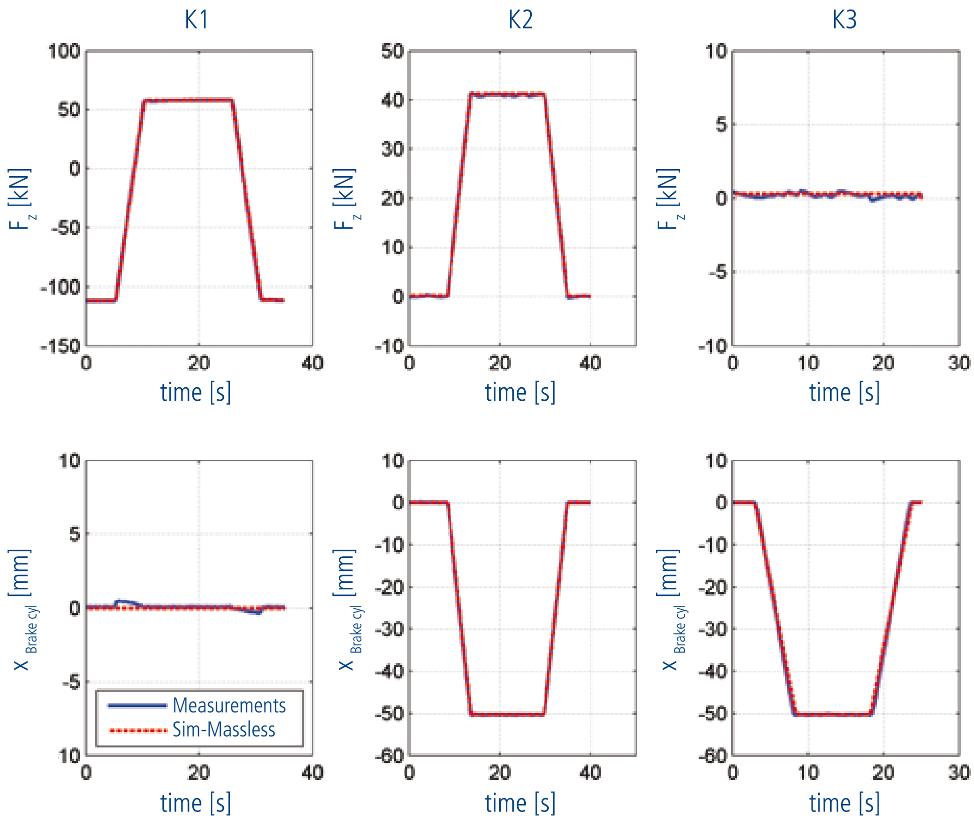

Fig 6: Test rig measurements at MAN Truck Truck & Bus AG vs. simulation with massless-model, test phase K1-K3, vertical force (above) and displacement of the hydraulic brake cylinder (bottom)