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Upgrading of the Hohenheim Tyre Model to a radial approach

For the passage of obstacle and the investigation of comfort relevant vibrations a number of tyre models has been developed in the past. Beside empirical approaches there are physical models with varying complexity. Most of the tire models have been developed for automotive applications. This contribution presents a new radial tyre model for the passage of obstacles. It is based on the existing Hohenheim Tyre Model and thus dedicated to model the behaviour of high volume agricultural tyres.

Keywords
Tyre, tyre model, passage of obstacles

Abstract
Landtechnik 66 (2011), no. 2, pp. 144-146, 5 figures,
4 references

Based on the upgrading of the Hohenheim Tyre Model [1] presented in [2] a radial tyre model has been developed, which further discretizes the tyre belt. It can be classified as a flexible tread band model and is capable to describe the behaviour of the contact patch in greater detail. As a result, the passage of short-wave undulations can be described without going back to empirical filtering approaches (compare to [2]). Furthermore, phenomena such as the displacement of the wheel load and the development of rolling resistance on a free rolling wheel can be described. During the development of the tyre model, attention was paid not just to a high simulation quality but also to a straightforward parameterization and short computation times. Within the contribution the model structure, the parameterization and the results of validation experiments are described.

Model structure

The model is supported by radially placed Voigt-Kelvin elements to the ground.

Spring and damper forces are calculated with the equations known from the existing Hohenheim Tyre Model. The Voigt-Kelvin elements rotate according to the angular velocity of the wheel. As a result, the interaction with the obstacle can be depicted realistically and the flexing of the tyre can be modeled. In this way the force distribution in the contact patch and thus the displacement of the wheel load and the rolling resistance can be modeled. The Voigt-Kelvin elements are uniformly distributed in a circular sector (**figure 1**). The size of the circular sector is defined before the simulation starts and is dependent on the maximum obstacle height. The advantage of this approach is the reduced number of radial elements and the resulting optimized computation time.

The angle between the radial elements has been varied between 1.5° and 3°. In order to specify the sampling of the ground profile three equidistant surface feelers are assigned to each radial element (**figure 1**). The displacement of a Voigt-Kelvin element equals the mean value of the three surface feelers. For the transmission of traction and brake forces the longitudinal model of the existing Hohenheim Tyre Model is added to the approach depicted in Fig. 1. The transient longitudinal force model acts in tangential direction on the resulting force application point of the radial springs. The resulting tyre forces are computed from the vectorial sum of all Voigt-Kelvin elements. Up to now the model is a planar approach. It is thinkable to integrate the lateral force model of the Hohenheim Tyre model.

Fig. 1

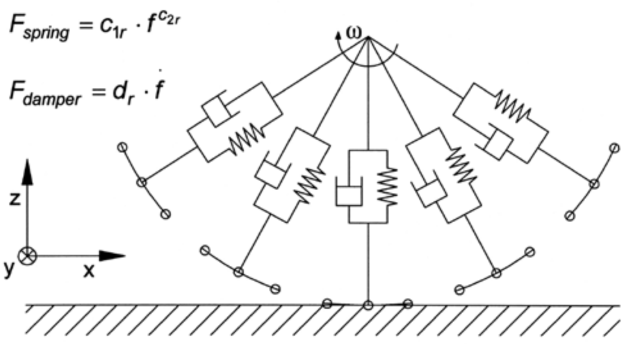

Fig. 1: Modular organisation of the radial model

Fig. 2

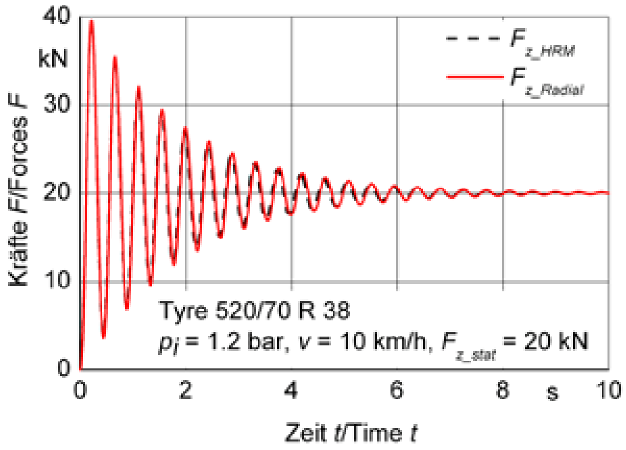

Fig. 2: Comparison of the oscillation decay of the Hohenheim Tyre Model (HRM) and the radial model

Parameterization

The parameterization of the radial elements is based on the parameter set of the Hohenheim Tyre model. In addition, the minimum obstacle length has to be defined. With a given design radius and the objective that during the obstacle passage at least two radial elements capture the undulation, the angle between the radial elements is calculated. After that the deflection of the model on a level surface is simulated. Thereby, the deflection f and the deformation rate $\dot{f}$ of the elements are recorded. These are the input parameters of the equations that compute the radial forces (**figure 1**). Subsequently, the spring stiffness c_{1r} and c_{2r} and the damping coefficient d_1r are varied iteratively, until the vertical dynamics of the radial approach match the dynamics of the existing Hohenheim Tyre Model. Results have shown that four iteration steps are sufficient. Finally the program offers the opportunity to write the results into a parameter file. Thus, once the obstacle dimensions are given, the parameterization is fully automated and takes just a couple of seconds. Model versions can be created and compared to each other in a simple way. The parameterization procedure was validated by comparing the oscillation decay behaviour of the tyre models. As it is shown exemplary in **figure 2** the divergence is small.

Experiments with the Institute owned single wheel tester [1] have shown, that the model underestimates the damping of the tyre. This may be attributed to the fact that the tyre used for the experiment has changed its properties since it has been parameterized. Moreover, until now the simulation model of the single wheel tester does not take the friction of the hinges into account. To achieve a better consistency between measured and simulated results, the vertical damping coefficient was increased by 30%.

Validation

For the validation of the radial tyre model experiments with the single wheel tester were conducted. In order to compare measured and simulated results, the MBS model (Multi-Body Simulation model) of the single wheel tester presented in [2] was used. A step-shaped obstacle with a height of 0.125 m was employed. Due to its length of 3.1 m reaction forces during the upward and the subsequent downward movement of the wheel can be evaluated independently. A 520/70 R 38 tractor tyre was tested. Tyre inflation pressure was set to 1.2 bar. The slip angle was adjusted to 0°. Within the **figures 3–5** measured and simulated longitudinal and vertical forces are compared to each other. Results of two different wheel loads and speeds are plotted.

The frequency of the oscillations induced by the obstacle passage are well corresponding during all test runs. The phase shift during the oscillation decay on or rather after the obstacle is small. The current model does not incorporate the radial runout of the tyre. Thus, the lug induced oscillations cannot be depicted. The simulation slightly underestimates the peak forces when the tyre hits the obstacle which holds true for the longitudinal forces in particular. This can be attributed to the fact that the radially placed elements do not interact up to now. They can deflect independently from each other. The magnitude of the vertical forces corresponds well, even though the tyre appears to have a higher damping rate in practice.

The computed displacement of the wheel load **e** varies between 1.5 and 2 cm depending on the wheel load applied. Values measured by Plesser [4] have the same order of magnitude. Rolling resistance torque is calculated by multiplying the vertical force by the displacement of the wheel load. The longitudinal force which acts on the lever arm r_l (distance between wheel hub and ground) compensates the rolling resistance. The model underestimated the rolling resistance noticeably. It needs to be investigated whether this can be attributed to the fact that the model does not take shear forces in the contact patch into account.

Conclusions

The presented tyre model has been developed to model tyre behaviour during the passage of obstacles. It is based on the Hohenheim Tyre Model, whereby the model equations of the radially placed Voigt-Kelvin elements have been adopted from the vertical model of the existing approach. Furthermore, the radial model has been complemented by the longitudinal model of the Hohenheim Tyre Model. A straight forward parameterization and short computation times were in the focus of attention during the development of the model.

Primary validation experiments show a good agreement of measured and simulated results. However, it became clear that the radial elements need to be linked to each other in order to improve simulation quality. In addition, coming model versions should be capable to describe radial runout of the tyre and the effect of the lugs on tyre vibration.

Fig. 3

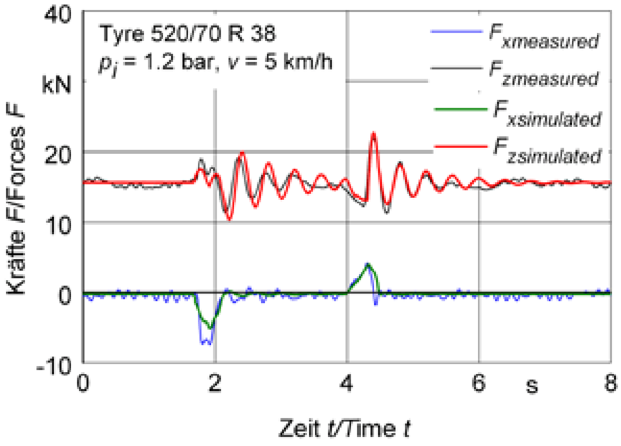

Fig. 3: Forces while driving over a step obstacle (F_z = 15.6 kN), measured and simulated

Fig. 4

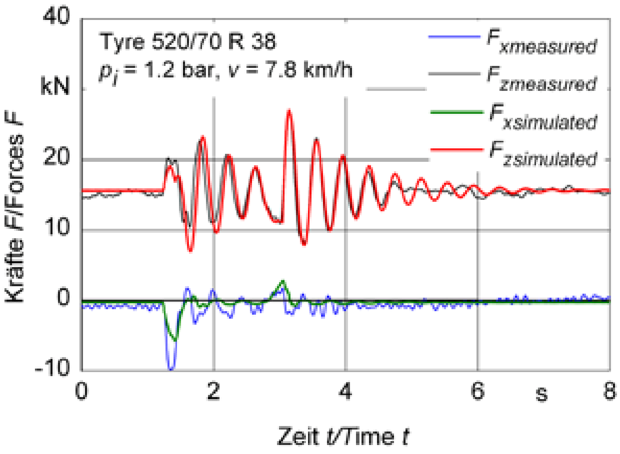

Fig. 4: Forces while driving over a step obstacle (F_z = 15.6 kN), measured and simulated

Literature

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Fig. 5

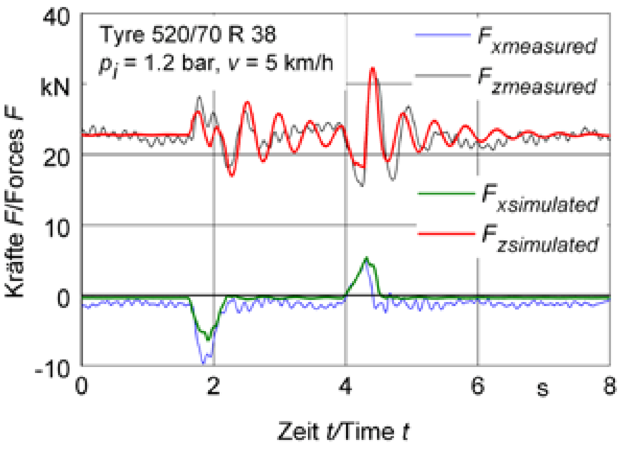

Fig. 5: Forces while driving over a step obstacle (F_z = 22.6 kN), measured and simulated