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Dynamic Longitudinal Tyre Model for Agricultural Tyres

Efforts are being made to increase modern tractor driving speeds up to 60 km/h or even more. Vibrations originating from unsuspended rear axles cause problems, since the rear tyres provide the complete suspension and damping there and their characteristics in this respect are rather poor. Thus, shaking caused by road unevenness and tyre asymmetry, combined with less than optimal tractor component settings, can lead to critical driving conditions. To be able to predict these, accurate tyre models are needed. For this reason a dynamic longitudinal tyre model is presented here.

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Dedicated to Prof. Dr. Ing. Dr. h.c. H. D. Kutzbach on occasion of his 65th anniversary.

Keywords

Tyre modelling, driving dynamics, transient behaviour, multibody simulation

While many modelling approaches are widely used for passenger car tyres, there are hardly any models available for tractor tyres. Due to the wheel suspension of passenger cars, which takes over the most of the vibration damping, the effect of the tyre on vehicle vibrations is significantly weaker than on tractors. Thus, these models use some simplifications, which cannot always be used when simulating soft tractor tyres. Additionally, most of these models are either steady state models or have limited transient behaviour capabilities. Hence, their use for tractor tyre simulation is possible only under restrictions. Therefore, a tractor tyre model with transient behaviour capabilities is developed at Hohenheim University.

Modelling of vertical forces

The Hohenheim Tyre Model shall be able to reproduce the three-dimensional transient behaviour of tractor tyres using a small number of input parameters as possible. Additionally, the used input parameters should be determinable on two test rigs available at Hohenheim University. The flat belt stand [1, 2] shall be used for determining vertical tyre parameters, while determining the longitudinal parameters shall be carried out on the single wheel tester [3, 4]. For the vertical force calculation, the equation set up by Plesser [2] is used:

$$F_Z = c_{1Z} \cdot f_Z^{c_{2Z}} + d_{1Z} \cdot \frac{1}{v^{d_{2Z}}} \cdot \dot{f}_Z \quad (1)$$

There F_z is the vertical force, f_z the vertical deflection and v the driving speed. The spring parameters c_{1z} and c_{2z}, as well as the damping coefficients d_{1z} and d_{2z} can be determined by drop tests on the flat belt test stand. Due to their wheel load and driving speed dependency, drop test at different speeds and wheel loads have to be done. Finally, the parameters are linearized between the measured values.

Since the rear axles of tractors are normally unsuspended, the excitations caused by road unevenness and tyre run-out are transferred directly to the vehicle body. It's eigenfrequency is generally at about 2 Hz. Excitations at these frequencies, caused by tyre run-out, occur at critical driving speed, which depends on tyre radius and lies between 30 and 50 km/h. As shown by Böhler [5], the tyre run-out can be measured and implemented as a Fourier series. He took the lug influence into account as well, and extended Pacejkaís MagicFormula [6].

In order to reduce the measuring efforts, the Hohenheim Tyre Model considers the tyre run-out by a simple sinus excitation with amplitude up to 5 mm, which accords to the usual tyre run-out range given by the manufacturer. Due to it's high frequency, lug excitation doesn't affect vehicle's handling performance and is thus neglected.

Modelling of longitudinal forces

Usually, the typical relationship between slip σ and net traction ratio $\kappa = F_x/F_z$, shown in *Figure 1*, is used for calculation of longitudinal forces. Thereby, the slip is used as input and net traction coefficient as output for the longitudinal force calculation, depending on the wheel load. The transient tyre behaviour can be reproduced by extensions like relaxation length. This method is already a good approximation and is often used for simulation of passenger car tyres. Tractor tyres however, have a notedly different transient behaviour, so the use of these tyre models for tractor tyre simulation is limited.

Unlike the modelling approach above, the Hohenheim Tyre Model assumes the longitudinal force to be the cause of slip (Fig. 1). Due to mathematical reasons, the used κ - σ relationship deviates from the commonly used one so the accuracy of the model is momentarily assured up to 30 % slip.

The calculations of the Hohenheim Tyre Model are based on equations set up by Bernard and Clover [7]. They calculate the transient slip and use it for longitudinal force calculation as input for the κ - σ relationship. The Hohenheim Tyre Model also calculates the transient slip, but assumes it to be equal to the longitudinal tyre deflection. All variables relevant for the calculation are shown in *Figure 2*.

The longitudinal deflection velocity of the tyre is calculated as follows:

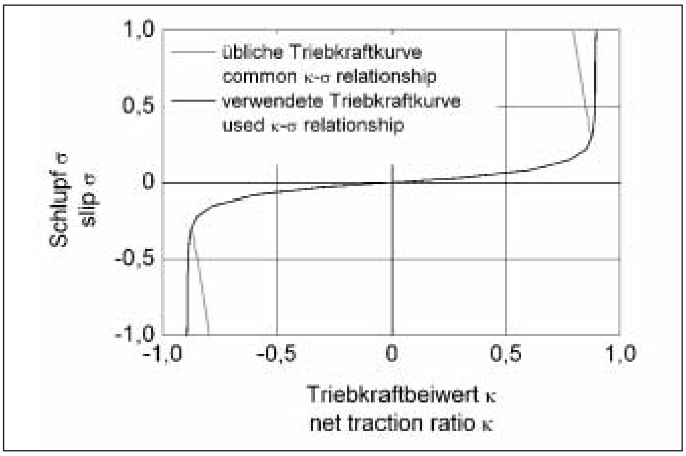

Fig. 1: Common and in the Hohenheim Tyre Model used κ - σ relationship

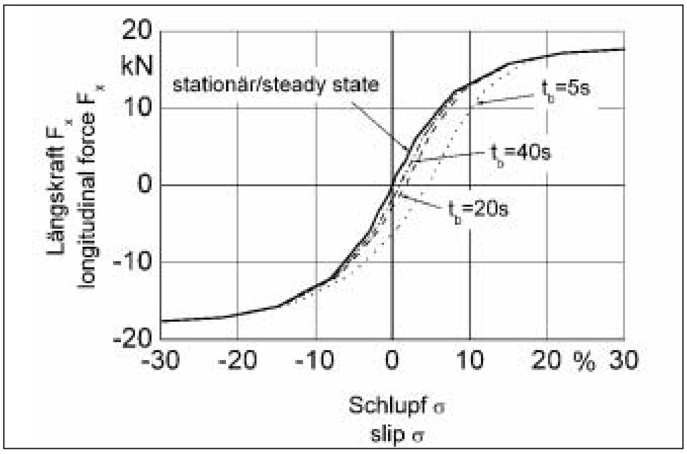

Fig. 3: Simulated longitudinal forces on a 520/70 R 34 tyre under transient slip conditions

$$\frac{d}{dt} f_x = r_{dyn} \cdot \omega - v_{tx} - \left| r_{dyn} \cdot \omega \right| \cdot \sigma_{st} \quad (2)$$

Analogous to the driven wheel, the longitudinal deflection for a braked wheel is calculated:

$$\frac{d}{dt} f_x = r_{dyn} \cdot \omega - v_{tx} - \left| v_{tx} \right| \cdot \sigma_{st} \quad (3)$$

There are: f_x - longitudinal deflection, r_{dyn} - tyre rolling radius given by the manufacturer, ω - angular velocity, σ_{st} - steady state slip as a function of net traction ratio κ, v_{tx} n real driving speed and v_{xth} - theoretical driving speed.

These deflection speeds and in another integration step calculated deflections are then used as the input for a Voigt-Kelvin-Element, set up by Plesser [2] for longitudinal force calculation:

$$F_x = c_{1x} \cdot f_x^{c_{2x}} + d_{1x} \cdot \frac{1}{v^{d_{2x}}} \cdot \dot{f}_x \quad (4)$$

As shown by Plesser, the lever of the rolling resistance is linked to braking torque and consequently with the longitudinal force. Simplifying, the calculated longitudinal deflection is added to the torque arm of the motion resistance and the following equation is used:

$$e = r_{st} \cdot \rho + f_x \quad (5)$$

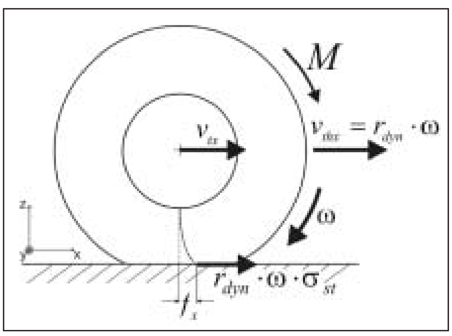

Fig.2: Driven wheel with relevant variables

Whereas r_{st} represents the distance between the wheel hub and the ground, ρ is the motion resistance ratio and f_x is the longitudinal tyre deflection. So calculated dynamic torque arm of rolling resistance is used in the equilibrium of torques equation.

Simulation results

The validation, as well as the parameter determination, of the tyre model will be carried out on the test stands of the Department of Agricultural Engineering. The parameters used in the following were determined on the flat belt test stand by Plesser [2] for a pulled wheel. The model was verified on a MBS-model of the flat belt test stand, as well as on a model of the single wheel tester [4]. *Figure 3* shows the longitudinal forces simulated with the single wheel tester model under transient slip conditions. Thereby, the wheel is accelerated from -30 % up to + 30 % slip. The simulated tyre dimensions are 570/70 R 34, inflated to 0.8 bar, and a static wheel load of 20 kN. The used parameters for the driven wheel are momentarily estimated values, which will be verified in near future.

Summary and future work

As already shown, the Hohenheim Tyre Model is generally able to reproduce the transient tyre behaviour correctly. Unlike most empirical models, the Hohenheim Tyre Model uses velocity vectors of the wheel as input while the slip is calculated. Additionally, physical parameters like stiffness and damping coefficients are used for force calculation. Thus, the rather hard to determine value of relaxation length has not be used as input parameter. Due to the high relevance of the lateral force for driving dynamics, the Hohenheim Tyre Model will be extended by this component in the near future. The physical principles of the longitudinal force calculation will be also used on lateral force. Furthermore, methods for parameter identification in longitudinal and lateral direction will be worked out. The emphasis will be on the low parameter number. A simple run over obstacles will be implemented as well.

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