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HOHENHEIM TYRE MODEL - A DYNAMIC MODEL FOR AGRICULTURAL TYRES

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Abstract: Concerning the fact that tractor speed had increased in last few years concern about safety and comfort had risen. Since most tractors do not have a suspended rear axle, the suspension is done solely by the tyers. Their insufficient damping properties, different excitations during the ride and inadequate adjustment of different components like cabin and front axle suspension can lead to critical driving situation and even loss of control over the vehicle.

This paper presents University of Hohenheim's three-dimensional dynamic tyre model, for agricultural tyres. This model has been developed and applied to a MBS-Model of a tractor. Result show that the model is able to reproduce the transient behavior of agricultural tyres within the driving dynamics and can be used within multibody simulation.

Key words: *safety, tyers, MBS model.*

INTRODUCTION

The driving speed of modern tractors increased during last years up to 60 km/h and even higher velocities are aspired. Therewith, the comfort and safety requirements are increasing. Since most tractors do not have a suspended rear axle, the suspension is done solely by the tyres, whose damping is rather low. Besides that, the tyres have some run-out, normally up to 5 mm which, in spite of tyre matching, can cause strong vibrations of the vehicle, especially near the vehicle's eigenfrequency. The eigenfrequency of a standard tractor is usually between 2 and 3 Hz, depending on vehicle's mass and the tyre pressure. Since the excitation is caused by the tyre run-out, there is a critical velocity, which is dependent on the rolling radius of the tyre and corresponds to the eigenfrequency. These vibrations affect firstly the comfort of the driver and secondly the safety, by affecting vehicle's driving dynamics behavior.

To be able to predict and to avoid critical situations at an early stage of development as well as to help to adjust harmonize components like cabin and front axle suspension with each other, a multi-body model (MBS - model) of a tractor can be used. Since the tyres are the direct link between the vehicle and the ground, their attributes have a tremendous influence on vehicle's behavior. Hence, an exact tyre model is a basis for a correct model of a tractor.

OBJECTIVE

Although different tyre models are already available, they are strongly focused on passenger car tyres and limited in their application to soft tractor tyres with their non-linear behavior. Furthermore, these models require specific test stands for their parameterization, which do not exist for the dimension of agricultural tyres. Thus, at the University of Hohenheim a three-dimensional dynamic tyre model for agricultural tyres has been developed and applied to a MBS-Model of a tractor. The focus of this tyre model is put on a small number of input parameters, which all can be determined on university's test stands. Besides that, the calculating speed has to be as high as possible.
The input parameters needed by the Hohenheim Tyre Model are shown in the figure below. All of them have a physical background and they can be determined on institute's test stands or found in the literature.

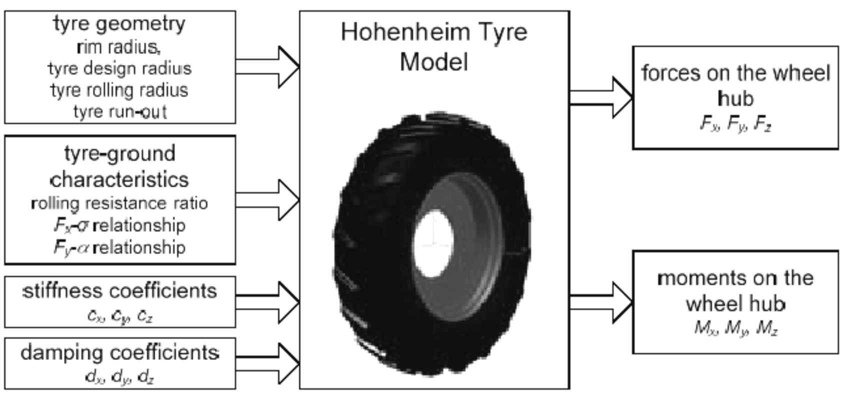

Figure 1: Hohenheim Tyre Model with its input parameters and output values.

As shown in Figure 1, the 13 input parameters can be divided into four main groups. Tyre geometry parameters can be directly measured or are given by the manufacturer. The tyreground characteristics, stiffness and damping coefficients can be found in the literature [1 - 5] or measured on institute's test stands.

It must be pointed out, that all these parameters are valid under particular operating conditions, which means particular inflation pressure and static wheel load. It is not possible to predict the tyre's behavior under changing conditions yet.

METHODS

The Hohenheim Tyre Model can be described as a physical model, since it is calculating the deflections of the tyre three-dimensionally. However, it can also be called empirical, since it contains some characteristic curves, namely $F_{x\text{-}\delta}$ and $F_{y\text{-}\alpha}$ relationships. It is built up in MATLAB/Simulink and can be coupled with any MBS-software via co-simulation. There are other coupling possibilities like code export of the MBS-model into MATLAB/Simulink. Their main advantage is the combination of the advantages of the respective software. However, an additional software licence is needed.

The MBS-software and MATLAB/Simulink exchange data at every calculation step which requires a fixed calculation frequency. For the Hohenheim Tyre Model, the recommended step size is usually 0,002 s, which is a good compromise between accuracy and calculating speed. As a matter of course, this can be adapted to the requirements of the respective vehicle model. The exchanged values are velocities and position vectors of the wheel which are needed for force and torque calculation within the tyre model. These are given back to the vehicle model, as shown in Figure 2.

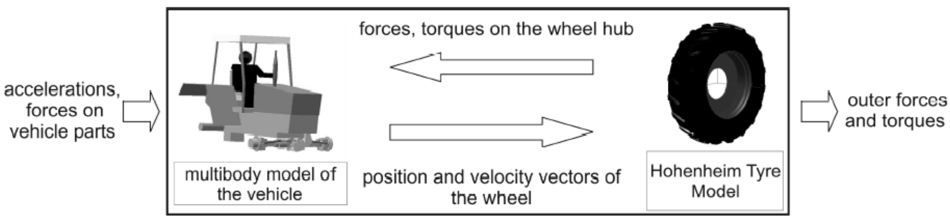

Figure 2: Exchange values of the Hohenheim Tyre Model.

The calculation of the forces is done by calculating the deflection velocity and the deflection, which are used as input for the three-dimensional spring-damper systems. For the pure longitudinal deflection the following equations are used for the braked and driven wheel:

$$\frac{d}{dt} f_x = r_{dyn} \cdot \tilde{\omega} - v_{tx} - |v_{tx}| \cdot \sigma_{st} \quad \text{and} \quad \frac{d}{dt} f_x = r_{dyn}\, \omega - \tilde{v}_{tx} - |r_{dyn} \cdot \omega| \cdot \sigma_{st} \tag{1}$$

where: f_x - longitudinal deflection, r_{dyn} - tyre rolling radius, ω - angular velocity of the wheel, σ_{st} - steady state slip as function of the longitudinal force, v_{tx} - real driving speed. These terms are nearer explained in some other publications already [6], so they are not explicated further here. However, it is important to point out that the slip definition is the in agricultural engineering widely used and reads as follows for the braked and driven wheel:

$$\sigma_{st} = \frac{r_{dyn} \cdot \omega - v_{tx}}{v_{tx}} \quad \text{and} \quad \sigma_{st} = \frac{r_{dyn} \cdot \omega - v_{tx}}{r_{dyn} \cdot \omega} \tag{2}$$

It has to be mentioned that the Hohenheim Tyre Model does not calculate the rolling radius of the tyre yet. For purposes of driving dynamics the use of the value given by the manufacturer is sufficient. However, r_{dyn} can be measured and mathematically approximated or given by a look-up table if needed.

The calculation of lateral deflection velocity for a wheel under pure lateral slip reads:

$$\frac{d}{dt}f_y = -v_y - |v_{tx}| \cdot \tan\alpha_{st}$$

where: f_y - lateral deflection, α_{st} - steady state slip angle as function of the lateral force and v_y - real lateral speed. The pure longitudinal and lateral slip, are special cases of combined slip which is met on a wheel, like shown in the Figure 3. Besides the real velocity of the wheel hub, v_{tx} and v_y, *and* the theoretical wheel velocity $r_{dyn}\ \omega$, the sliding velocity v_{sl} of the tread (hatched area) over ground is also depicted. The deflections f_x and f_y are the integral of differences between the velocity of the wheel v_{ges} and the sliding velocity v_{sl} see equations (1), (3) and also equations (4) - (7).

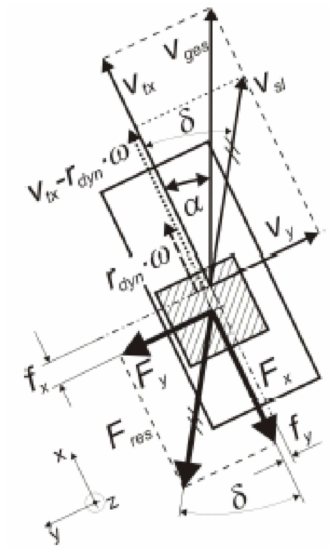

Figure 3. Braked wheel with applied slip angle α and the acting forces with the force angle δ.

Additionally, Figure 3 shows the forces acting on the tyre. Note that the force angle δ is not equal to α. For steady state conditions, the wheel hub velocity and the tread velocity are equal, so no further deflection takes place. In this case, the direction of the resulting force on the tyre, F_{res} and the sliding direction of the tread are equal. Since the equations (1) and (3) represent pure longitudinal and lateral slip, they have to be coupled for combined slip conditions. For the longitudinal deflection velocity this is done by following equation for the braked wheel:

$$\frac{d}{dt}f_x = r_{dyn}\cdot\omega - v_{tx} - |v_{tx}|\cdot\sigma_{st} + \frac{|v_{tx}|\cdot tan\alpha_{st}}{tan\sigma} \qquad (4)$$

And by the equation (5) for the driven wheel:

$$\frac{d}{dt}f_x = r_{dyn}\cdot\omega - v_{tx} - |r_{dyn}\cdot\omega|\cdot\sigma_{st} - \frac{|r_{dyn}\omega|\cdot tan\alpha_{st}}{tan\delta}. \qquad (5)$$

The lateral deflection velocity for combined slip conditions for the braked wheel reads:

$$\frac{d}{dt}f_y = -v_y - |v_{tx}|\cdot tan\alpha_{st} - |vtx\cdot\sigma_{st}|\cdot\frac{F_y}{|F_x|} \qquad (6)$$

And for the driven wheel:

$$\frac{d}{dt}f_y = -v_y - r_{dyn}\cdot\omega\cdot tan\alpha_{st} - |r_{dyn}\cdot\omega\cdot\sigma_{st}|\cdot\frac{F_y}{|F_x|} \qquad (7)$$

The hereby calculated deflection velocities and the corresponding deflections are then used as input for spring-damper systems as shown below for the longitudinal and lateral force:

$$F_x = c_{1x}\cdot f_x^{c_{2x}} + d_x f_x \text{ and } F_y = c_{1y}\cdot f_y^{c_{2y}} + d_y\cdot f_y \qquad (8)$$

where: F - force, c_1 - stiffness, c_2 - stiffness coefficient and d - damping coefficient, indices x and y stand for longitudinal respectively lateral direction.

RESULTS

The tyre model is verified using the flat belt test rig and the single wheel tester, which were also modeled in SIMPACK. The test rigs are used for parameter determination as well. They were described in different publications before [2,3,7] and will thus not be explained here.

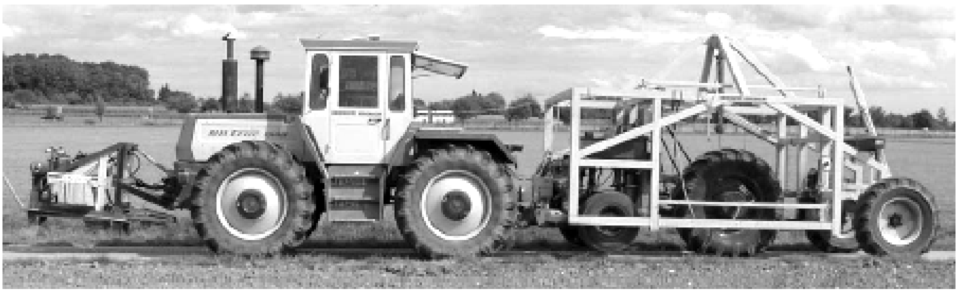

Figure 4: The single wheel tester.

The input parameters used for these calculations were: c_{1x} = 300 kN/m^{c2x}, c_{2x} = 0,95, d_x = 0,27 kNs/m, c_{1y} = 88,25 kN/m^{c2y}, c_{2y} = 0,96 and d_y = 2,7 kNs/m. They were obtained on the single wheel tester. Since the deflection velocities are relatively low, the damping has a rather small influence on the simulation results. To avoid inaccuracies due to the complexity of the test stand, measured velocities, the wheel load and the slip angle were used as input. The following figure shows model validation for the pulled wheel done on the single wheel tester. The tested tyre was a 520/70 R 38 tyre with an inflation pressure of 1,2 bar and a static wheel load of 20 kN.

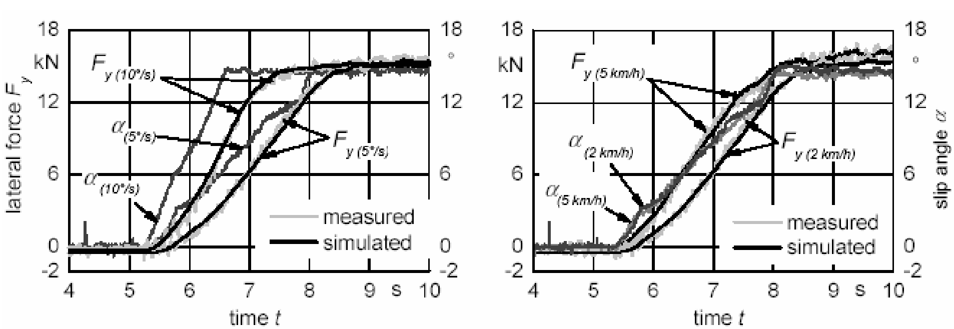

Figure 5: Pulled wheel at 2 km/h and steering velocities of 5 °/s and 10 °/s (left graph) and at 2 km/h and 5 km/h and a steering velocity of 5 °/s (right graph).

As shown in the figures above, the typical first order behavior of the lateral force can be simulated with the Hohenheim Tyre Model with high accuracy at different driving and steering velocities. Some validation results for combined slip conditions are shown below. For these tests, the slip angle was kept constant and the wheel was accelerated from -60% to +60% slip, as shown in the left graph. The deviations occurring in the figure below are caused by inaccuracies due to varying road surface which add up in the right graph. Thus, they appear higher than in the left. This kind of curves can be often found in the literature and is usually measured under steady state conditions, where the maximum of the lateral force is at negative longitudinal force [8]. Dynamic effects occurring due to tyre deflection cause a movement of the maximum to the positive longitudinal force area, as shown in Figure 6.

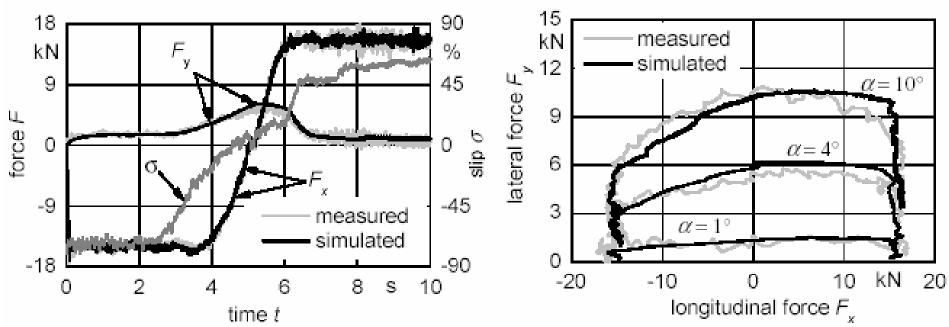

Figure 6: Wheel with a slip angle α = 4° and acceleration from -60% to +60% slip within 5 s (left graph) and the same manoeuvre with different slip angles (right graph).

CONCLUSION

First comparisons between the simulation results and measured data show that the Hohenheim Tyre Model is able to reproduce the transient behaviour of agricultural tyres within the driving dynamics relevant frequency range and can be used for multibody simulations. It is numerically robust, since the slip is not used as an input value. The short calculating time (real-time factor 1, at 500 Hz) is to be mentioned as well. Another attribute of Hohenheim Tyre Model is a low number of parameters, which all have a physical background and can be determined on university's test stands.

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DINAMIČKI MODEL PNEUMATIKA ZA POLJOPRIVREDU

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Sadržaj: Obzirom na činjenicu da se opseg radnih i transportnih brzina traktora stalno povećava, u poslednjih pet godina zahtevi u pogledu konfora i sigurnosti su na višem nivou. Kako većina traktora nema suspenziju zadnjeg mosta onda se ona može izvesti samo preko pneumatika. Njihove nedovoljne amortizacione osobine, zatim različita opterećenja tokom vožnje i neadekvatna podešenost ostalih komponenti (kabina, amortizeri na prednjem mostu) mogu dovesti do do rizičnih situacija tokom vožnje pa i do gubitka kontrole nad vozilom. U ovom radu je prikazan tro-dimezioni dinamički simulacioni MBS model vozila (Multibody Simulation Model) predložen od Univerziteta u Hohenhajmu. Takođe, na Univerzitetu je razvijen i MBS model pneumatika usklađen sa MBS modelom traktora. Upoređenje simuliranih i merenih rezultata ukazuje da je model u stanju da realno predstavi dinamičko ponašanje pneumatika tokom vožnje. Takođe, ovaj model pneumatika ima mali broj ulaznih parametara koji se lako mogu odrediti.

Ključne reči: *bezbednost, pneumatici, MBS model.*