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Quantified validation of the Hohenheim Tyre Model for driving dynamics investigations

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The Hohenheim Tyre Model was developed for investigations regarding driving dynamics as well as driving comfort. Before its application as development tool, it has to be thoroughly validated. Regarding vertical and lateral investigations, the model is validated by comparing simulation results with measurements on single tyres and on whole vehicles. A single wheel tester and an equipped test tractor are used to determine the forces acting between the tyre and the road surface as well as the driving conditions during certain manoeuvres. The conducted road tests are reproduced using multi-body simulation models (MBS) simulation models in connection with the Hohenheim Tyre Model. By using separate evaluation metrics regarding the magnitude and phase behaviour, a good correlation between measurements and simulation results is shown.

Keywords
Agricultural tyres, tyre model, validation, driving dynamics

Due to the structural changes in agriculture, tractors are increasingly used for transport tasks. The transport volume and permissible total masses are limited by the legislation and are already fully exhausted. A further increase in transport capacity can only be achieved by increasing the driving velocity. This leads to rising demands towards driving safety of the vehicle combinations and towards the isolation of the driver regarding whole body vibration (VO (EU) Nr. 167/2013). By using multibody simulation models (MBS), an evaluation of the driving safety and the driving comfort can already be conducted in the development stage of a vehicle. The tyre model, which is used to calculate the transmittable forces acting in between the tyre and the road surface as well as the spring and damping behaviour of the tyre, is of major importance for these simulation models. To be applicable for driving dynamics and driving comfort investigations, these tyre models have to be validated using test stand and road test measurements.

The tyre model developed at the Institute of Agricultural Engineering at the University of Hohenheim is applicable for both driving dynamics and driving comfort investigation purposes. It has already been validated regarding investigations in longitudinal and vertical direction for level surfaces as well as for discrete obstacles. The validation regarding lateral dynamics is limited towards level surfaces so far. Additionally, the influence of the wheel load on the lateral force transmission has not been investigated yet (Witzel 2015).

The influence of the wheel load on the lateral force transmission in the tyre's contact patch was already experimentally investigated using artificial wheel load excitations (Schlotter 2006). In this contribution, the characteristic force transmission principle is reproduced using an appropriate simulation model of the single wheel tester. Subsequently, the tyre model is validated for whole vehicle applications. For this purpose, road tests with a test tractor are conducted. These tests are then re-

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produced using open-loop simulations with a MBS whole vehicle simulation model, which is provided with the recorded driving velocity and steering angle progressions. In automotive engineering, simulation models are often only validated using subjective or optical comparisons between measurements and simulation results, i. e. regarding the force progressions. To increase the resilience for further validation of the Hohenheim Tyre Model, separate metrics for the evaluation of the magnitude and phase correlation are introduced.

Test record

A single wheel tester was designed at the Institute of Agricultural Engineering for the investigation of the horizontal force transmission characteristics of large volume agricultural tyres on different surface types (Armbruster 1991, Barrelmeyer 1996). In this test stand, a single tyre with up to 2 m in diameter and up to 800 mm in width can be investigated due to its mounting on a measuring hub. The wheel load F_z can be controlled within a range from 0 to 60 kN using a hydraulic cylinder. Additionally, a static wheel load $F_{z,stat}$ can be applied by using different ballast masses for measurements on rough surfaces. A second hydraulic cylinder controls the slip angle α of the tyre between 0 and 16°. The wheel can also be decelerated or accelerated via a hydraulic motor to regulate different longitudinal slip values. By decoupling the hydraulic motor, the wheel is able to roll torque-free. Additionally, the camber angle β can be adjusted manually between −20 and +20°.

Schlotter (2006) used the test stand described to investigate the influence of wheel load fluctuations on the lateral force transmission in the tyre's contact patch. Fluctuations around the static wheel load $F_{z,stat}$ can be caused by different circumstances. On the one hand, tyre-induced wheel load fluctuations are caused by inhomogeneity of the dynamic rolling radius r_{dyn}. This also includes fluctuations of the wheel load, which are excited into the tyre's contact patch via the chassis as reaction to certain driving manoeuvres. On the other hand, road-induced wheel load fluctuations are caused by an excitation of the tyre's contact patch while moving on rough surfaces. Based on these findings, Schlotter conducted two series of experiments. Using the wheel load control cylinder of the single wheel tester, he first excited the tyre with reproducible sinusoidal progressions of the wheel load around certain static wheel loads. The transmittable lateral force F_y decreased with an increasing magnitude of the wheel load fluctuation compared to static conditions. In summary, this leads to a declining relation between the wheel load and the transmittable lateral force. In a second step, Schlotter generated the wheel load fluctuations by passing over different discrete obstacles (pulse, step and ramp) by applying different static wheel loads $F_{z,stat}$ using ballast masses. These investigations also led to a non-linear relation between the dynamic wheel loads and the lateral forces. Schlotter determined the interaction of the contact patch area and the contact patch pressure distribution as origin of this behaviour. For small wheel loads, an increase in wheel load leads to a quick increase in contact patch area while the contact patch pressure distribution is relatively constant. For larger wheel loads, the increase in contact patch area is less distinctive. Additionally, the contact patch is more unequally loaded. This leads to reduced transmittable lateral forces.

For whole vehicle investigations, wheel load fluctuations can only be generated by means of dynamic driving manoeuvres. Passes over discrete obstacles for stationary driving conditions are not feasible for test drivers. This is especially valid for the adjustment of stationary slip angles with respect to various environmental effects. For this reason, this research is conducted using standardised driving manoeuvres for the determination of the lateral vehicle characteristics. The manoeuvres are slightly modified to be applicable for agricultural tractors. These tests allow to conduct reproducible vehicle excitations. On the one hand, steady state circling can be used for stationary driving conditions. The variation of the parameters radius r of the test track and the driving velocity v_D conduce to the adjustment of a constant lateral acceleration α_y and constant slip angles α_i at the different wheels. On the other hand, sine steering and evasive manoeuvres allow the investigation of the transient force transmission principles between the road surface and the tyres for dynamic changes of the driving conditions. The dynamics during these tests can easily be influenced by the driving velocity v_D.

To execute the road tests described, a Fendt Favorit 509C test tractor is instrumented with measurement equipment (Figure 1). Key elements of the measuring setup are the three measuring rims (Späth 2004) at the two rear wheels and at the right front wheel. In combination with three angular sensors for the determination of each angular wheel position and acceleration sensors for the determination of the wheel's inertia forces, the wheel hub forces F_x, F_y and F_z are measured in the wheel-fixed coordinate system. The current driving condition is determined by an Inertia Measuring Unit (IMU), which is attached just below the vehicle's centre of gravity. It records the accelerations α_x, α_y and α_z as well as the angular velocities ω_x, ω_y and ω_z. An optical velocity sensor below the rear axle is used to obtain the velocity components v_x, v_y and the vehicle's attitude angle $\beta_{attitude}$. Additionally, the steering angle δ is measured at the right wheel carrier. In combination with the angular velocities determined by the IMU, the steering angle rate of change $d\delta / dt$ and the velocity components are used to calculate the slip angles α_i at the wheels. Road tests were conducted with the instrumented test tractor at the Agricultural Experimental Station Ihinger Hof and on an airport runway in summer 2016.

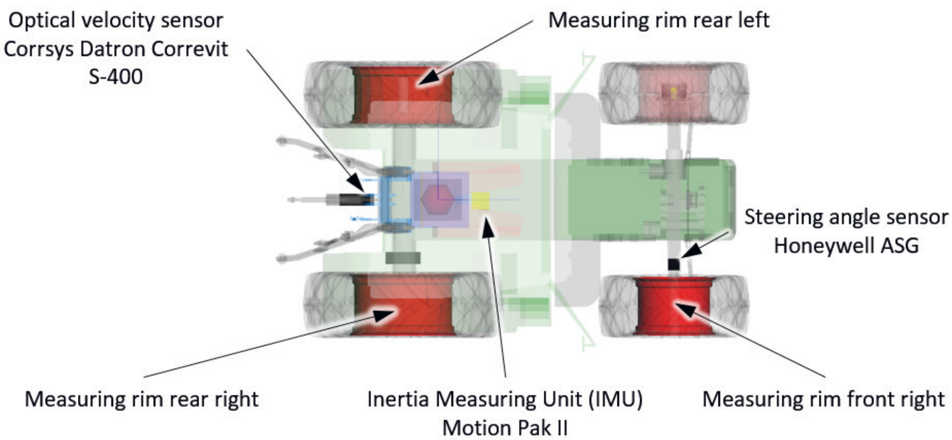

Figure 1: Top view of the Fendt Favorit 509C test tractor measurement setup

Simulation model setup

To compare measurements and simulation results, the experiments described in the previous section are reproduced using adequate simulation models. The single wheel tester is modelled as single mass oscillator with sufficient accuracy (Equation 1). Since the wheel hub is connected to the tester's frame via a lower and an upper link with appropriate lengths, the movement of the wheel hub in the considered deflection range can be described as purely vertical. Additionally, the wheel mounting is almost ideally stiff in lateral direction. The wheel hub with its connected wheel as well as ballast masses or a controlled vertical force respectively and proportionate masses of the links are summarised in an oscillating mass m. The joint friction is considered as vertical damping force F_D. The Hohenheim Tyre Model is connected to the model in between the road surface and the wheel hub as crucial force element. In vertical direction, the tyre model calculates a resulting dynamic wheel load F_z. For a correct representation of the angular velocity ω_y and the longitudinal force F_x at the wheel, the principle of angular momentum is additionally considered with respect to the mass moment of inertia of the wheel and the test stand's drive unit (Equation 2). The tyre model is manually provided with additional test parameters like the driving velocity v_D, the slip angle α, the camber angle β and the vertical distance between the road surface's zero level and the wheel hub h for the consideration of obstacle passes and the calculation of the lateral force F_y. Alternatively, they are provided based on measurements according to an open-loop simulation run.

$$m \cdot \ddot{z} - m \cdot g - F_D - F_z = 0 \qquad \text{(Eq. 1)}$$

$$M_y - J \cdot \omega_y = 0 \qquad \text{(Eq. 2)}$$

Although the kinematic relations for a whole vehicle are more complex than for the single wheel tester, a MBS simulation model of the test tractor is used for the validation regarding whole vehicle applications of the tyre model. This MBS simulation model is based on the Fendt Favorit 509C model developed by Böhler (2001) and is assembled in the MBS software SIMPACK. Joints, constraints and force elements are used to attach the different substructures to the chassis, which consists of the motor, the transmission and the rear axle. The hydro-pneumatically suspended front axle, the rubber bushing suspended driver's cabin, the driving seat and the four wheels are modelled in these respective substructures. Four Hohenheim Tyre Models are included as external force elements in MATLAB/Simulink within the MBS model's wheel substructures to induce the forces acting in between the road surface and the tyres. Additionally, a drive and a steering angle controller are included in MATLAB/Simulink to reproduce the driving velocity and steering angle progressions recorded in the test runs. Both simulation partners SIMPACK and MATLAB/Simulink are coupled via the Co-simulation interface SIMAT. The vehicle's driving condition and the calculated tyre forces and torques are exchanged at a fixed time step of 0.2 ms.

The Hohenheim Tyre Model, which is used for investigations of the lateral dynamics at the single wheel tester and the test tractor, has been extensively described (Witzel 2015). To be applicable for both driving dynamics and driving comfort investigations, a spoke approach is used (Figure 2). The spokes are arranged with an angular resolution of 2 to 3° over a certain angular sector – depending on the road surface profile – to reduce computational time. Each spoke is equipped with radial, tangential and lateral Voigt-Kelvin elements. Depending on the relevant force transmission characteristics in the respective coordinate directions, the elements are implemented as linear or nonlinear. Additionally, the model is able to reproduce obstacle passes by using interradial springs in between adjacent radial elements with respect to the local deflection behaviour of the tyre. The tyre-surface interaction for each spoke is modelled using a stick-slip approach. Special scan points are used for the force transmission as well as for scanning the road surface. Different coefficients of friction in longitudinal and lateral direction result in an anisotropism regarding the horizontal force transmission, which is characteristic for agricultural tyres. With that, the reproduction of a friction ellipsis is possible. Additionally, two further force elements are defined in between the wheel hub and the spoke connector with respect to the transient force transmission in longitudinal and lateral direction. A torsional spring-damper element allows a relative rotational movement of the tread against the rim. The axial Voigt-Kelvin element allows a lateral deflection of the tread against the rim respectively.

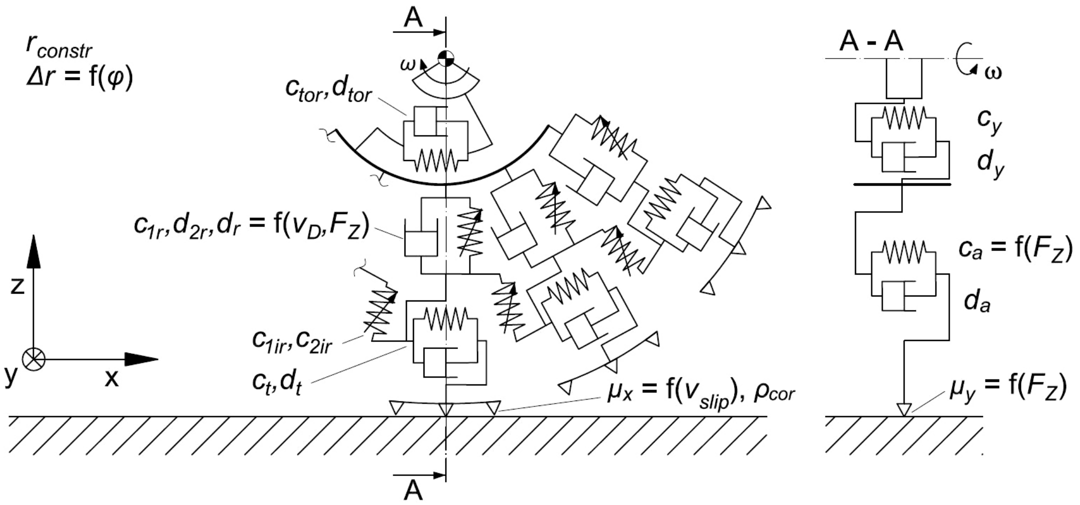

Figure 2: Structure and parameters of the Hohenheim Tyre Model in side view (left) and sectional view (right)

For the investigation of the lateral force transmission, the lateral force elements c_a and d_a on each spoke were revised during this research. The model structure is not altered by these changes. The force elements are merely parameterised in more detail. In contrast to a constant stiffness, the lateral stiffness c_a is now provided as function of the wheel load F_z (Figure 3, left). This parameter limits the maximum transmittable lateral force for a certain wheel load and a constant slip angle. Since the lateral damping coefficient da has no significant influence on the progressions of the lateral force and the realigning torque, it is kept at a constant value of 0.6 kNs/m to provide numerical stability. To fully take the declining relation in between the lateral force F_y and the wheel load F_z into account, the dependency of the lateral coefficient of friction μ_y on the mean contact pressure is approximated by altering μ_y depending on the wheel load (Figure 3, right). This step is necessary since the tyre model – with respect to its two dimensional structure of the tyre-surface contact – is not able to take lateral changes in contact patch area into account. The variation of the coefficient of friction has an effect on the declining lateral force-wheel load relation. Additionally, all parameters are provided continuously wheel load dependent to increase model accuracy. To reduce computational time, the provision of wheel load dependent parameters can be deactivated, when only small wheel load fluctuations are expected.

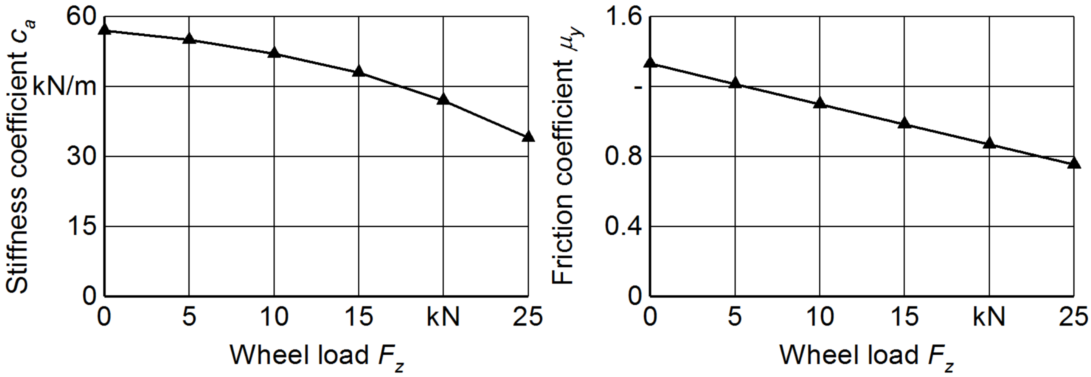

Figure 3: Axial stiffness coefficient c_a (left) and friction coefficient μ_y (right) as a function of the wheel load to reproduce the declining lateral force-wheel load-characteristic

Validation method

For validation purposes in automotive engineering, often only subjective or optical comparisons of measurements and simulation results are used. These do not present quantitatively resilient information for the quality of the simulation model. Additionally, diverse errors – i. e. regarding the magnitude and phase behaviour – can only be evaluated insufficiently or even not at all.

For an objective evaluation, error or correlation functions can be used. A correct reproduction of both the phase behaviour of a force progression or a dynamic driving condition as well as the corresponding magnitude behaviour is important for simulative investigations regarding driving dynamics. For the validation of the Hohenheim Tyre Model, a robust and simple method is used, which evaluates the phase and magnitude correlation separately at first and then combines the errors to a comprehensive error (Geers 1984). This method is also used in other sectors of automotive engineering (Krausz 2016). In comparison to a similar, constitutive method (Russell 1997), it can be revealed, that the introduced error metrics are not symmetrical regarding the choice of reference data (Sarin et al. 2010). This restriction is neglected in the following, since the measurements are always defined as reference data. Essentially more complex methods are used to validate simulation models with higher demands towards phase, magnitude but also slope behaviour – i. e. during the development and application of advanced driver assistance systems (Sarin et al. 2010). These methods do not present additional benefits for driving dynamics investigations though.

Generally, two data sets f_i, which are to be compared – in case of a simulation model validation the measurements and simulation results – are designated with the indices 1 and 2. As basis for the error metrics, the mean time value of the quadratic integrals of the simulation data φ_{11} and of the measurement data φ_{22} are generated at first for a time interval from t_1 (start of the manoeuvre) and t_2 (end of the manoeuvre) (Equation 3 and Equation 4). Additionally, the correlation φ_{12} between the simulation and measurement data is determined (Equation 5).

$$\varphi_{11} = \frac{\int_{t_1}^{t_2} f_1(t)^2 \, dt}{t_2 - t_1} \quad \text{(Eq. 3)}$$

$$\varphi_{22} = \frac{\int_{t_1}^{t_2} f_2(t)^2 \, dt}{t_2 - t_1} \quad \text{(Eq. 4)}$$

$$\varphi_{12} = \frac{\int_{t_1}^{t_2} f_1 \cdot f_2 \, dt}{t_2 - t_1} \quad \text{(Eq. 5)}$$

Geers develops a magnitude error ε_m (Equation 6) and a phase error ε_p (Equation 7) from these quantities. The metrics are designed in such a way, that they do not influence each other for the evaluation of two time dependent time series. This allows a separate evaluation of the phase and magnitude correlation with simple means. To additionally achieve an overall statement of correlation, a comprehensive error ε_c (Equation 8) is introduced.

$$\varepsilon_m = \sqrt{\frac{\varphi_{11}}{\varphi_{22}}} - 1 \quad \text{(Eq. 6)}$$

$$\varepsilon_p = 1 - \frac{\varphi_{12}}{\sqrt{\varphi_{11} \cdot \varphi_{22}}} \quad \text{(Eq. 7)}$$

$$\varepsilon_c = \sqrt{\varepsilon_m^2 + \varepsilon_p^2} \quad \text{(Eq. 8)}$$

Validation results

In a first step in the verification and validation process of the Hohenheim Tyre Model, the experiments conducted by Schlotter (2006) with a single tyre in the single wheel tester are reproduced by corresponding simulation models. Since the Pirelli TM700 520/70 R34 tyre, which was used for these investigations, is no longer available at the Institute of Agricultural Engineering, parameters of a similar Goodyear DT812 520/70 R38 tyre were adducted for first comparisons. This tyre is also mounted on the rear axle of the Fendt Favorit 509C test tractor. The two tyres have different rim diameters and slightly different overall outer diameters respectively. The transient lateral force transmission behaviour is especially dependent on the tyre wall height, which is identical for the two tyres. For this reason, measurements from Schlotter can be compared to simulation results from a very similar tyre as a first approach.

The verification of the wheel load influence on the lateral force transmission is conducted by comparing the quasi-stationary operating conditions with dynamic operating conditions (Figure 4). The declining relation between the lateral force F_y and the wheel load F_z , which was measured by Schlotter, is reproduced by the more detailed parameterisation of the lateral force elements (Figure 4, upper left). Additionally, it is noticeable, that by applying sinusoidal wheel load fluctuations with little dynamics (Figure 4, lower left), the mean transmittable lateral force is reduced in comparison to the stationary operating point (Figure 4, upper right). This behaviour was also noticed by SCHLOTTER in his experiments.

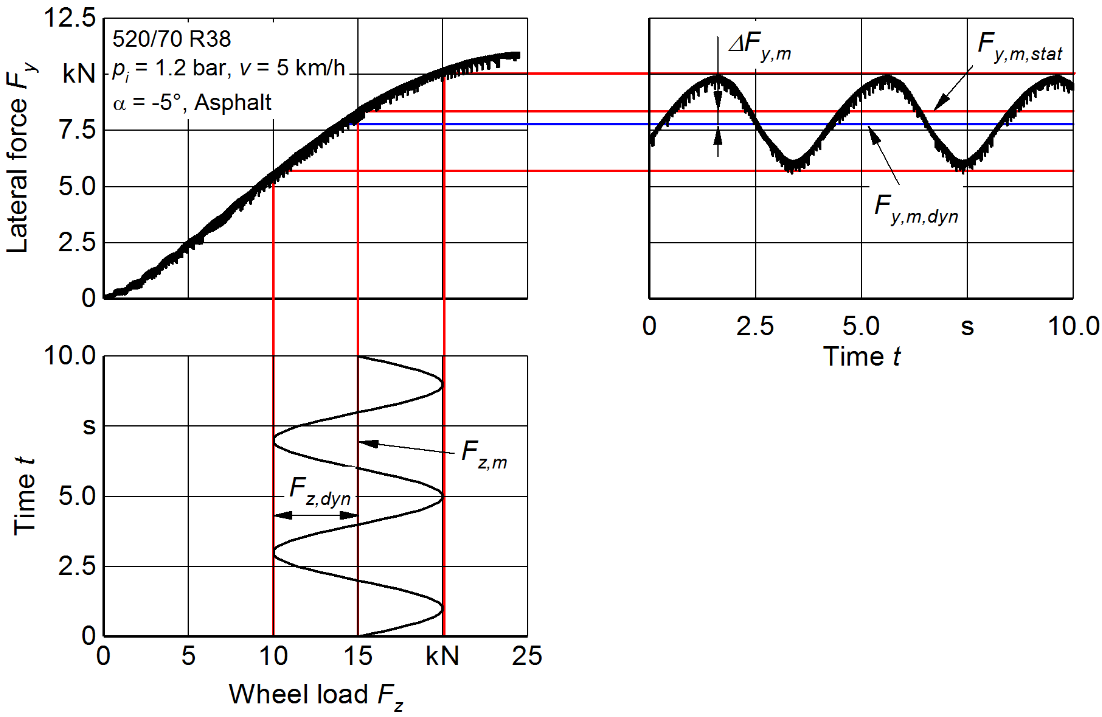

Figure 4: Transmittable lateral force reduction during dynamic wheel load excitation

In a consecutive step, the artificially impressed wheel load fluctuations – corresponding to tyre-induced wheel load fluctuations – are replaced with road-induced wheel load fluctuations. These are generated by passes over a ramp-shaped 900 mm long and 125 mm high obstacle. The wheel loads F_z (red dotted line) and lateral forces F_y (red continuous line) calculated by the tyre model are compared to measurements by SCHLOTTER (black lines) for a stationary slip angle of α = 5° and two different driving velocities of 5 and 10 km/h (Figure 5). For a better comparability with his measurements by SCHLOTTER, the results are also visualised in the distance domain. A good correlation between measurements and simulation results can especially be determined for the phase behaviour. This correlation is widely levelled though by applying the validation method described, Table 1. Large errors occur for both the magnitude and phase behaviour, resulting in comprehensive errors between $\varepsilon_{c,min}$ = 0.4559 and $\varepsilon_{c,max}$ = 0.6167. Possibly, differences in stiffness and damping properties of the two tyres compared can be identified as one reason for the large deviations. These also result in a different magnitude behaviour of the lateral force transmission.

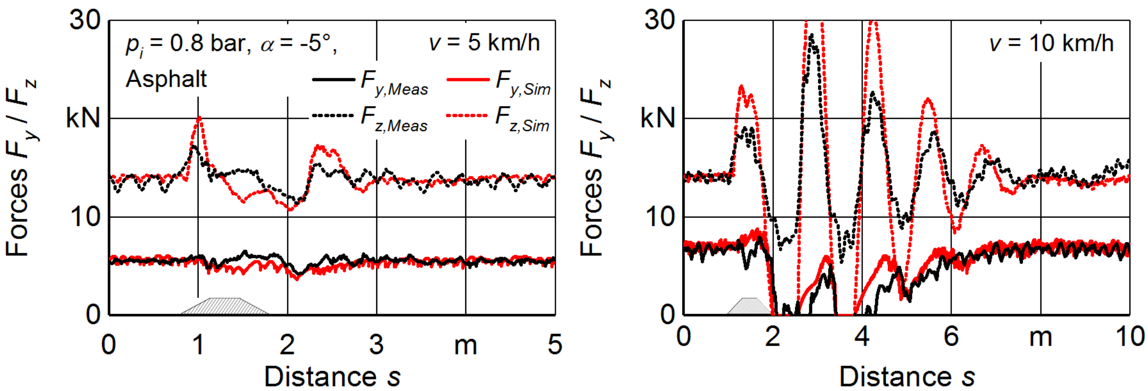

Figure 5: Velocity dependency of the wheel load F_z and lateral force F_y during passes over ramp shaped obstacles

Table 1: Comparison of simulation results and measurements for passes over a ramp-shaped obstacles with a slip angle of α = 5° and a tyre inflation pressure of p_i = 0.8 bar, maximum values highlighted in bold font

Driving velocity in km/h	Direction of force	Magnitude error ε_m	Phase error ε_p	Comprehensive ε_c
5	F_y	-0.3803	0.3619	0.5250
	F_z	-0.3718	0.3780	0.5302
10	F_y	**-0.4000**	**0.4694**	**0.6167**
	F_z	-0.2788	0.3608	0.4559

To eliminate the differences in tyre types, the validation of the tyre model regarding the lateral dynamics is concluded using the test tractor, which is fitted with Goodyear DT812 tyres with dimensions of 480/70 R24 and 520/70 R38. The Hohenheim Tyre Model was also parameterised for these tyres using single tyre test stand measurements (Witzel 2015). Exemplary, the validation of the tyre model is described for the driving manoeuvres sine steering (Figure 6) and an evasive manoeuvre (Figure 7). In each of these figures, the force progressions of the wheel loads F_z (dotted lines) and the lateral forces F_y (continuous lines) of the right front wheel (FR, upper right), the left rear wheel (RL, lower left) and the right rear wheel (RR, lower right) are visualised. Measurements are always represented by black lines, simulation results by red lines. A high correlation between measurements and simulation results is determined both optically and by using the error metrics introduced earlier. On the one hand, this finding is valid for the magnitude behaviour with a maximum magnitude error of $\varepsilon_{m,max}$ = −0.1267 for the lateral force progression of the rear left wheel during the sine steer manoeuvre as well as for the phase behaviour with a maximum phase error of $\varepsilon_{p,max}$ = 0.0622 for the lateral force progression of the rear right wheel during the same manoeuvre (Table 2). The maximum comprehensive error of $\varepsilon_{c,max}$ = 0.1401 can also be determined for the rear left wheel for a sine steer manoeuvre. The lateral force progression of the right front wheel for a sine steer manoeuvre is defined as an outlier (Table 2, values in parenthesis). The simulation data shows a time offset, which is evaluated as large magnitude error using the validation method presented.

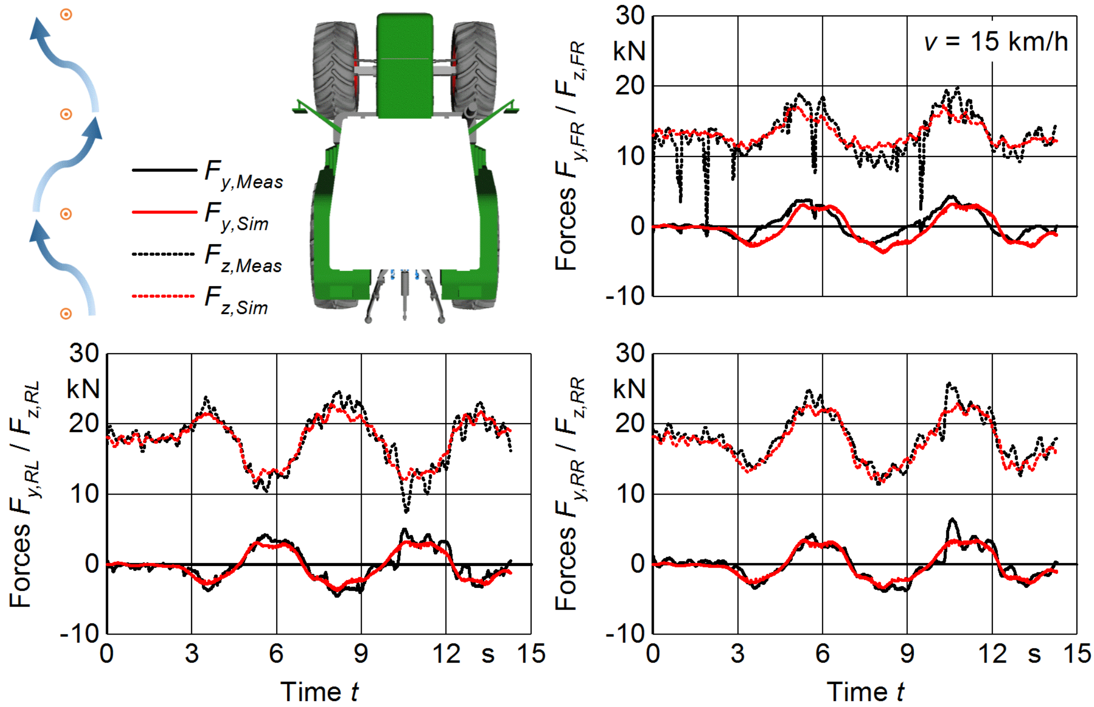

Figure 6: Comparison of the wheel loads F_z and lateral forces F_y at three wheels of the test tractor during a sine steer manoeuvre with a pylon distance of l = 10 m

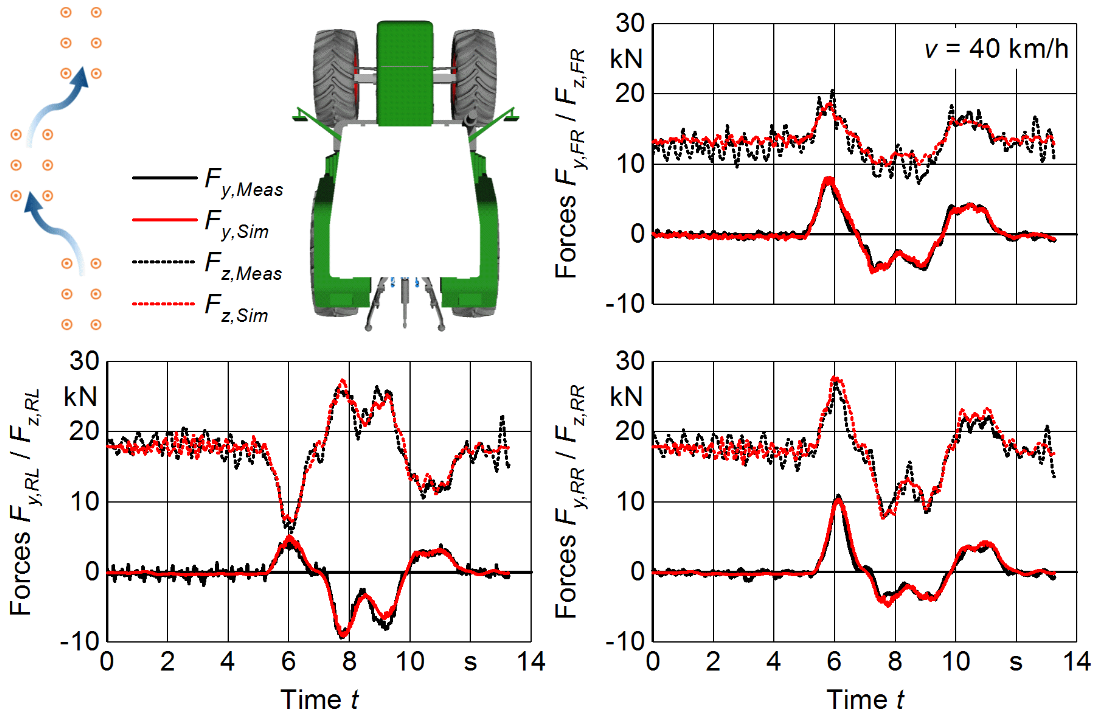

Figure 7: Comparison of the wheel loads F_z and lateral forces F_y at three wheels of the test tractor during a double-lane change manoeuvre with a driving velocity of v_D = 40 km/h

Table 2: Comparison of simulation results and measurements for road tests with the test tractor with a tyre inflation pressure of p_i = 1.2 bar, maximum values highlighted in bold font, non-reasonable data in parentheses

Fahrversuch	Wheel	Direction of force	Magnitude error ε_m	Phase error ε_p	Comprehensive error ε_c
Sine steer l = 10 m v_F = 15 km/h	FR	F_y	(0.5589)	(0.0354)	(0.5600)
		F_z	0.0070	0.0126	0.0144
	RL	F_y	**-0.1267**	0.0600	**0.1401**
		F_z	-0.0081	0.0029	0.0086
	RR	F_y	-0.1058	**0.0622**	0.1227
		F_z	-0.0342	0.0021	0.0343
Double-lane change v_D = 40 km/h	FR	F_y	0.0198	0.0126	0.0234
		F_z	0.0053	**0.0345**	0.0948
	RL	F_y	-0.0196	0.0254	0.0322
		F_z	0.0022	0.0032	0.0039
	RR	F_y	**0.0948**	0.0158	**0.0961**
		F_z	0.0166	0.0037	0.0170

Conclusions

The tyre model, which was developed at the Institute of Agricultural Engineering at the University of Hohenheim, is applicable for both driving dynamics and driving comfort investigations. This contribution concludes its verification and validation regarding driving dynamics originally started by Witzel (2015). In a first step, experiments conducted in earlier research with respect to the influence of wheel load fluctuations on the lateral force transmission for single tyres (Schlotter 2006) were reproduced using adequate simulation models. These investigations show a general correlation between measurements and simulation results. Partially, larger magnitude and phase errors occur, which are mainly based on the different characteristics of two slightly different tyres used for the comparison. Subsequently, road tests with a test tractor were conducted. The validation of the lateral force transmission is undertaken using dynamic operating conditions during sine steer and evasive manoeuvres. A high correlation between measurements and simulation results is determined for the whole vehicle investigations. This correlation is determined using resilient error metrics. Separate error metrics for the phase and magnitude correlation additionally allow a differentiated validation process. The investigations on the whole vehicle showed a maximum magnitude error of $\varepsilon_{m,ma}$ = −0.1267, a maximum phase error of $\varepsilon_{p,max}$ = 0.0622 and a maximum comprehensive error of $\varepsilon_{p,max}$ = 0.1401.

The Hohenheim Tyre Model is also applicable for driving comfort investigations. Its validity was already shown by Witzel (2015) for passes over discrete obstacles. In further research at the Institute of Agricultural Engineering, the tyre model will first be validated on complex road surfaces like the standard roadway defined in the EU regulations regarding the evaluation of the driving seat (DV (EU) 1322/2014). Subsequently, virtual investigations regarding the driver's isolation from whole body vibration will be conducted as defined in the Delegated Regulation1322/2014 (DV (EU) 1322/2014) or in the ISO 2631/1 international standard (ISO 2631/1 1985). Based on a new Fendt 313 Vario test tractor available at the Institute of Agricultural Engineering, the influence of different undercarriage systems of a tractor built in standard architecture on driving safety and driving comfort will be investigated. A simulation model of this test tractor will be used as virtual development platform. Ultimate goal of these investigations is the determination of the potential of standard tractors regarding driving safety and driving comfort in comparison to tractors built in trac-architecture, which have been extensively investigated at the TU Berlin (Hoppe 2007).

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