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Parametrization of an Integrated Superelastic Tyre Model and Application in the Overall Vehicle Simulation

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An industrial truck's driving and tipping stability as well as the vibration load on the driver are considerably influenced by the tyre properties. For the detailed assessment of dynamic vehicle behaviour, multibody simulations are used which can provide precise information on vehicle behaviour. In order to realistically reproduce the tyre properties during these simulations, a reliable and valid tyre model is required which transmits the occurring forces and moments into the vehicle structure. It is of particular interest that the model is able to map the special nonlinear properties of industrial truck tyres with regard to vertical and lateral dynamics.

This paper presents a three-dimensional, nonlinear tyre model for superelastic tyres that can be used to examine the driving dynamics of industrial trucks during different driving manoeuvres. After a short description of the basic model structure, the parametrization of the model is discussed, adapting the theoretical approach to the real tire behavior. In addition to experimental data, simulation results are also used for this, which are generated by means of a self-developed structural mechanics approach. The tyre model is then verified by comparing the simulation and measurement results and its validity in various areas of application is examined.

[Keywords: Superelastic tyre, Tyre model, Multibody simulation, Vehicle dynamics simulation, Finite element method]

Die Fahr- und Kippstabilität sowie die Schwingungsbelastung auf den Fahrer werden bei Flurförderzeugen erheblich durch die Reifeneigenschaften beeinflusst. Für die detaillierte Beurteilung des dynamischen Fahrzeugverhaltens kommen Mehrkörpersimulationen zum Einsatz, durch die genaue Aussagen über das Fahrzeugverhalten getroffen werden können. Um die Reifeneigenschaften im Rahmen dieser Simulationen realitätsnah abbilden zu können, wird ein zuverlässiges und valides Reifenmodell benötigt, welches die auftretenden Kräfte und Momente in die Fahrzeugstruktur einleitet. Hierbei ist von besonderem Interesse, dass die speziellen nichtlinearen Eigenschaften von Flurförderzeugreifen hinsichtlich der Vertikal- und Querdynamik durch das Modell abgebildet werden können.

In diesem Beitrag wird ein dreidimensionales, nichtlineares Reifenmodell für Superelastikreifen vorgestellt, mit dem die Fahrdynamik von Flurförderzeugen bei unterschiedlichen Fahrmanövern untersucht werden kann. Nach einer kurzen Beschreibung des grundsätzlichen Modellaufbaus wird auf die Parametrierung des Modells eingegangen, wodurch der theoretische Ansatz an das reale Reifenverhalten angepasst wird. Neben experimentellen Daten werden hierfür zusätzlich Simulationsergebnisse herangezogen, welche durch einen eigenentwickelten strukturmechanischen Ansatz generiert werden. Im Anschluss wird das Reifenmodell durch den Vergleich zwischen Simulations- und Messergebnissen validiert bzw. verifiziert, wodurch die Gültigkeit in verschiedenen Einsatzgebieten überprüft wird.

[Schlüsselwörter: Superelastikreifen, Reifenmodell, Mehrkörpersimulation, Fahrdynamiksimulation, Finite-Elemente-Methode]

1 INTRODUCTION

Tyres play an essential role in vehicle dynamics as all forces and the resulting moments act on the vehicle via the tyre/ground contact. In industrial trucks, the vibration and impact load transmitted to the vehicle chassis is only absorbed by the tyres. In addition, tyre properties significantly influence a vehicle's driving and tilting stability. These aspects underline the importance of simulation models capable of precisely predicting in advance a tyre's effect on vehicle behaviour. In addition, extensive and profound knowledge of the dynamic behaviour of different types of tyres makes it possible to choose the right tyre for any vehicle model and application during both initial outfitting and spare tyre retrofitting. Particular focus is placed on designing a model that is versatile and easy to implement into an overall vehicle model. Two previous research projects

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have already yielded separate tyre models describing vertical dynamics [Oh17] and lateral dynamics [Bus15]. Stepanyuk [SKB16a] used the lateral dynamics model developed by Busch based on artificial neural networks and transferred it into an analytical approach, which greatly facilitated the application of the model in multibody simulation. The model presented in this paper merges these separate model types in a modified form into an expanded integrated tyre model. With this three-dimensional, nonlinear tyre model, it is possible to carry out a wide range of investigations into vehicle dynamics and comfort.

2 Description of the Tyre Model Structure

This paragraph will outline the theoretical approach and the implementation of the tyre model in MSC ADAMS. Furthermore, the physical principles behind this approach are explained and related to the model parameters.

2.1 Tyre Model Requirements

For the formulation of a suitable model type, the requirements concerning the model have to be defined in detail. As already mentioned, the objective is to expand the planar approach developed by Oh [Oh17] and transform it into a three-dimensional model. Thus, the model must be able to correctly map all three contact forces in x-, y-, and z-direction and the resulting moments. Figure 1 is a schematic illustration of the contact forces acting on the tyre's contact patch due to rolling and the associated vehicle velocity v_x, the slip angle α during cornering, and the vehicle mass. The contact patch is the contact area that results from the vertical load and thus establishes the contact between tyre and ground. During operation, industrial truck tyres can have very high slip angles and slip rates, showing a specific lateral dynamics behaviour that the model must take into account. The model can therefore serve to address not only vehicle dynamics but also comfort-related issues that might arise, for example, when the tyre passes over obstacles.

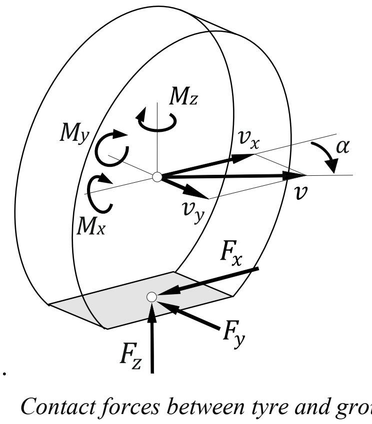

Figure 1. *Contact forces between tyre and ground and resulting moments*

2.2 Structure of the Tyre Model

The basic structure of the tyre model is a spokes model which is expressed as a separate multibody system. This structure is complemented by an analytical approach that makes it possible to represent the build-up of dynamic lateral force. The multibody system, designed to map the mechanical properties of the tyre, is mainly based on the Hohenheim Tyre Model (cf. [Fer08] and [Wit15]) and also partially draws on the model developed by Oh [Oh17]. As mentioned above, Oh's model only addresses vertical dynamics and thus regards the tyre as a plane, while the Hohenheim Tyre Model is three-dimensional. The flexible rubber layer of the tyre is idealized in the radial, axial and tangential directions using an array of multiple spring-damper systems. The tread, which establishes the contact between tyre and ground, is modelled by a chain of rigid bodies, the contact elements. In addition to the translational components, the force elements also contain rotational components around the x- and y-axis, which represent the adaptation of the contact area to the road surface. Thus only the rotational degree of freedom around the z-axis is blocked. The contact elements are linked to each other and to the rim via the spring-damper elements. The interaction between tyre and ground is described using a contact condition between the contact elements and the ground surface geometry, which allows a continuous rolling contact to be modelled. The contact condition also encompasses a separate definition of the model's separate frictional properties in the longitudinal and the lateral direction. This way, the anisotropy in horizontal force transmission resulting from the tyre profile and the rim geometry can be taken into account. A schematic illustration of the model's structure is shown in Figure 2. The drawing on the left is a side view of the spokes model, while the one on the right is a sectional view describing the axial behaviour. In addition, the illustration includes the parameters that determine the mechanical behaviour of the tyre.

The tangential stiffnesses c_{xi} and dampings d_{xi} ensure that the tyre can transmit the drive torque and the braking torque to the ground surface. The axial components c_{yi} and d_{yi} describe the build-up of cornering force. The vertical dynamics behaviour is mainly characterized by the radial parameters c_{zi} and d_{zi}. The geometric dimensions of the tyre can be varied by changing the size of the contact elements and/or their position relative to the rim, which means that the model has a complete parametric structure. Discretization can be adapted via the number of elements N_{ce}. The number is determined by the width of an element in relation to the tyre radius. For the following simulations the tyre is discretized by 72 elements. Previous studies with 108 elements have shown that increasing the number of elements does not significantly change the results.

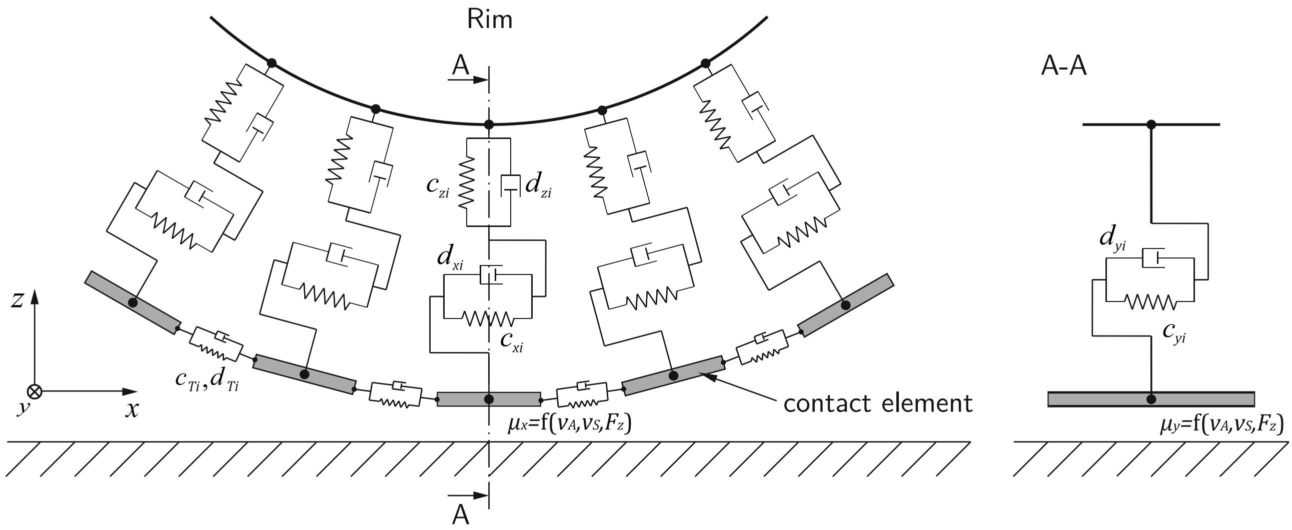

Figure 2. *Schematic drawing of the model's structure with the corresponding mechanical parameters*

Force transmission in the three spatial directions of the spokes is implemented in ADAMS via a nonlinear three-dimensional force element. The nonlinear stiffnesses and dampings are considered in relation to the deformations or deformation rates in the force elements. The friction contact is implemented via a Coulomb friction model. In this model, the friction coefficients μ_x and μ_y are computed in relation to the adhesion and sliding velocity (v_A, v_S) as well as the wheel load F_z.

As already indicated, special attention must be paid to the modelling of the dynamic lateral force. As steering manoeuvres of industrial trucks can result in high slip rates, the tyre displays damping effects that can be seen as hysteresis in the lateral force [SKB16a]. To map this effect, a dynamic portion depending on the slip rate and the vehicle velocity is added to the lateral force component of the model described above. This portion follows from the analytical approach taken by Stepanyuk and is additionally implemented into the multibody system of the tyre as an external force. The calculations used within the model to precisely determine the tyre forces will be addressed in the following section.

As mentioned, the model is intended for use within the MSC ADAMS MBS software [MSC17]. In order to design the model as rapidly as possible, the structure will be implemented automatically by an internal programming language [MSC16]. The integrated code will create the model depending on the defined mechanical and geometric parameters in the desired position of the simulation environment, thus ensuring easy handling.

2.3 Calculating the Tyre Forces

The tyre forces in the three directions in space are transferred into the model via the contact elements in the contact patch. Adding up the spring and damping forces in the relevant directions will yield F_x, F_y, and F_z between the contact patch and the rim. To determine these forces, the values for deformation $u(t)$ and deformation rate $\dot{u}(t)$ are needed in addition. These values are obtained from the relative motion of two coordinate systems located on the rim and on the respective contact element.

Vertical force transmission, which is the main influencing factor for the tyre's deflection behaviour in the static and in the dynamic state, is determined as follows within the model:

$$F_z(t) = \sum_{i=1}^{N(t)} c_{zi} u_{zi}(t) + d_{zi} \dot{u}_{zi}(t). \qquad (1)$$

The vertical force generated by the rotation of the contact elements or by the expansion of the contact patch is comparatively low and therefore has no significant influence on the vertical force transmission of the model. In the calculation, $N(t)$ is the number of contact elements in the contact patch, which can vary over time and is continuously calculated by the contact model during the simulation. It can be assumed that all elements are deformed to the same extent or at the same rate. This assumption is justified by the angle between the spokes being very small, so that there are no major deviations between global and local deformations or deformation rates. Thus, the curves for a spoke are obtained from the nonlinear functions of tyre stiffness $c_z(u_z, \dot{u}_z)$ and tyre damping $d_z(u_z, \dot{u}_z)$ during the simulation.

$$c_{zi} = \frac{c_z(u_z, \dot{u}_z)}{N(t)} \quad \text{bzw.} \qquad d_{zi} = \frac{d_z(u_z, \dot{u}_z)}{N(t)} \qquad (2)$$

The stiffness and damping splines are implemented in dependence of the deformation or deformation rate so that the nonlinearity regarding these parameters is taken into account.

The transmission behaviour of the longitudinal force consists of two components. Besides the spring-damper force

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between the rim and the contact element, $F_x(t)$, the force of the coupling elements between the contact elements, $F_T(t)$, also needs to be considered. The sum of these forces yields the total longitudinal force:

$$F_x^{tot.}(t) = F_x(t) + F_T(t) \tag{3}$$

The two components can be determined as follows:

$$F_x(t) = \sum_{i=1}^{N(t)} c_{xi} u_{xi}(t) + d_{xi} \dot{u}_{xi}(t) \tag{4}$$

$$F_T(t) = \sum_{i=1}^{N(t)} c_{Ti} u_{Ti}(t) + d_{Ti} \dot{u}_{Ti}(t). \tag{5}$$

The sum of the longitudinal force constitutes the reaction force to the friction force in the direction of travel and is essential to properly map the longitudinal slip. In this context, the stiffness c_x is also a spline that is dependent on deformation in the x direction. The damping coefficient d_x is expressed as a constant as simulations have shown that its effect is very limited. The stiffness c_T and the damping d_T in the contact area are also described by linear factors. The method used to distribute the stiffness and the damping over the spokes in the contact patch is identical to that used for the vertical force (see equation (2)).

As mentioned above, the lateral force comprises a quasi-static and a dynamic component. Quasi-static force transmission is also determined via the sum of the spring and damping forces in the y direction. The definition of stiffness and damping is identical to that of the tyre longitudinal force.

$$F_y^{stat}(t) = \sum_{i=1}^{N(t)} c_{yi} u_{yi}(t) + d_{yi} \dot{u}_{yi}(t) \tag{6}$$

As mentioned, the dynamic approach is based on the approach developed by Stepanyuk [SKB16a] and will only be briefly outlined in this paper. The difference between this and Stepanyuk's model is that, owing to the basic structure of the model, the quasi-static portion is not determined by means of an analytical function but in the model using the above equation. This makes parametrization easier as the force only depends on the mechanical parameters describing stiffness, damping and friction.

Various research (e.g. [Sch05]) has shown that the build-up of lateral force at a highly dynamic change of the slip angle can be represented by a PT_1 transfer function. Thus, the general formula for the dynamic model can be described as follows:

$$T\,\dot{F}_y^{dyn}(t) + F_y^{dyn}(t) = F_y^{stat}(t), \tag{7}$$

T represents the time constant, which determines the value of the hysteresis and is derived from measurement data. The expression of the dynamic lateral force can be derived from equation (7).

$$F_y^{dyn}(t) = F_y^{stat}(t) - T\,\dot{F}_y^{dyn}(t) \tag{8}$$

Assuming a discrete-time system with an increment of Δt and the formulation of the backward difference quotient, the following formula is obtained:

$$F_y^{dyn}(t) = F_y^{stat}(t) - \frac{T}{\Delta t}\left(F_{y,n}^{dyn}(t) - F_{y,n-1}^{dyn}(t)\right). \tag{9}$$

This yields the formula used to calculate the lateral force during dynamic wheel adjustment.

$$F_y^{dyn}(t) = \frac{1}{1+\frac{T}{\Delta t}}\left(F_y^{stat}(t) + \frac{T}{\Delta t} F_{y,n-1}^{dyn}(t)\right). \tag{10}$$

This approach is implemented into the multibody system of the tyre, thus allowing a calculation of the lateral force during highly dynamic vehicle manoeuvres.

The next step is to develop a contact model that properly represents the frictional properties of the tyre. In this context, it is highly important to note that the friction coefficient can be defined as a function of the relative velocity and the direction of load. The Coulomb friction model implemented in ADAMS requires the adhesion and sliding friction coefficients (μ_H, μ_G) and the corresponding velocities (v_H, v_G) to be entered. This allows the friction characteristics during locking or slipping of the tyre to be adapted. The transition between adhesion friction and sliding friction is represented by a nonlinear function. The graph of this function is shown in Figure 3a).

However, this friction model only allows the definition of friction coefficients and relative velocities that are identical in all spatial directions. This assumption is not correct, though, when it comes to modelling the friction contact of tyres. Therefore, the model must be expanded. In order to take the anisotropic tyre characteristics into account, the model is modified to allow definitions of separate friction coefficients for the longitudinal and the lateral direction. The limits of the transmissible friction forces in the two directions of force are represented by a circle of forces warped into an elliptic shape (see Figure 3b) [Wit15].

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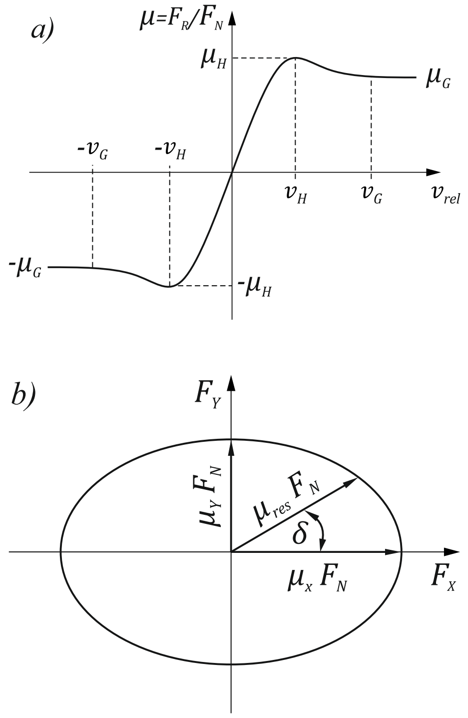

Figure 3. *Differentiation of the adhesion and sliding friction coefficients (a); Circle of frictional forces describing anisotropic friction (b)*

The resultant friction coefficient μ_{res} for the combination of a longitudinal force F_x and a lateral force F_y can thus be determined by means of an ellipse formula after the friction coefficients μ_x and μ_y have been defined.

$$\mu_{res} = \frac{\mu_y}{\sqrt{1 - \varepsilon^2 \cos^2 \delta}} \tag{11}$$

In this context, ε represents the numerical eccentricity and is determined by:

$$\varepsilon = \frac{\sqrt{{\mu_x}^2 - {\mu_y}^2}}{\mu_x}. \tag{12}$$

The angle of force δ describes the correlation between the longitudinal and the lateral force and is determined as follows:

$$\delta = \tan^{-1} \frac{F_y}{|F_x|}. \tag{13}$$

Using this approach, the anisotropy of force transmission can be mapped and the friction coefficient can be represented in a universal valid manner.

3 Parametrization of the Tyre Model

Adapting the presented model approach to the actual behaviour of the tyre requires determining the aforementioned tyre-specific model parameters. This is accomplished by using both experimental data and structural mechanics analyses as some parameters are difficult to ascertain by experimental means only. The structure of the structural mechanics model and its application in parameter identification has already been described in great detail in [PB18]. This topic will therefore not be addressed to the same extent in this paper.
The tests described below as well as the validations following in the next section were all carried out at ambient temperature. It should be noted that the mechanical properties of tyres change significantly with increasing temperature. The stiffness, damping and friction coefficients decrease considerably at higher temperatures or under long load periods. Therefore, the identified parameters are not valid for strongly heated tyres.

The radial stiffness and damping of the tyre are determined by conducting tests with a non-rolling wheel. For this purpose, a hydropulse test stand is fitted with a special tyre mounting construction (see Figure 4). A servo-hydraulic cylinder continuously presses the centerline of the tyre against a platform and the resultant force is measured. Both quasi-static and dynamic tests are possible. This allows assessing the dependence of stiffness and damping on wheel load and deflection rate.

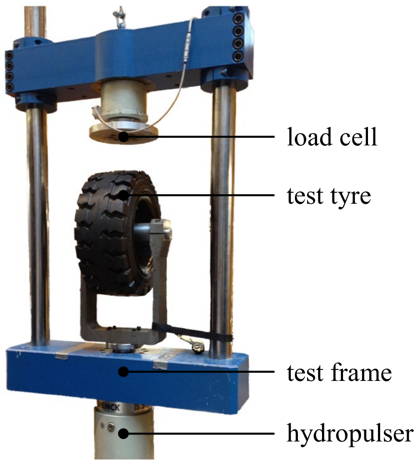

Figure 4. *Hydropulse test stand used to examine a tyre's vertical deflection behaviour*

To determine the stiffness, deflection characteristic curves are established for different deflection rates. The results indicate that the influence of the deflection rate is very limited. Therefore, the nonlinear spline of the stiffness can be derived directly from the measured force-deflection curve. It is determined and implemented into the model as follows:

$$c_z(u_z) = \frac{\mathrm{d}F_z(u_z)}{\mathrm{d}u_z}. \tag{14}$$

[ODF+13] and numerous studies have shown that damping is far more dependent on the rate of loading than stiffness.

Thus, the tyre is loaded by a dynamic force and the resulting deflection signal is recorded by sensors.

$$F(t) = F_m + A \sin(2\pi\ f\ t). \qquad (15)$$

By varying the average wheel load F_m, the amplitude A or the frequency f, it is possible to determine the damping in relation to these factors. The test stand is idealized as a single-degree-of-freedom system, which allows the damping to be determined using the established stiffness. Since frequency and wheel load are the most significant influencing factors, they are taken into consideration when the damping is implemented into the model.

All other parameters are determined using a industrial tyre test stand of the MTL, which was specifically designed for testing industrial truck tyres. It is an external drum test stand that can be used to measure longitudinal and lateral force transmission in relation to various parameters. Since the design of the test stand has already been described in numerous publications (cf. [BB12], [Bus15]), it will not be discussed in detail in this paper.
The longitudinal and lateral stiffness of the tyre is also determined using a quasi-static test method, since it can be assumed that it will not be significantly influenced by the rate of loading. To determine the longitudinal stiffness, the tyre is pressed against the drum in a braked state and at a predefined wheel load. Then, the drum is rotated very slowly so that the tyre is deformed longitudinally. The rotation of the drum and the measured force signal yield the nonlinear force-deflection curve. In order to determine the lateral force-deflection behaviour, the tyre only has to be turned by 90°. Based on these correlations, the nonlinear splines of lateral and longitudinal stiffness can be determined using the above approach (see equation (14)) and can then be integrated into the model. Unfortunately, due to the drum's insufficient dynamic range, this method cannot be used to determine the dampings in the longitudinal and lateral direction. However, as briefly mentioned above, these parameters do not significantly influence the model behaviour and will therefore be defined in the framework of the model verifications.

The friction parameters μ_x, μ_y and the corresponding velocities v_A. v_S are derived from the measured curves of longitudinal force versus slip and of lateral force versus slip angle. These curves can be used to derive the function of the friction coefficient in the respective direction depending on the relative velocity. In this context, the friction coefficients are determined as follows:

$$\mu_x = \frac{F_x(S_x)}{F_z} \quad , \quad \mu_y = \frac{F_y(\alpha)}{F_z} \qquad (16)$$

S_x is the longitudinal slip and α is the slip angle. The relative velocity can be derived from the difference between the longitudinal velocity of the drum and that of the tyre. The graphs of these functions then yield the adhesion and sliding friction coefficients and the respective relative velocities, which can then be implemented into the friction model in relation to the wheel load.

The last step of model parametrization is to determine the parameters c_{Ti} and d_{Ti} of the tangential spring-damper system. They are located between the contact elements and are thus responsible for the correct representation of the contact patch. Determining these parameters by experimental means is extremely difficult. Therefore, a structural mechanics model based on the finite element method is used for this purpose. This model simulates static and dynamic deflection of the tyre and yields the deformation and deformation rate of the contact patch in the longitudinal direction. By implementing this data in ADAMS, the stiffness and the damping can be optimized in such a way that the requirements

$$u_T^{FEM}(t) = u_T^{MKS}(t) \quad \text{bzw.} \quad \dot{u}_T^{FEM}(t) = \dot{u}_T^{MKS}(t) \qquad (17)$$

are fulfilled.

4 Validation and Verification of the Tyre Model

In this section, the integrated tyre model shall be verified and validated by means of experimental data to prove its validity. Different sets of measurement data are used for this purpose, so that all the different aspects of the model can be examined. The test tyre is a 180/70-8 from Solideal, which is a common variant for vehicles with a load capacity of 2 t.

The first step is to compare the vertical force transmission of the rolling tyre with simulation results. In a previous project [ODF+13], the drum test stand was fitted with an additional component that permits an analysis of the vertical vibration behaviour of the tyre. To this end, the tyre is integrated into a test mount that is pivoted on one side and therefore able to perform a free, almost linear movement in the vertical direction. The tyre is supported at the topmost point of the drum and thus assumes the defined speed. By placing a cleat on the drum, it is possible to separately analyse the vertical dynamics and thus the transmission behaviour of the vertical force when the tyre is passing over the obstacle. Displacement and acceleration sensors on the pivoted mount serve to measure the vertical kinematic behaviour. In addition to varying the speed, it is also possible to change the wheel load by adding weights to the test mount.
By creating a detailed multibody model of the test stand, which also integrates the tyre model, the measurement results can be compared with the simulation results. This way, the vertical force behaviour can be validated. To this end, the acceleration signal is used at a predefined position. While the frequency of the resultant vibration indicates the stiffness of the tyre, the decay behaviour provides information on the damping properties. Figure 5 shows the

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measured and the simulated vertical acceleration when the tyre is rolling over a cleat of 5 mm in height. The results show that the model can accurately reproduce both the frequency of the vibration and the decay behaviour.

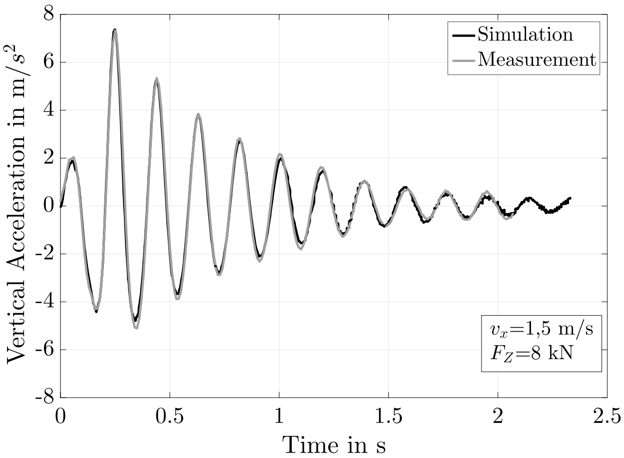

Figure 5. *Validation of vertical force transmission behaviour for a vertical load of F_z =8 kN and a velocity of v_x=1,5 m/s*

Figure 6, depicting the acceleration signal for a different wheel load and vehicle velocity, also shows close concurrence of the curves. The displayed measurement curves are the mean value of five measurement runs, which guarantees the reproducibility of the results. The comparison between measurement and simulation at other wheel loads and velocities showed a similarly high agreement.

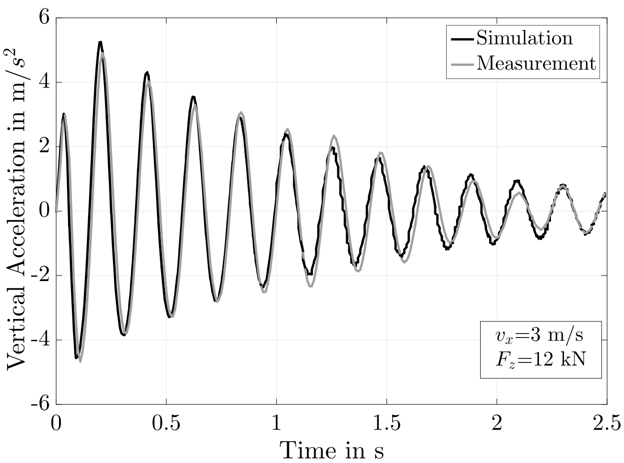

Figure 6. *Validation of vertical force transmission behaviour for a vertical load of F_z =12 kN and a velocity of v_x=3 m/s*

Thus it could be demonstrated that the model is able to correctly map vertical force transmission at different wheel loads and velocities. The result also confirms the viability of the above-described method used to determine the stiffness and damping parameters in the vertical direction.

The next step is to test the validity of the model with respect to longitudinal force transmission. The stationary transmission behaviour of the force in this direction is characterized by curves representing the longitudinal friction coefficient in relation to the longitudinal slip. Usually, two different types of slip are considered in this context – drive slip and brake slip. Since in the real-life operation of industrial trucks the transmission of the braking torque has a higher relevance, only the slip during braking is examined in this context. In order to examine the force transmission behaviour during braking, the tyre is tested and measured in the drum test stand using predefined wheel loads and velocities. The tyre is pressed against the rotating drum at a defined wheel load and is then decelerated from a free-rolling state to a state of complete locking. The measured longitudinal velocities of the drum v_D and the tyre v_T can then be used to determine the longitudinal brake slip S_{Bx}:

$$S_{Bx} = \frac{v_D - v_T}{v_D}. \tag{18}$$

The longitudinal friction coefficient μ_x can be derived from the measured longitudinal force F_x and the wheel load F_z (see equation (16)). This coefficient then serves to accurately map the longitudinal force transmission behaviour as a function of the brake slip. The resulting curve shows both the maximum force transmission and the transition between sliding and adhesion friction.

Figure 7 shows a comparison of simulation and measurement results, using $F_z = 10$ kN as an example. It is apparent that the curve simulated by the model closely matches the curve of measured values.

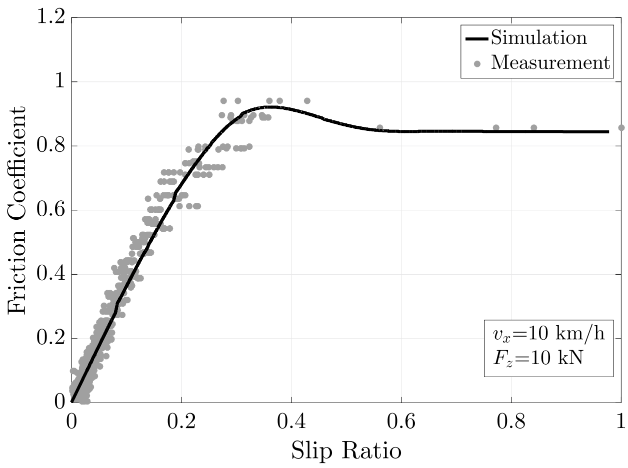

Figure 7. *Validation of the stationary longitudinal force transmission behaviour*

It is especially in the area where the highest longitudinal force is transmitted ($S_B \approx 0.35$) and which thus is of particular interest, there is a high degree of agreement between simulation and measurement data. Unfortunately, the measuring technology of the test stand is not precise enough to map the transitional area between sliding and adhesion friction in sufficient detail.

After successful verification of the model's validity regarding the transmission of vertical and longitudinal force, the

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final step is to examine the behaviour of the quasi-static and the dynamic lateral force. Experimental testing is once more conducted using the drum test stand, which permits a comprehensive analysis of lateral force transmission. In addition to the wheel load and the velocity, it is also possible to vary the slip angle rate $\dot{\alpha}$. Varying the slip rate makes it possible to differentiate between examining the quasi-static behaviour and the dynamic behaviour.
Figure 8 shows a comparison between the simulated and the measured lateral force in relation to the slip angle at a slip rate of 1°/s and a velocity of 12 km/h. The curves match very closely at all wheel loads examined. This means that the presented model approach is capable of mapping the quasi-static lateral force very precisely without the implementation of an additional analytical model.

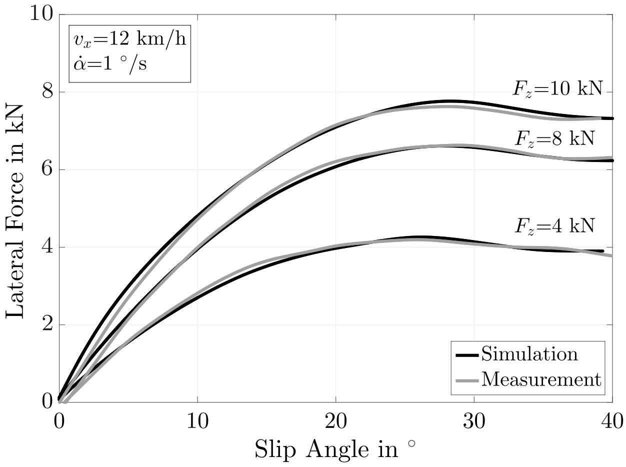

Figure 8. *Validation of quasi-static lateral force transmission behaviour at different wheel loads*

As described above, the lateral force characteristic curve shows damping effects in the form of hysteresis when the dynamic portion is increased. For the purpose of validating the lateral dynamics model approach, Figure 9 shows the simulated and the experimental characteristic curves at an increased slip angle and different wheel loads.

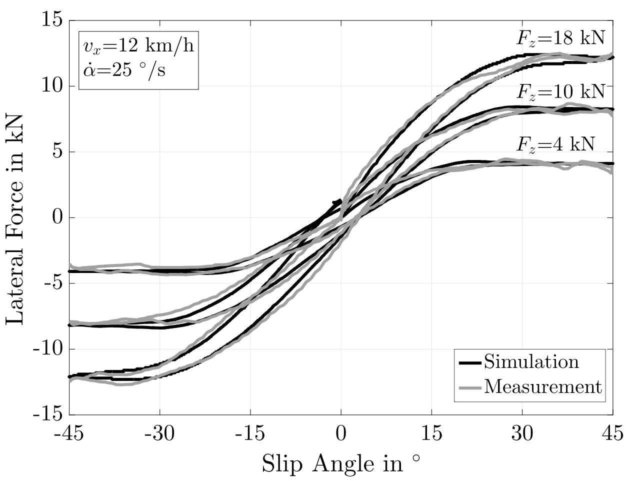

Figure 9. *Validation of dynamic lateral force transmission behaviour at different wheel loads*

The comparison also shows a good agreement. Aside from the wheel load, the velocity and the slip rate were also varied in additional simulation runs and the results were compared with measured data. The validity of the model was also proved in these respects.

Validation by means of the various test stand results has shown the three-dimensional model approach to have a high quality in all three directions. In addition, the high degree of concurrence between simulation and measurement data indicates that the model is highly accurate in terms of parameter identification.

However, it should be noted that the mechanical properties of the tyre are influenced by the internal temperature and the utilization level of the tyre. The extent to which these influences are reflected in the recorded measurement curves was not examined in this paper.

5 Application in Overall Vehicle Simulation

After the validity of the tyre model has been successfully demonstrated by means of the various test stand results, the next step is to integrate the tyre model into an overall vehicle simulation. In addition to cleat tests conducted to verify whether the tyre model accurately transmits shocks and vibration behaviour into the vehicle chassis, these simulations also encompass various cornering manoeuvres to examine the proper build-up of lateral force.

Vertical force transmission of the tyre into the vehicle is analysed using various cleat test runs. The heights of the cleats and the vehicle velocities are identical to those used in the drum test stand for model validation. Once again, the vertical acceleration signal, which is picked up near the front axle, is used to compare the measurement and simulation results. The multibody system of the forklift is derived from detailed CAD data provided by the manufacturer. The weight and moments of inertia have been determined in advance and integrated into the simulation model (cf. [SKB16b]). This model is merely a rigid body model, which considers the relative movements of the separate components to a limited extent only. This idealization has a significant effect on the representation of the vibration behaviour and has to be factored in when the measured and the simulated data is compared. Since lifting a load leads to greatly intensified relative movements between the components on the forklift mast, the following validations will be using values measured without a load.
In order to minimize external influences and thus increase repeatability of the measured data, several test runs are conducted and averaged for the purpose of validation. By way of example, Figure 10 depicts measured and simulated acceleration signals at a velocity of 2 m/s and 3 m/s without a load. At both velocities, there is a high degree of concurrence. In particular, the acceleration peaks that occur immediately after the tyre has hit the cleat are represented

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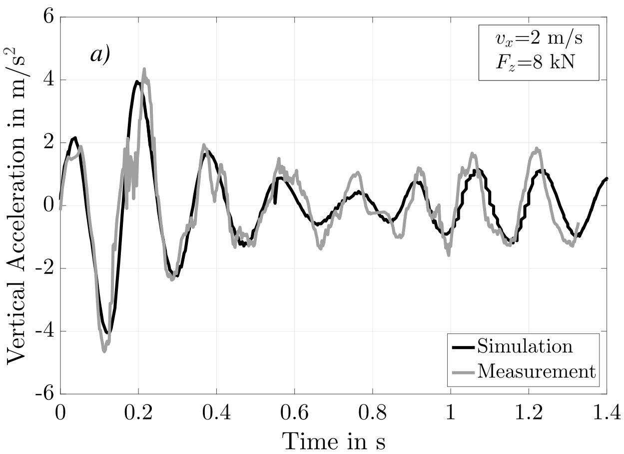
very accurately. A further increase in velocity greatly intensifies the influence of the vibrating components of the forklift mast and thus causes the vibration signal generated by the tyres to be superimposed. This means that the appropriate data cannot be compared.



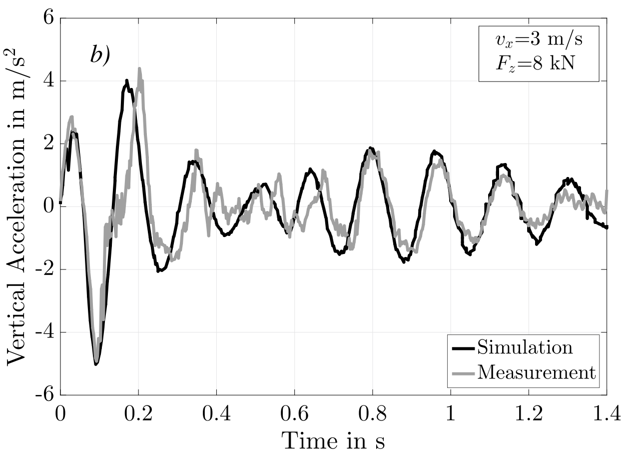


Figure 10. *Comparison of measured and simulated vertical acceleration at v_x=2 m/s (a) and v_x=3 m/s (b)*

The comparison thus confirms that the tyre model correctly transmits vertical force into the vehicle chassis. Although the model structure is relatively simple and external influences have only been factored in to a limited extent, it is possible to achieve high degrees of concurrence between measurement and simulation results. Deviations are largely due to the vehicle model, which should have a higher degree of detail.

To verify whether the stationary and the dynamic lateral force are also properly transmitted in the context of an overall vehicle simulation, different cornering manoeuvres are simulated and compared with experimental data. In order to validate the model in a stationary state, the curves of the trajectories in the vehicle's centre of gravity during stationary circling are compared. In addition to different steering angles and thus circle diameters, different vehicle velocities are examined. Figure 11 shows an example of a comparison between a measured and a simulated trajectory generated at a velocity of v_x = 3.2 m/s. The simulated trajectory closely matches the measured one, which confirms that the stationary model for lateral force transmission has been correctly implemented and parametrized.

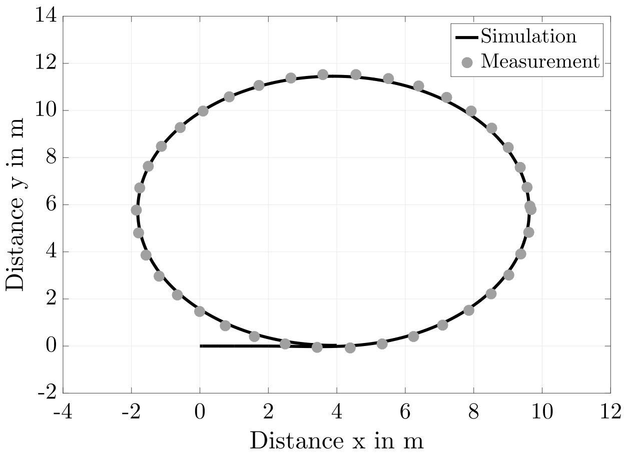


Figure 11. *Comparison between a measured and a simulated trajectory during stationary circling*

To validate the dynamic lateral force transmission, a simulated and an experimental L test are carried out. The concurrence between the trajectories generated by the simulation and the experimental run is assessed. In this test, the vehicle is initially accelerated on a straight course up to a defined velocity. Then, a 90-degree turn is initiated via an abrupt steering manoeuvre. This sudden manoeuvre results in high slip rates, which makes it particularly suitable for testing the dynamic model. Figure 12 provides a comparison of the centre-of-gravity trajectories generated during the measured and the simulated L tests. The vehicle has a velocity of 4.5 m/s and initiates the turn at a slip rate of 36.6 °/s. The comparison of the trajectories shows that they are almost identical, indicating that the model functions as intended.

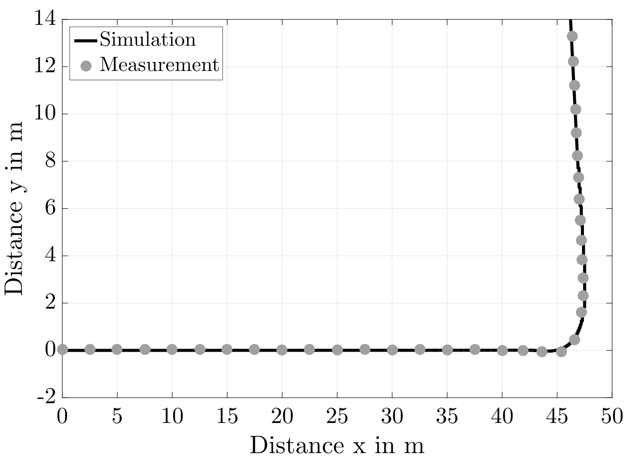


Figure 12. *Comparison of a measured and a simulated L test trajectory*

Consequently, the general conclusion is that the implementation and thus the merging of the two model types has been a success. The tyre model is suitable not only for comfort behaviour tests, as demonstrated by the tests involving cleats, but also for assessing vehicle dynamics aspects in the context of driving and tilting stability tests.

6 SUMMARY AND OUTLOOK

This publication has presented a successfully validated integrated tyre model for superelastic tyres, which can be applied both in vehicle comfort and vehicle dynamics simulations. The model was designed as a three-dimensional spokes model and integrated into a vehicle model as a separate multibody system by means of an internal programming language. The model sets itself apart from those developed by Oh or Busch and Stepanyuk as it encompasses force transmission in all spatial dimensions. It can thus be used in a wide range of applications. The parametrization of the model approach was mostly accomplished using experimental data, which had been obtained by means of a hydropulser and a drum test stand. Those model parameters that proved difficult to derive from experimental data were determined using a self-developed structural mechanics model and then implemented into the multibody system. Subsequently, the model was verified by means of multiple simulation runs and validated by comparing the results with the test stand results. The comparisons between these results consistently showed a high degree of concurrence. The next step was to assess the validity of the model in the context of an overall vehicle simulation. For this purpose, cleat tests as well as different cornering manoeuvres were simulated. The data obtained were then compared with data obtained during corresponding measuring runs. The results were once more highly similar, which means that the model meets the necessary requirements.

In order to adapt the model to an even broader range of applications, future efforts will focus not only on the separate transmission of lateral and longitudinal force behaviour but also on the overlapping of these forces. This can serve to assess whether the formulated contact model is capable of correctly representing the state of the resultant slip. Another aspect of great significance to industrial truck tyres is the camber moment, which is generated due to deformation of the contact patch during cornering and thus has an impact on the vehicle's tilting behaviour. To complete the modelling process, the proper representation of this moment must be examined, as well. In addition, the degree of detail of the overall vehicle model regarding vibration behaviour must be enhanced. Addressing all of these remaining issues would diminish the doubts that arose during the comparison of simulated and measured cleat tests and would thus allow a more meaningful conclusion regarding the tyre model's validity. In order to assign the model a general validity, the influence of temperature and wear on the mechanical tyre properties must be investigated. Here it would be of interest up to which internal temperature or degree of wear the parameter sets taken up are valid.

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