Parameterization and Modelling of Large Off-Road Tyres for Ride Analyses

Part 2 - Parameterization and Validation of Tyre Models

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Highlights

- Test data is used to parameterize various tyre models.

- The parameterization process is discussed in detail.

- Parameterisation effort varies greatly between tyre models.

- Tyre models are validated against measured test results.

- Single point contact model can't be used for ride simulations.

- FTire model shows the most promising results.

1   Abstract

Every mathematical model used in a simulation is an idealization and simplification of reality. Vehicle dynamic simulations that go beyond the fundamental investigations require complex multi-body simulation models. All components used in the simulation model, used to describe the real vehicle, must be modelled with sufficient accuracy to ensure the validity of results.

The tyre-road interaction presents one of the biggest challenges in creating an accurate vehicle model. The tyre is a complex assembly of a variety of components. The resulting force generation is thus

nonlinear and depends on the operating environments (road surface, inflation pressure, etc) and imposed states (slip angle, magnitude of loading, etc.).

Research has focused on describing the forces generated in the tyre-road contact area for many years. Many tyre models have been proposed and developed but proper validation studies are less accessible. These models were mostly developed and validated for passenger car tyres for application on relatively smooth roads. The improvement of ride comfort, safety and structural integrity of large off-road vehicles, over rough terrain, has become more significant in the development process of heavy vehicles.

The construction of off-road tyres is significantly different to that of on-road tyres eg. aspect ratio, construction of carcass, tread pattern, size, load capacity and other, posing the following questions:

i)   Can tyre models, that were developed for passenger car tyres, also be used to accurately represent large off-road tyres?

ii)   Can the parameterization process, that often require drum or flat track test data, be successfully adapted to data that is feasible to obtain for large off-road tyres as discussed in [1].

This paper investigates whether existing tyre models can be used to accurately describe the vertical behaviour of large off road tyres while driving over uneven terrain. The parameterisation process of four tyre models or contact models are discussed in detail. [1] Presented an extensive set of experimentally determined parameterization and validation data for a large off-road tyre. Both laboratory and field test are performed for various loads, inflation pressures and terrain inputs. The parameterized models are then validated against test results on various hard but rough off-road terrain and the results are discussed.

Keywords:

parameterization, tyre model, vertical tyre response, FTire, Single point contact, Enveloping contact, Volume contact, validation

2   Introduction to tyre modelling

Researchers have followed many different approaches to accurately model the tyre-road interaction. Many tyre models were developed for handling analyses on smooth road surfaces. Many of these models are empirical and should rather be classified as curve fits or lookup tables that interpolate experimental test results. They are used to study the vehicle dynamic responses to steering, throttle input and braking. For these analyses the vertical force variation in the tyre is less important and simple vertical models are often used.

However, to analyze the ride comfort and durability of a vehicle over rough terrain, the vertical forces generated in the tyre become significant. Vertical forces are also extremely important when simulating handling, road holding and braking over uneven terrain where wheel hop or loss of tyre terrain contact can be expected. The road surfaces, which are used for these analyses, are not smooth but have short wavelength obstacles. Many of the terrain features are often smaller than the tyre circumference and in the case of large off-road tyres, smaller than the tread blocks.

To accurately describe the vertical tyre–road interactions, different tyre models and corresponding road contact models have been developed. These contact models include one point contact models; roller contact models; fixed footprint contact models, radial spring models, flexible ring contact models; finite element models and many more.

The one point contact model is the most extensively used contact model. The modelling approach uses a single point on the road surface to represent the tyre contact patch. The tyre is generally represented by a parallel spring and damper assembly. As discussed by Zegelaar [2] this contact model is valid for road obstacles with wavelengths longer than 3 meters and a slope smaller than 5%. The model can thus not be used for short wavelength obstacles such as the Belgian paving or corrugations and is therefore not suitable for ride simulations over rough terrain. It is however included in this study for reference as it is widely abused in ride models.

The roller contact model, also known as the rigid ring tyre model, was developed to filter the road input [3]. The model approximates the tyre as a rigid ring, or disk. The tyre contact patch is

represented with a single contact. Contrary to the one point contact model, the contact point of the roller contact model is not restricted to lie vertically below the tyre axis of rotation. Consequently, the roller contact model filters the road surface and small wavelength road irregularities are filtered out. This approximation may be valid for very stiff tyres such as commercial truck tyres at high inflation pressures or solid rubber tyres.

The fixed footprint contact model uses a static contact area where the stiffness and damping is linearly distributed. The model averages the road irregularities, resulting in a smoother road excitation [4].

The radial spring model was developed to improve the behaviour of the rigid ring tyre model. The tyre is modelled using circumferentially distributed spring elements. Zegelaar [2] states that a radial spring contact model with nonlinear degressive radial springs could be used to accurately predict the tyre behaviour over discrete obstacles.

Flexible ring contact models represent the tyre as a deformable tread band. The flexible ring is modelled as a deformable beam and is thus able to incorporate the vibrational eigen frequencies of the tyre belt. As discussed by Zegelaar [2] the contact model is able to show the characteristic dip in the vertical force while rolling over cleats.

Finite element models are based on the detailed modelling of the tyre structure. These models are very powerful as the tyre geometry and tyre deformation as well as different material properties and other physical properties such as the air in the tyre, temperature effects, defects etc., can be accounted for. The drawback of this modelling approach is that the solving time is generally very long and makes the application for vehicle dynamics analyses infeasible.

The ability of a tyre model to accurately predict the tyre behaviour is not only dependent on the modelling approach that is used. It is also determined by the accuracy and availability of measured tyre data. Since most tyre models rely heavily on parameterization data, the validity of the model is directly dependent on the availability of the test data. If the required data cannot be obtained the model becomes obsolete.

The tyre that was analysed for this project was the Michelin XZL 16.0R20 all-terrain tyre. The tyre could be used for on or off road terrain. The tyre is rated for a maximum speed of 88km/h and a maximum load of 6595kg.

The acquisition of the required experimental test data for a large off-road tyre poses numerous challenges. The data is however significant in creating tyre models that are accurately representing the real tyre behaviour. A method of acquiring parameterisation test data, as well as an extensive data set, is discussed in great detail in [1].

The multi body dynamics analysis software, ADAMS/View is often used to simulate various vehicles manoeuvres. ADAMS/Tire is the module within ADAMS/View that is used to solve different tyre models [5]. Tyre models that are included in the module include motorcycle tyre models and soft soil tyre models. These models were not considered as they fall out of the scope of this investigation. All tyre models available in ADAMS/Tire, which can be used for vehicle dynamic simulations over uneven terrain, were investigated. Although some of the models are not suitable for ride simulations, they are included in this investigation as reference. The focus of this paper is to investigate the vertical tyre behaviour of various tyre models while driving over uneven terrain and not to investigate the lateral and longitudinal behaviour. A PAC89' tyre will be used to describe the lateral and longitudinal forces and the tyre moments during a simulation as the model is widely used and proven to show good correlation between measured and simulated results. The following tyre models and their corresponding contact models will be discussed in this paper:

- PAC89 tyre model with a One Point Contact model (OPC)
- PAC89 tyre model with a 3D Equivalent Volume Contact model (VC)
- PAC89 tyre model with a 3D Enveloping Contact model (3D ENV)
- FTire model

2.1   One Point Contact model

The One Point Contact model is the simplest and most extensively used contact model that is available. It represents the wheel as a spring and damper arrangement that trace a single point on the

road surface that is vertically below the wheel centre. The One Point Contact model, as implemented in ADAMS/Tire shows greater resemblance to a roller contact model. In this model the tyre and rim is considered to be a disk. The contact point is at the intersection between the wheel centre plane and the road tangent plane that has the shortest distance to the wheel centre. In this way the contact point is not constrained to be vertically below the wheel centre. This contact model is the default contact method for the Pacejka group of tyre models in ADAMS. Diagrams of both the point contact and roller contact models are shown in Fig.1.

point contact   roller contact
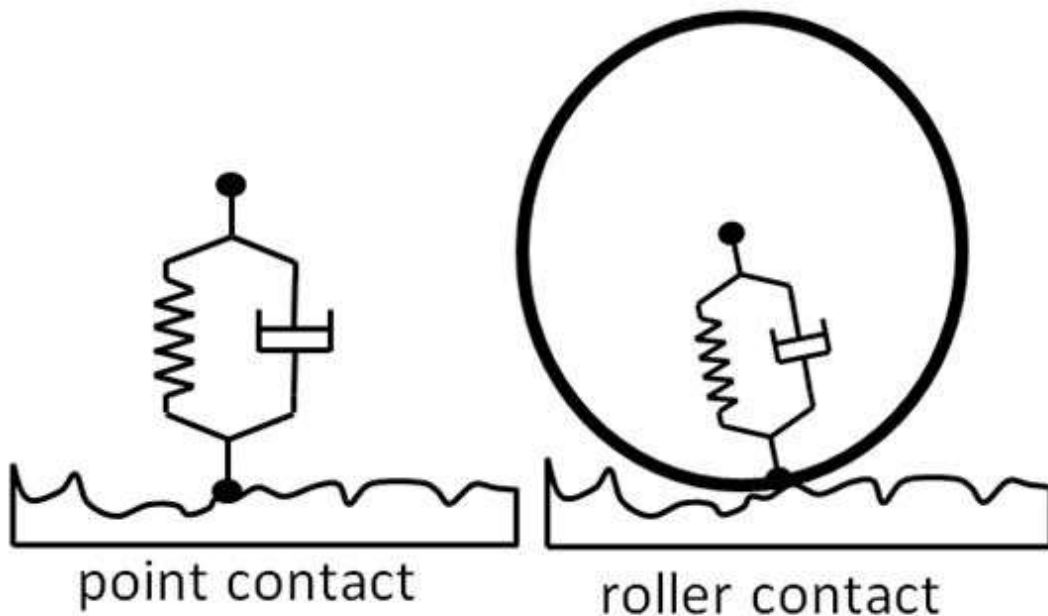
Figure 1 One point and roller contact models

This contact model can be described using only three parameters, namely the unloaded tyre diameter, tyre stiffness and a damping coefficient. The tyre stiffness, k_z, can either be described as linear or nonlinear while the damping can only be described with a constant damping coefficient, c_z. The vertical force is then described by Eq. (1).

$$F_z = k_z \rho + c_z \dot{\rho} \tag{1}$$

The data from the static deflection versus load test, as shown in Fig. 3 of [1], was fitted with a linear curve fit to determine a spring stiffness of 656.6N/mm at an inflation pressure of 300kPa. It was determined that the linear fit was not an accurate representation. To improve the accuracy of the tyre stiffness description the average loads were also calculated for every 5mm displacement, to create a nonlinear force displacement curve. This curve would later be used to describe the nonlinear tyre stiffness.

The non-rolling dynamic damping coefficient for the Michelin 16.00R20 tyre was calculated as 2.66Ns/mm and the dynamic rolling tyre damping 3.25Ns/mm [1].

2.2   3D Equivalent Volume Contact model

If the 3D Equivalent Volume Contact model is selected in the ADAMS tyre property file, the solver uses the intersection volume, between the un-deflected tyre and road, to calculate the normal force. From the intersection volume the solver computes the effective normal tyre contact plane, tyre deflection, tyre to road contact point, and the effective road friction (Fig. 2).
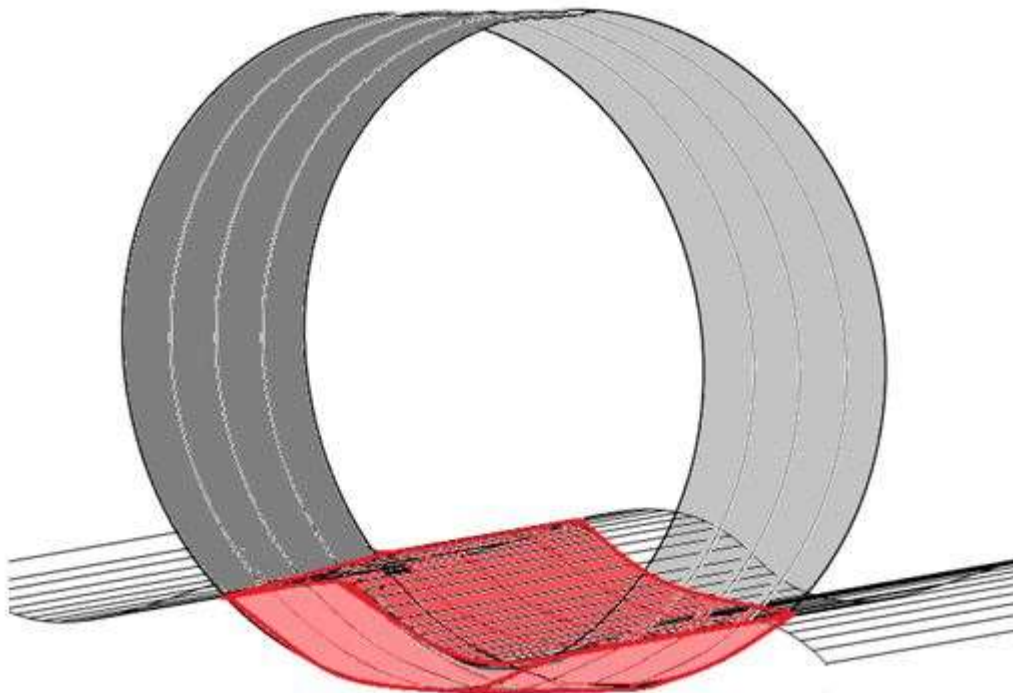

Figure 2 3D Equivalent Volume Contact model

The 3D Equivalent Volume Contact model must be used with a 3D shell road. In a 3D shell road-definition-file the road is modelled as discrete triangular patches. The solver calculates the displaced volume of these patches and the tyre disks to determine the equivalent contact point of the tyre.

The vertical force is then described by the same formula as was used in the One Point Contact model (Eq. 1). Furthermore the cross-sectional tyre shape needs to be defined.

The cross-sectional shape was defined using fractions of the tyre radius and width. The carcass shape was assumed to be symmetrical and only needed to be defined for one half of the tyre width. Absolute coordinate values for the shape are computed by multiplying relative values with the unloaded radius and half-width of the tire. The measured outer contour, as shown in Fig. 12 of [1], was used to determine the required ratios. A maximum of 10 points could be used to describe the carcass shape in the tyre property file. The cross-sectional shape data, as discussed in [1], was used to describe the tyre shape.
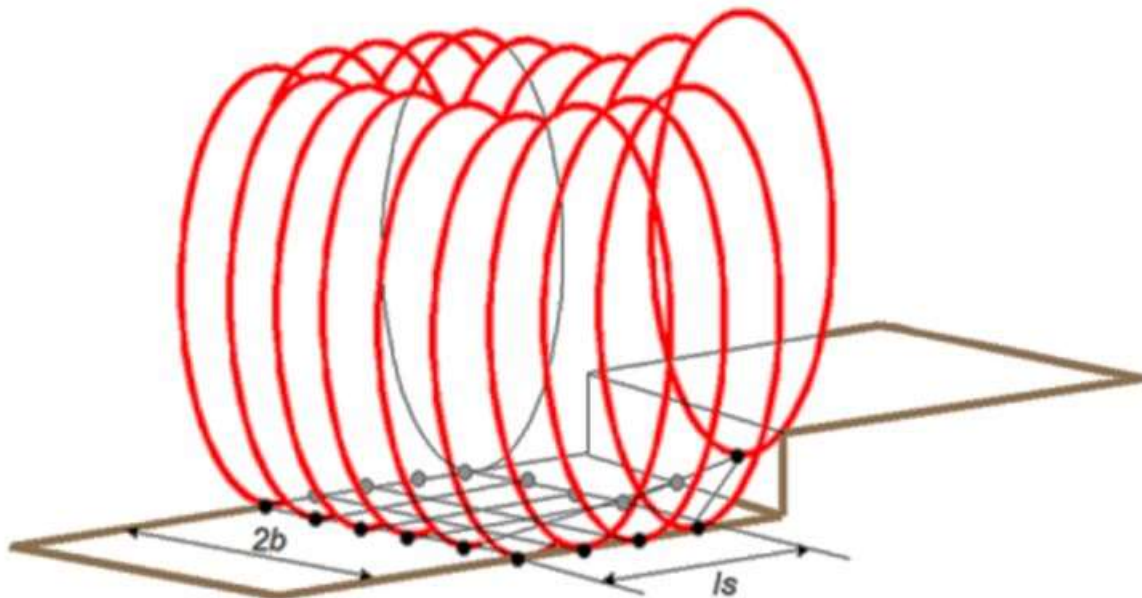

Figure 3 Elliptical cam model as proposed by Schmeitz [5]

2.3   3D Enveloping Contact model

The 3D Enveloping Contact model was based on the work done by Schmeitz [6]. Schmeitz proposed a cam model to emulate the enveloping behaviour of the tyre. The model comprises a number of cams

that are positioned in such a way that they correspond to the outside of the contact patch, as shown in Fig. 3. Due to the shape of the cams this model is also referred to as the "tandem-egg" model.

The positions and orientations of all cams are calculated during the simulation to determine the effective road height, slope, and curvature as well as the effective road camber. The effective road parameters are then used to determine the forces that are generated in the contact patch. The tyre stiffness and damping was described in the same way as was the done in the One Point Contact tyre model.

The tyre contact patch dimensions are described by the half contact patch length, a, as shown in Eq. 2. R_0 is the unloaded tyre radius and ρ_z the tyre deflection.

$$a = p_{A1}R_0\left(\frac{\rho_z}{R_0} + p_{A2}\sqrt{\frac{\rho_z}{R_0}}\right) \tag{2}$$

The measured tyre contact patch areas were used to find the parameters p_{A1} and p_{A2}. The Parameters were determined to be 0.15 and 7.06 respectively. Figure 4 shows the measured half tyre contact patch length and the fitted results.
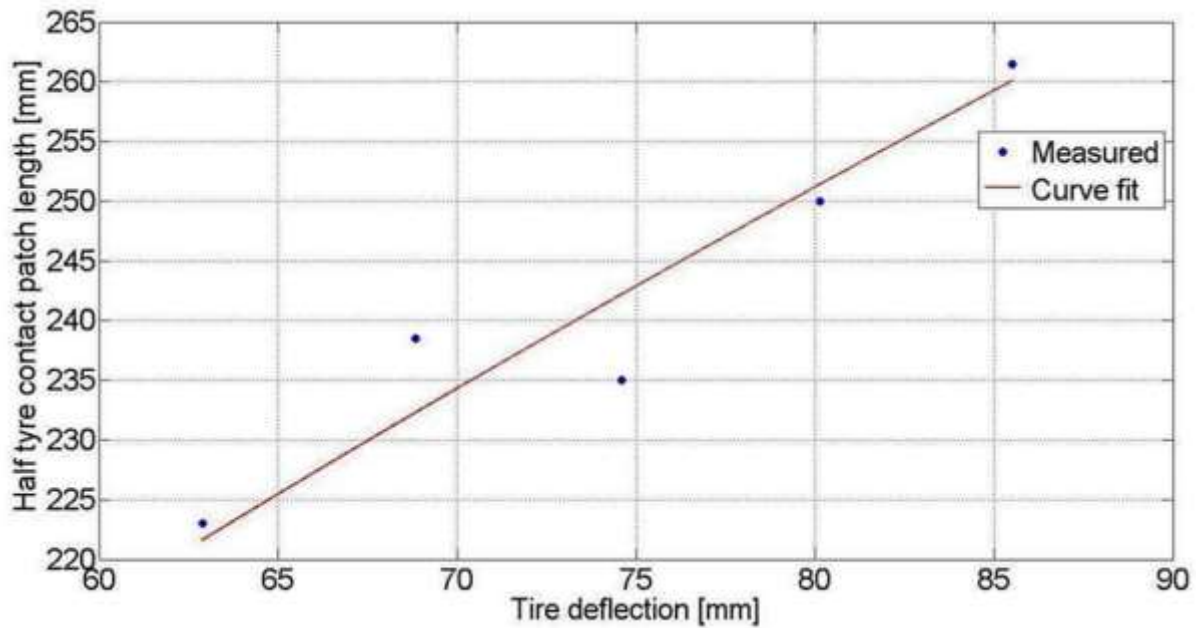

Figure 4 Half contact patch length vs. tyre deflection

The figure shows that the half contact patch length formula can be used to adequately describe the contact patch length. The half contact patch width is described by Eq. 3.

$$b = p_{B1}W_0\left(\frac{\rho_z}{R_0} + p_{B2}\sqrt{\frac{\rho_z}{R_0}} + p_{B3}\frac{\rho_z}{R_0}\sqrt{\frac{\rho_z}{R_0}}\right) \tag{3}$$

An optimization process was used to fit this formula to the measured results. The three parameters, p_{B1}, p_{B2} and p_{B3}, were determined to be 0.0017, 16006.5 and -77819.1 respectively.
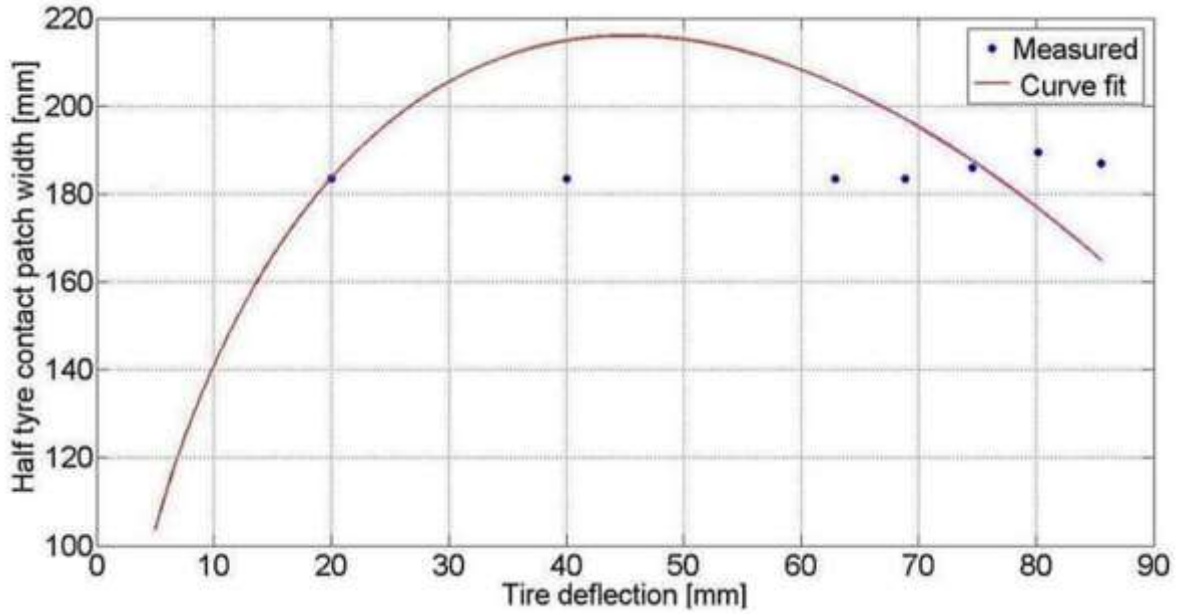

Figure 5 Half contact patch width vs. tyre deflection

The curve fit result is shown in Fig. 5. The figure shows that the contact width is largely independent of the tyre deflection. The curve fit, with the given formula, cannot be used to accurately describe the tyre contact width for large off road tyres. The solver will calculate a wider contact patch for tyre deflections between 20 and 75 mm and a smaller width for tyre deflections above 75 mm. The shape of the cams is given by:

$$\left(\frac{x_e}{p_{ae}R_0}\right)^{p_{ce}} + \left(\frac{y_e}{p_{be}R_0}\right)^{p_{ce}} = 1 \tag{4}$$

The coefficients p_{ae}, p_{be} and p_{ce} describe the shape of the individual cam. The ADAMS help file [5] informs the user to compare the shape of a deflected tyre with the ellipsoid shape to derive these parameters. Figure 6 shows an attempt at following this procedure. The figure shows the tyre test trailer with a ballast load of 5440kg.
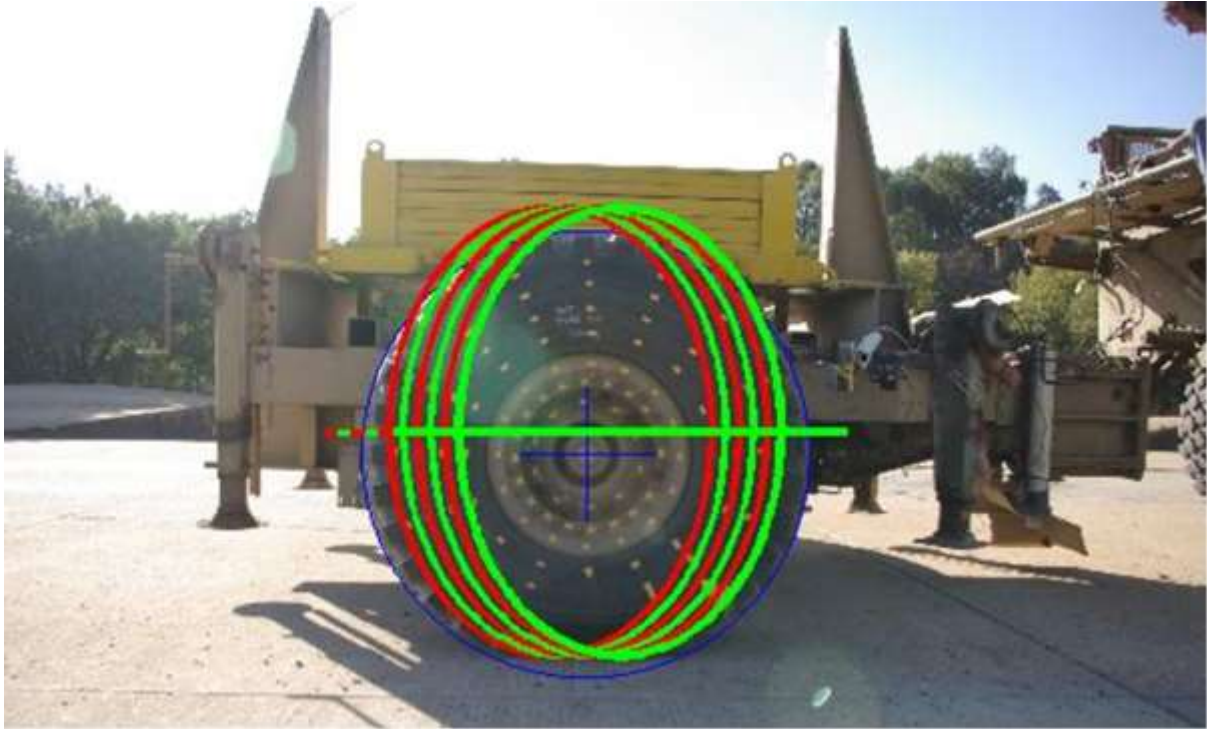

Figure 6 Ellipsoid shape

The shape of the cams were only dependent on the unloaded tyre radius, not on the tyre load. The large aspect ratio of tyres made it difficult to find the correct parameters for the cams. The shape of the cams represent the tyre the best when using the values 0.72, 1 and 2 for the parameters p_{ae}, p_{be} and p_{ce} respectively.

In the lateral direction the cams were equally spaced over the entire contact width. In the longitudinal direction they were spaced equally over the base length l_s. The base length is dependent on the contact length and is given by:

$$l_s = 2p_{ls}a \tag{5}$$

The tandem base length factor was chosen to be 0.9. Schmeitz[6] has shown that 5 cams along the length and 6 along the width of the contact patch showed the best accuracy-performance ratio for a passenger car tyre. The number of cams across the contact patch width and length, for the larger tyre, were chosen to be 10 and 15 respectively. More cams were used along the contact patch length because the contact patch is longer than it is wide.
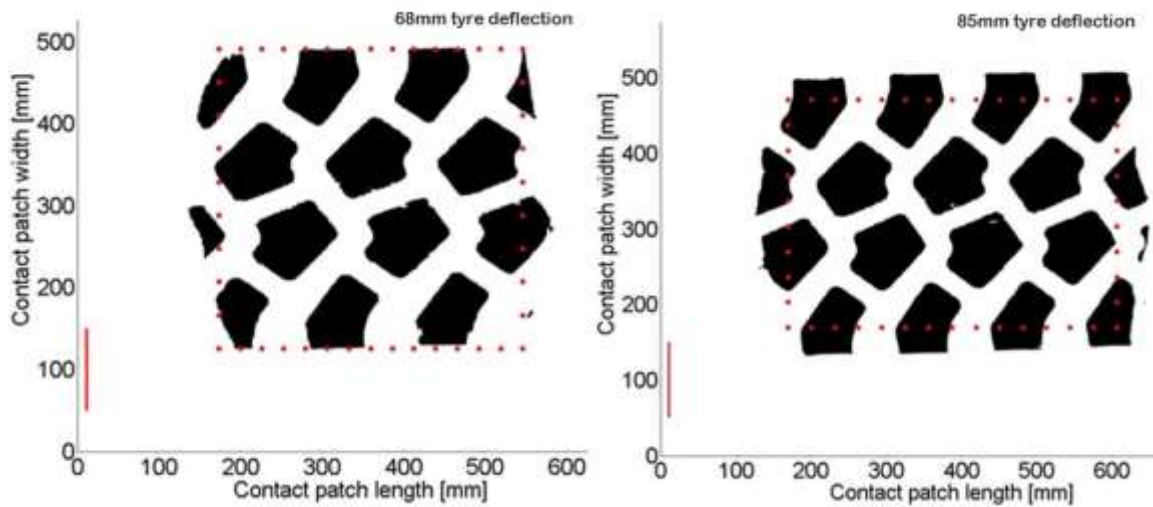

Figure 7 Predicted 3D Enveloping Contact patch dimensions

The calculated contact patch dimensions could be compared to the measured results. Figure 7 shows the result of the comparison. The red markers indicate the contact points of the individual cams. The contact patch on the left shows the tyre with a 68mm tyre deflection. The figure shows that the calculated contact patch width is slightly larger than the measured result. The contact patch on the right shows a contact patch from a tyre with a deflection of 85 mm. As expected the calculated contact patch width is smaller than the measured result. The overall accuracy of the fits is acceptable.

2.4   FTire

FTire (Flexible Structure Tyre Model) is a full 3D nonlinear in-plane and out-of-plane tyre model. The Model was developed by Gipser [7] over the past 15 years. FTire was developed for vehicle comfort simulations and the prediction of road loads with extremely short wave-lengths obstacles but
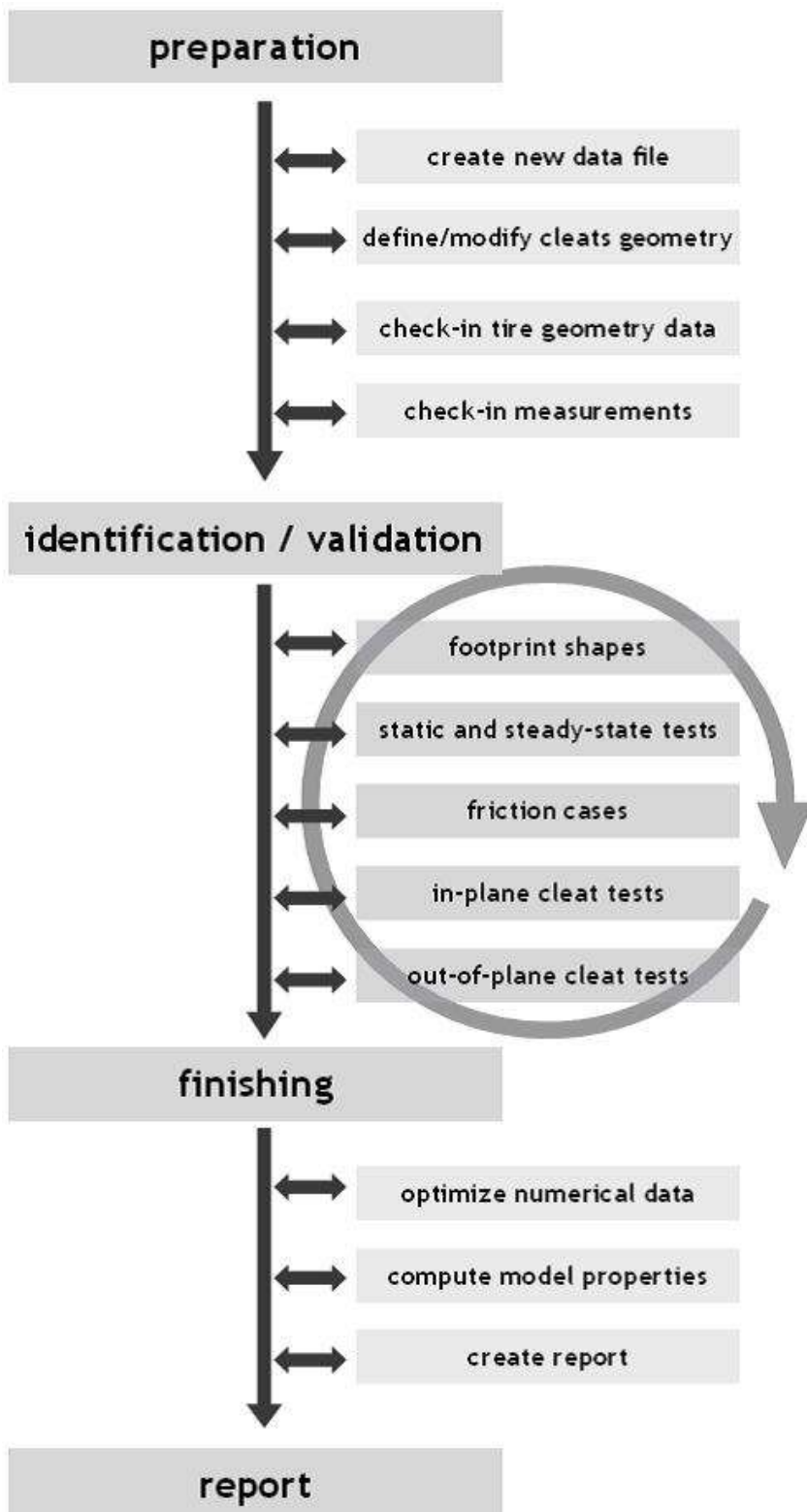
Figure 8 FTire parameterization procedure (Gipser, 2002)

can also be used for handling simulations. This model is based on a structural dynamics approach where the previously discussed models are based on analytical foundations. The tyre model describes the tyre belt as a flexible ring that can flex and extend in the radial, tangential and lateral directions. The tyre model can be seen as a very course, nonlinear finite element model.

The program FTire/fit is structured in such a way that it guides the user through the parameterization process. The process is summarized in Fig. 8. In the first stage of the parameter identification process the tyre footprints were used. The tyre footprints hold much information regarding the tyre stiffness and shape of the tyre. An accurate representation of the road input was imperative to accurately predict the tyre behaviour. Furthermore, if the tyre model was able to reproduce an accurate set of footprints, the tyre shape and overall stiffness is modelled accurately. The stiffness of the tread rubber also influences the shape of the footprints. The tread stiffness was determined experimentally[1].

Figure 9 shows the footprints of the test tyre at an inflation pressure of 300kPa and two different load conditions, 68mm and 85mm tyre deflection. The calculated footprint shape of the FTire model is shown on the same figure and is indicated with a red line. From the figures it can be seen that the model is able to accurately predict the dimensions and the shape of the tyre contact patch.
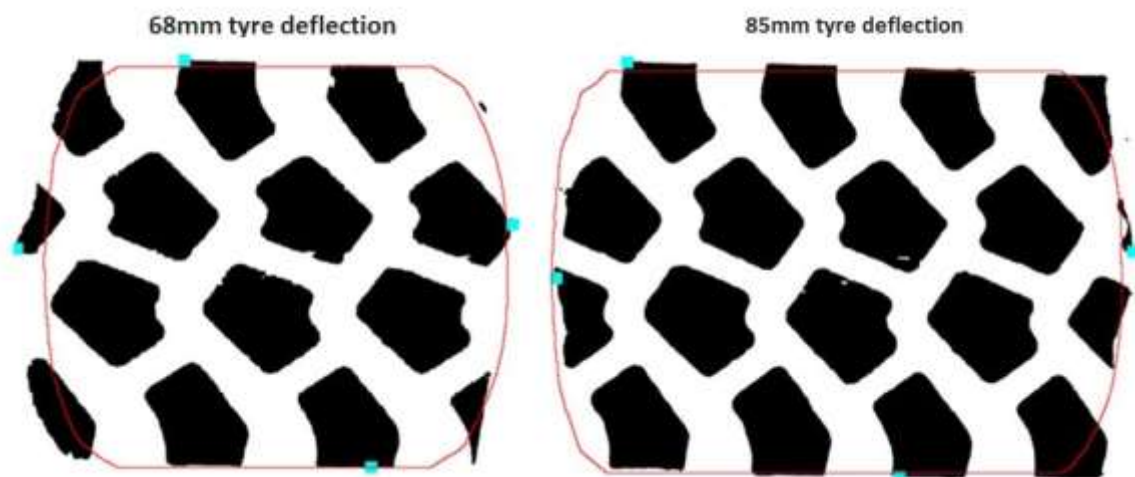

Figure 9 Footprint, 300kPa; left - 4000kg load; right - 6000kg load

To identify the radial tyre stiffness parameters of the tyre model the load deflection test on a flat surface was used. Figure 10 shows the measured and predicted static load deflection curves at a 300kPa inflation pressure. It was found that the tyre model is able to accurately reproduce the load deflection curves for various tyre pressures.
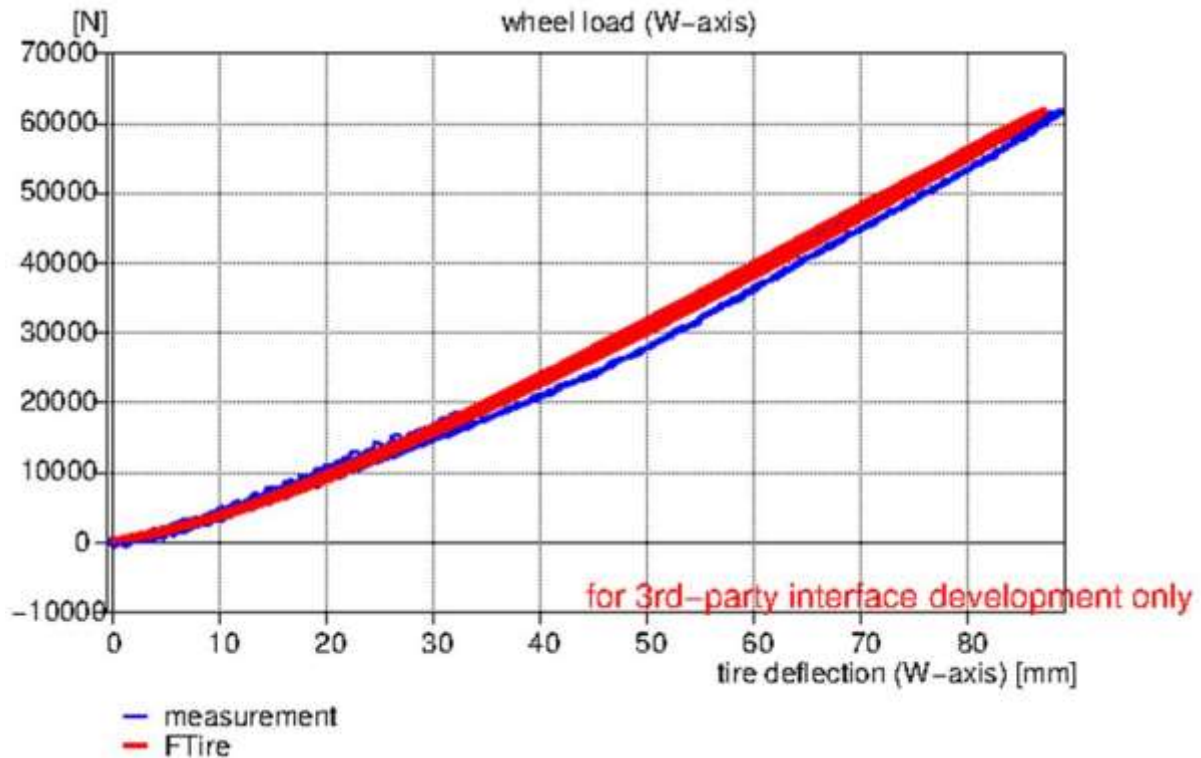

Figure 10 Vertical stiffness on flat surface, 300kPa inflation pressure, 0 deg camber angle

The static tyre behaviour over a cleat holds information about the lateral and longitudinal belt stiffness. The transversal cleat tests were used to identify the in plane belt bending stiffness. Figure 11 shows the comparison between the measured and simulated load deflection curve. From the figure it can be seen that the model is capable of accurately predicting the forces on a flat surface and on transversal cleats. The model is able to accurately predict the load deflection curve for cleats of various dimensions.

The contact nodes in the FTire model could be arranged to represent the test tyre tread pattern. It was expected that the pattern would influence the tyre behaviour, especially during tests with small wavelength road unevenness. To investigate this, an FTire model was created where the contact

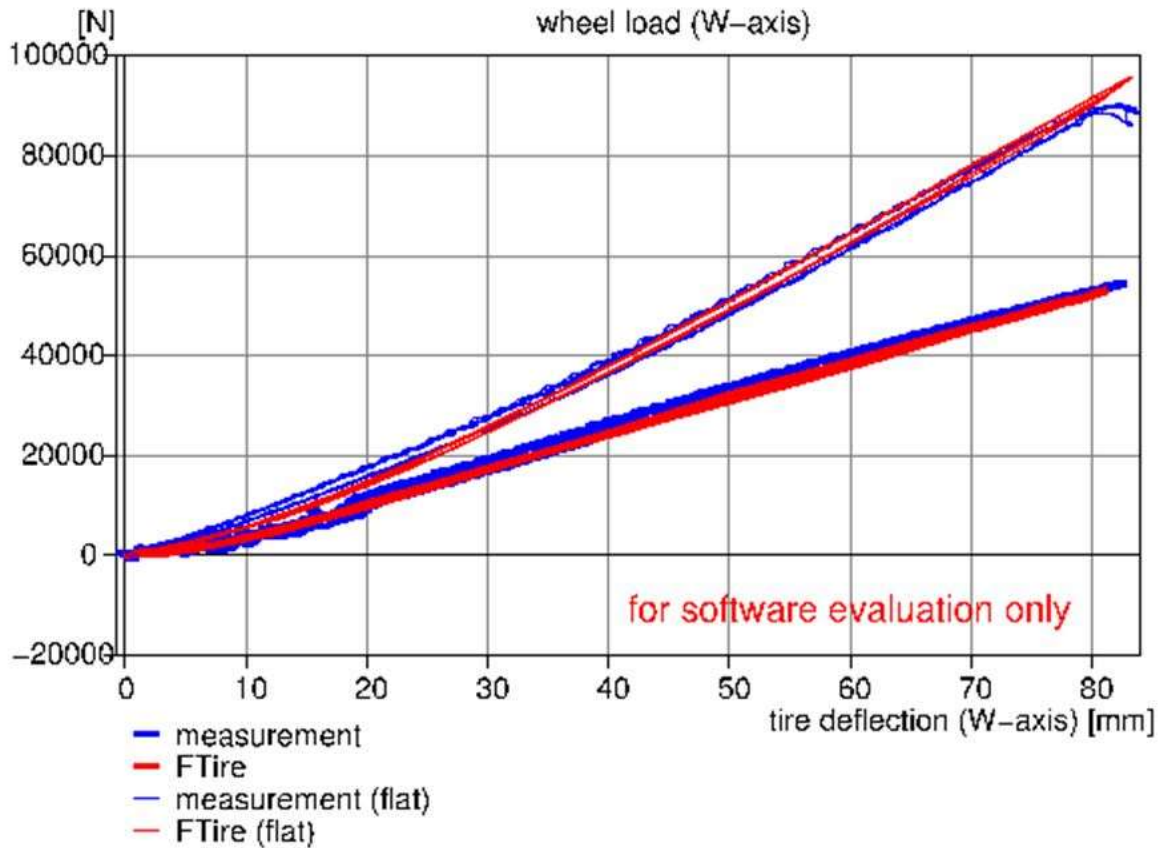elements were arranged equally and a second model was created where the tread pattern was accounted for.

Figure 11 Vertical stiffness on 38 x38 mm cleat, 550kPa inflation pressure, 0 deg camber angle

The parameters that are used to describe the damping behaviour of the FTire model could not be parameterized within FTire/fit. The FTire/fit relied on dynamic drum cleat tests to extract the parameter values. Due to the size and load of the tyre, dynamic drum cleat tests were not possible.

The damping behaviour of the tyre model is modelled using viscous damping elements between belt nodes and the rim. The damping behaviour was described, in the property file and during the parameterization process, with modal damping values. The model then tries to match these damping values. The help file [8] stated that the damping model was of limited accuracy because the rubber damping was more accurately described by frequency independent hysteresis loops. It further stated

that the measured damping values are too low in many cases and should be increased to achieve similar damping behaviour between the real tyre and the FTire model.

A FTire model was created, using the modal analysis damping values, and used in a simulation over 50 mm cleats. The simulation results were compared to the measured tyre response to determine whether the tyre showed an acceptable damping behaviour. The damping parameters were then adjusted accordingly to improve the correlation between the measured and predicted tyre behaviour.

This was a brute force method but it was found to be effective but time consuming. A concern using the described method was that the relationship between the three modal damping values could not be established. The parameters were only altered so that the ratio between them remained unchanged and held the same relationship as the measured damping values.

The measured and predicted tyre damping behaviour is shown in Fig. 12. Section A shows the first section where the tyre is in contact with the obstacle, while second section, B, shows the free response of the trailer.
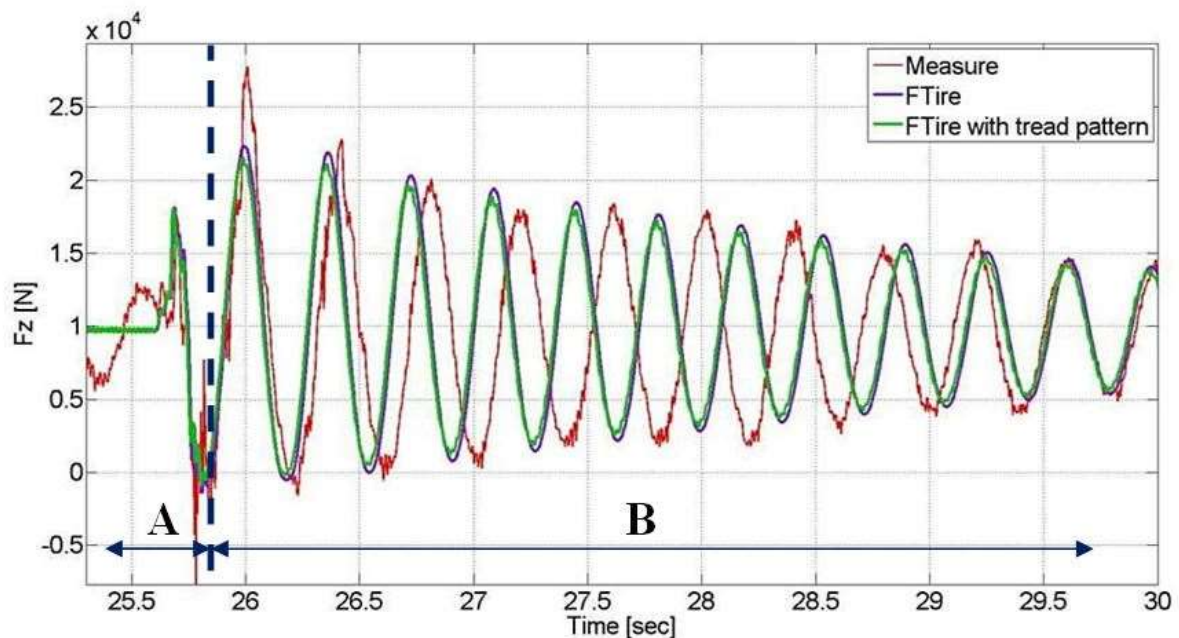

Figure 12 FTire Simulation result, 50mm cleat, 36% of LI, 27km/h

Tests conducted to determine the side force slip angle relationships [1] of the tyre were also used in the parameterization process. The force generation due to a change in the tyre slip angle did however not fall in the scope of this investigation and will thus not be discussed in detail.

3   Tyre model validation

The validation process is an important part of any mathematical model created to emulate physical systems. In the process simulation results are compared to experimental test data. The datasets are compared to determine their agreement and to find inconsistencies. Inconsistencies can then be related to the incorrect measurements or to uncertainties in the model. Inaccuracies in the measured data may arise from altering measurement conditions or due to inaccurate instruments. Uncertainties in the simulation model can be related to either the model inputs, the numerical approximations or in the model itself. The validation process is summarized in Fig. 13.
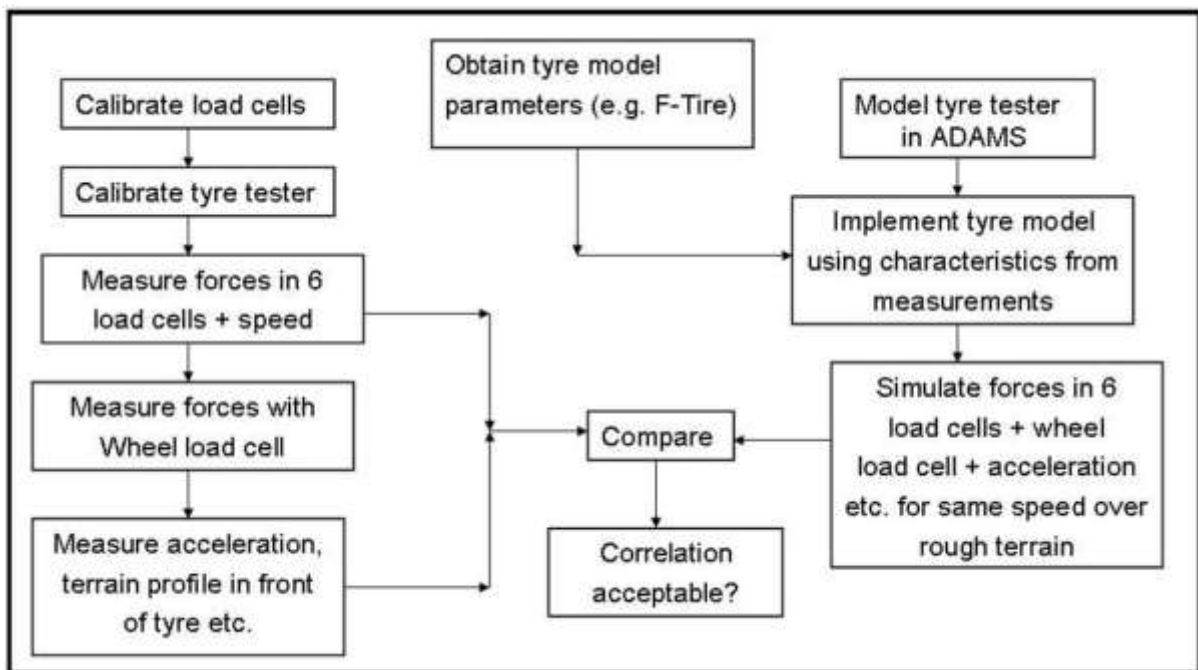

Figure 13 Validation process [8]

Figure 14 shows the measured and simulated vertical tyre response while negotiating a 50mm cleat at a constant speed of 12km/h. The figure shows that the One Point Follower Contact model, OPC, generates a larger force while the tyre encounters the cleat but a smaller disturbance once the cleat is

passed. This behaviour could be related to the approach used to model the tyre. The tyre was modelled as a rigid disc which did not allow the model tyre to envelop around the cleat, so called swallowing of the obstacle. The response thus shows similarities to an impulse disturbance.
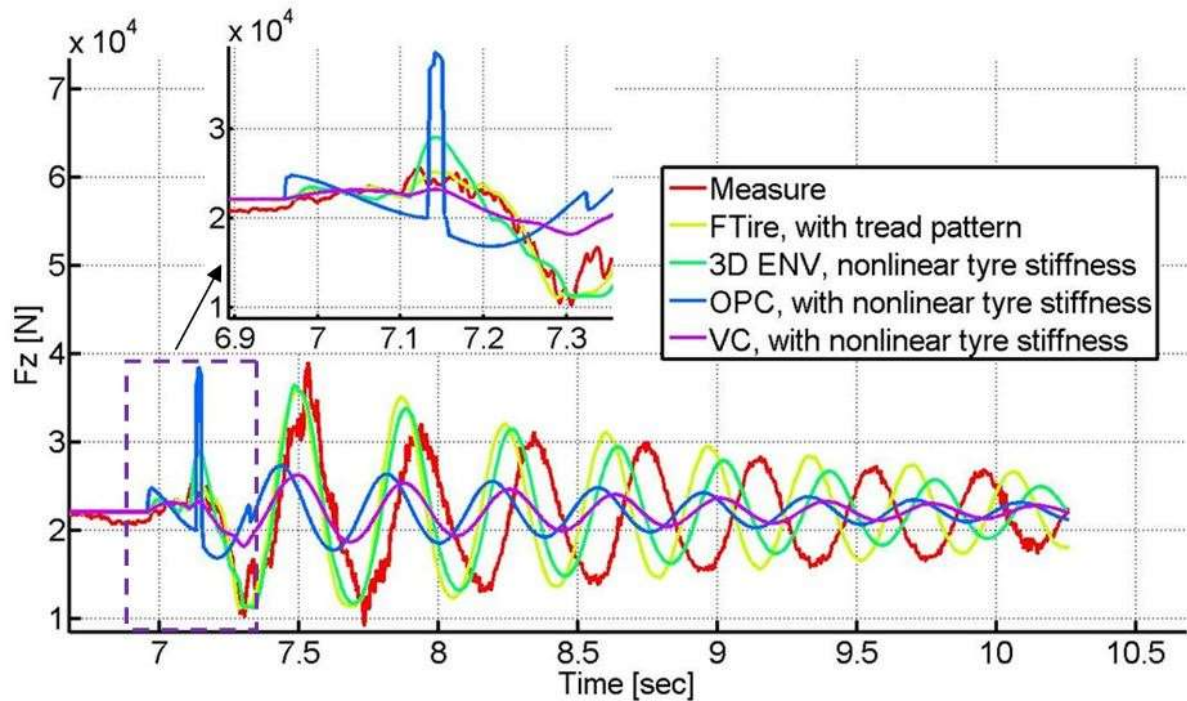

Figure 14 Measured and simulation result, 50mm cleat

The vertical tyre response of the 3D Equivalent Volume Contact model, VC, predicted the tyre behaviour while negotiating the cleat better than the One Point Follower contact model. The later response due to the disturbance did not show any noticeable improvements compared to the One Point Follower contact model.

The 3D Enveloping Contact model was developed to improve the tyre model behaviour over cleats by incorporating the lengthening and swallowing effect of a real tyre. Due to the improved tyre response, while negotiating the cleat, the resulting free response of the simulation is similar to the measured response.

The FTire model replicated the measured tyre behaviour most accurately during impact with the cleat. The response after the tyre negotiated the cleat was similar to the simulation results when the 3D Enveloping Contact model was used.

The figures show that the period of the trailer oscillations were larger in the measured results compared to the simulation results. This could be attributed to the way the connection between the trailer and the towing vehicle was modeled. In ADAMS the trailer hook was modeled so that the hook was constrained in the vertical and lateral direction. A force was applied to the hook in the longitudinal direction to facilitate the longitudinal acceleration. The rotations about all three axes were not constrained at the hook. This modeling approach removes the vertical degree of freedom of the tow hitch. The ADAMS model is thus stiffer compared to real test trailer. Contributing to this behavior is the difficulty to accurately model the highly nonlinear tyre damping. The main focus is on the tyre behavior where the tyre is in contact with the obstacle, simulating off-road driving, while the free response behaviour is of lesser importance.
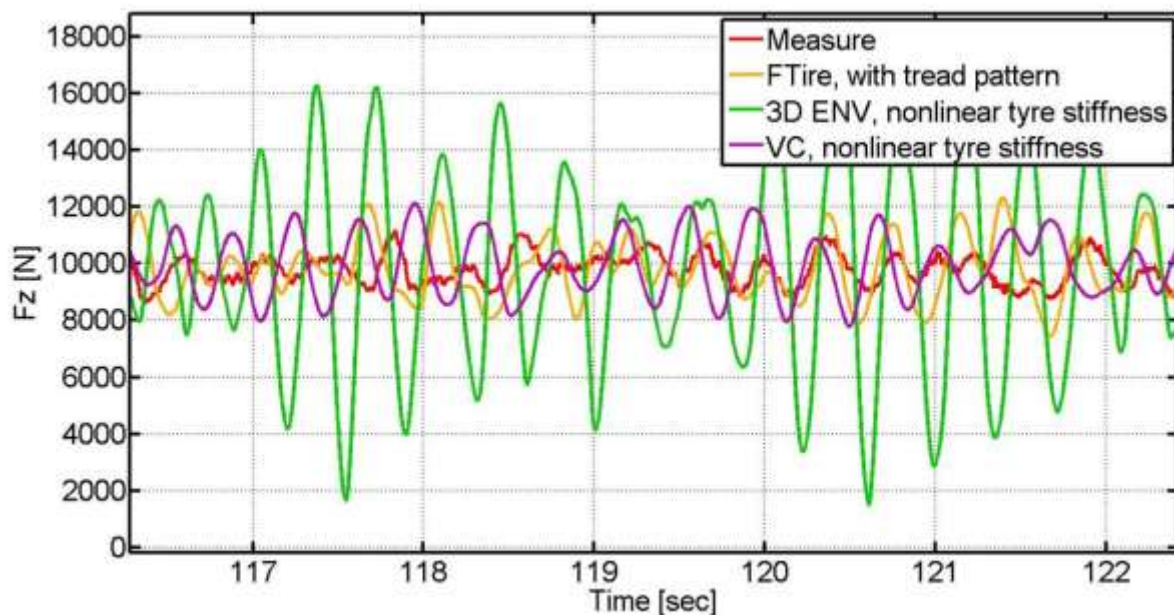

Figure 15 Simulation results of the Belgian paving test track, 36 % of the LI, 3km/h

Comparing measured and simulation results to determine whether the model can accurately describe the tyre forces while rolling over the test tracks, is challenging. The results of a Belgian paving test, at

a test velocity of 5km/h, is shown in Fig. 15. The direct time domain comparison is problematic as a small change in the tyre path during the simulation could result in a different input condition. The only conclusion that can be made from Fig. 15 is that the 3D Enveloping Contact model gives unacceptable results. Simulation results of the One Point Contact model are omitted due to unrealistic tyre model behaviour.

The results were therefore analyzed using the probability theorem rather than applying a time domain comparison. Figure 16 shows a histogram of the vertical tyre forces for the measured data as well as the simulation results for the various tyre models. The distribution of the vertical tyre forces, as shown in the histogram, describe the overall tyre behaviour during the test. An oscillatory response of the vertical tyre forces would result in a wide spread distribution. The One Point Contact model and the 3D Enveloping Contact model show this behaviour. A narrow distribution indicates that the tyre forces are located near the static tyre load. The mean measured and simulated tyre force should be similar to the static tyre load. A difference would be found when the ratio of ground contact loss differed between the simulation and the actual test. It could also indicate different damping behaviours.
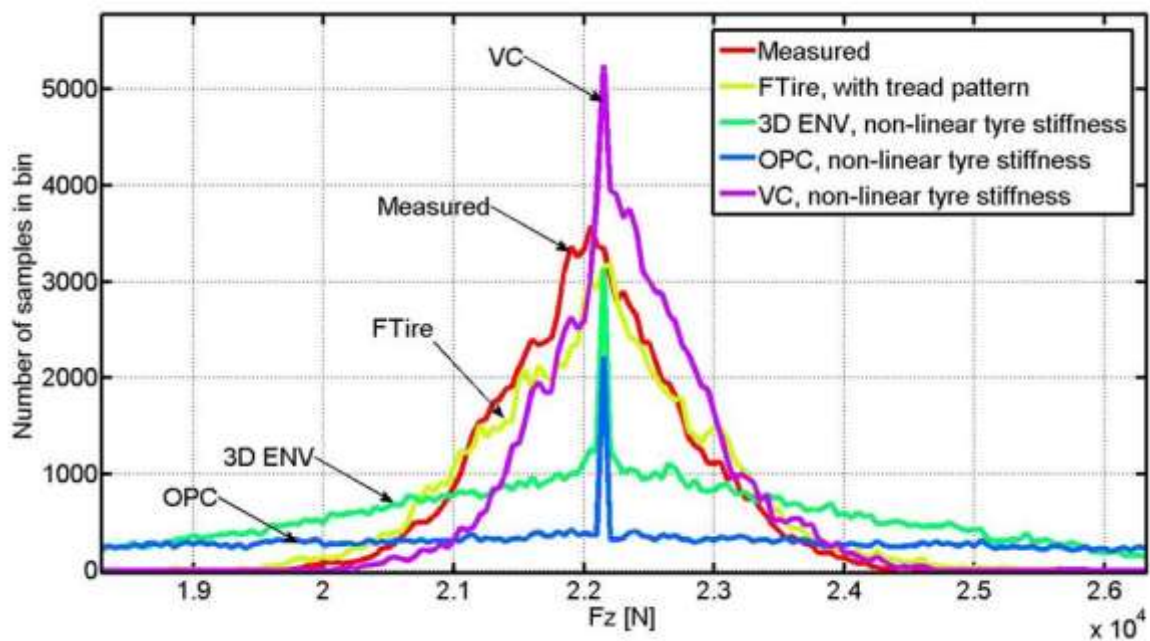

Figure 16 Normal load probability density of the Belgian paving test, 36% of the LI, 3km/h

Figure 16 indicates that the FTire model is capable of describing the vertical tyre behavior for simulations over the Belgian paving test track. The Volume Contact model also shows acceptable results. The One Point Contact model and the 3D Enveloping Contact model are unsuitable for use over uneven terrain.

The corrugation test tracks were used to simulate driving manoeuvres on dirt roads that developed a series of regular bumps with short spacing on the road surface. The measured and predicted tyre forces did not conform to a normal distribution in many cases as is shown in Fig. 17. The FTire model was capable to accurately predict the normal tyre forces. All other tyre models predicted unrealistic tyre responses during the simulations.

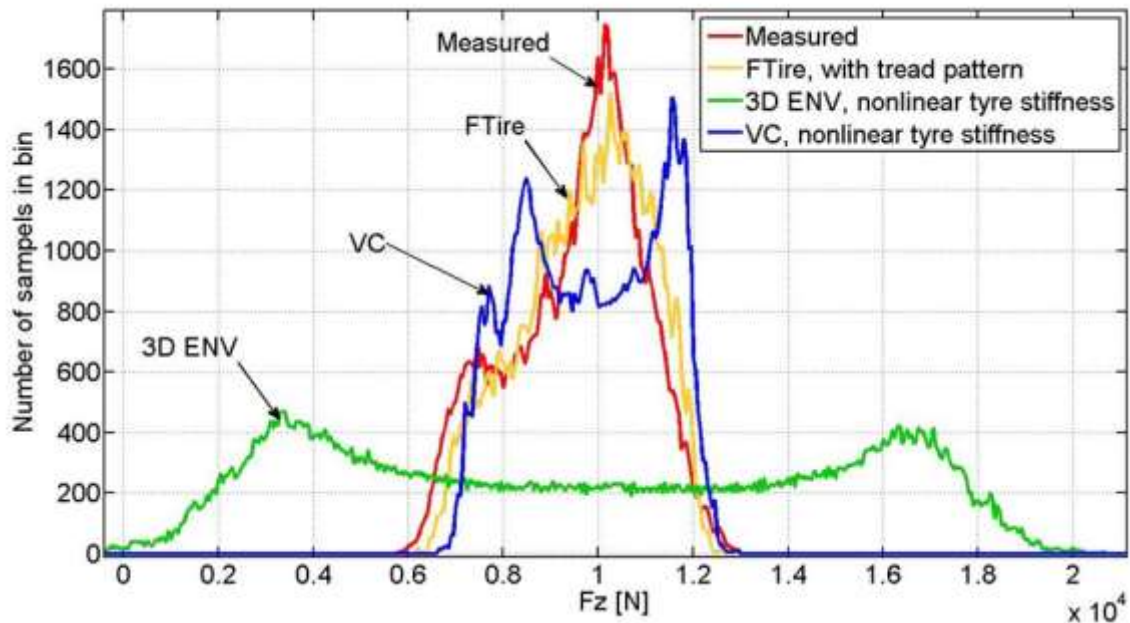Figure 17 Normal load probability density of the Parallel corrugation test, 36% of LI, 3km/h

4   Conclusions

Tyre modelling techniques and their approaches to modelling the tyre road interface have been discussed. Four tyre models implemented in the multi body dynamics simulation software, ADAMS, were discussed in detail. The One Point Contact, Volume Contact and the 3D Enveloping Contact

models were developed by researchers and implemented by MSC Software in ADAMS. The FTire model, developed by COSIN scientific software, was also discussed.

The tyre models rely heavily on experimental test data for the parameterization process. The amount of test data required and the effort in obtaining the required test data varies greatly amongst the tyre models. The availability of the test data should thus be considered when selecting a tyre model. The process of acquiring the required test data was discussed in great detail in [1].

The parameterization process of the tyre models was also discussed. The parameterization of the Pacejka contact models was found to be manageable and apparent. The model behaviour of the One Point Contact and the Volume Contact model was unsatisfactory and could not be improved further. The parameters that determine the model behaviour have been determined and fit experimental data well. It was found that the description of the tyre contact patch width, as used in the 3D Enveloping Contact model, could not be used to accurately describe the footprint width of large truck tyres. To improve the filtering behaviour of the 3D Enveloping Contact model, the number of cams used should be investigated further. The cams influence the filtering behaviour and it is hypothesized that a correct number of cams could impact the accuracy of the tyre response. The number of cams required would depend on the road surface, simulation speed and load range.

The FTire parameterization process proved to be the most challenging. Multiple parameterization iterations were completed, using an extensive data set, until a tyre model with the required accuracy was created. To date, only static parameterization tests could be used in the parameterization program FTire/fit. The parameters that define the dynamic behaviour of the FTire model were determined by a crude trail and error process.

A contribution can be made if further measurement procedures are investigated that can be used during the parameterization process of FTire. It is proposed that the following should be investigated:

I. Develop test equipment that can be used to determine dynamic tyre behaviour in a laboratory. A significant contribution to the improvement of the model can be made with the development of such test equipment. High speed field tests are in many cases

impractical especially for large tyres. A test rig that can be used to investigate the tyre behaviour at low speeds could hold useful information.

   II.   Include lateral and longitudinal test data in the parameterization process. These tests can be used to further improve the FTire model.

   III.   Develop test equipment that can be used to investigate camber effects. Static and dynamic test should include camber test data.

Simulations over the test tracks have shown that the FTire model shows the best correlation between the measured and simulated tyre response. The Volume Contact model could predict the tyre behaviour with acceptable accuracy in some cases. The accuracy of the results were however dependent on the test velocity. The Volume Contact model could not predict the impulse tyre response over short, high amplitude, road irregularities.

Group	Obstacle	Tyre model/ Contact model			
		FTire	3D ENV	VC	OPC
Discrete Obstacles	Cleats	Comparable, best	Comparable	Not representative	Not representative
	Trapezoidal bump	Comparable, best	Not representative	Comparable	Not representative
Rough tracks	Belgian paving	Comparable, best	Not representative	Comparable, under certain conditions	Not representative
	Fatigue track	Comparable, best	Not representative	Comparable	Not representative
	Parallel corrugations	Comparable, best	Not representative	Comparable	Not representative
	Angled corrugations	Comparable, best	Not representative	Comparable	Not representative

Table 1 Summary of the Validation metrics

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The results of the validation metrics were summarized in Table 1. The FTire model could in most cases reproduce the best results. The 3D Enveloping Contact model can be used for simulations over discrete obstacles similar to the cleats. The 3D Equivalent Volume Contact model showed acceptable results for simulations where the tyre negotiated the trapezoidal bump, Fatigue track and the Corrugation tracks.

It is concluded that existing tyre models could be used to predict the tyre forces generated by large off-road tyres over rough terrain. The best overall results were found when the FTire model was used. This tyre model predicted the tyre response accurately over a wide range of obstacles and rough test tacks. Tyre models that use a 3D Enveloping Contact model could be used for simulations over discrete obstacles. The 3D Equivalent Volume Contact model may be useful in specific cases where the peak impulse loads were not important. The One Point Contact model could not predict acceptable results and should be avoided for all ride simulations.

5 Notation

Symbol	Unit	Description
a	m	half contact lenght
b	m	half contact width
c_z	Ns/m	vertical tyre damping coefficient
F_z	N	vertical tyre force
k_z	N/m	vertical tyre stiffness
l_s	m	tandem based length
p_{A1}		tyre contact length parameter A1
p_{A2}		tyre contact length parameter A2
p_{ae}		length factor of cam
p_{B1}		tyre contact width parameter B1
p_{B2}		tyre contact width parameter B3
p_{B3}		tyre contact width parameter B3
p_{be}		height factor of cam
p_{ce}		ellipse response

p_{ls}		cam base length coefficient
R_0	m	unloaded tyre radius
W_0	m	nominal section width of the tyre
x		variable
x_e	m	x value of cam
y_e	m	y value of cam
ρ_z	m	tyre deflection in z direction
$\dot{\rho}$	m/s	tyre deflection velocity
ρ	m	tyre deflection

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