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TME180 - Automotive Engineering Project

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Abstract

Tyres are an important component that affect the energy consumption and performance of all vehicles, including trucks. The vital properties of the tyre realising these phenomena are the rolling resistance and lateral slip stiffness, respectively. This project aims to investigate and understand the influence of vertical load on the rolling resistance coefficient and lateral slip stiffness and based on these investigations, evaluate the possibility of creating simple and physically interpretable yet tune-able tyre models. The lateral slip stiffness investigation involves modelling a physical tire model based on the brush model with parabolic pressure distribution and a curve-fit model based on the 'openPBS' tool non-linear tyre model. These models are tuned to match the data extracted from the VTI experiments of test performed with a truck tyre on the tyre testing facility. The lateral slip stiffness generated using both models is compared with the experimental test data, and found that the curve-fit model presents a better approximation of the test data. However, dense measurement data in terms of more data sets for varying vertical loads as well as more number of measured lateral force vs. slip points for each vertical load data set will be required to confirm this conclusion. The variation of RRC with lifted and non-lifted axles is studied using test data availed from a report by Lennart Cider, discussing the rollout tests of two different trucks with lifted and non-lifted axles. A proposition of load independent wheel bearing torque loss is considered as an alternate explanation for the varying rolling resistance and a study conducted for the same shows that the friction losses in the wheel bearings could approximately account for around only 13.5% of the estimated change in rolling resistance. Hence this cause solely is an unlikely explanation of the difference in rolling resistance. A finite element study of the tyre contact patch and the vertical load offset could aid the understanding of this phenomena better.

Keywords: Truck tyre, tyre model, rolling resistance, slip stiffness, bearing loss.

Contents

1

Introduction

This chapter presents the background to the project, as well as the purpose, delimitaions and limitations.

1.1 Background

Long heavy combination vehicles have the potential to save energy, CO_2 and other transportation costs. Since the CO_2 emissions and fuel consumption are interdependent, which further depends on the rolling resistance of the heavy commercial vehicles, specifically their tyres.

Rolling resistance is the force lost when the tyres rotate under vertical loading as the truck moves when compared to the nominal transmission froce and torque. Over a long and continuous operating cycle, the reduction in rolling resistance brings down the costs of transportation significantly. Lower rolling resistance implies less loss of energy, lower fuel consumption leading to reduced CO_2 emissions. Tyres affect the energy consumption and road grip for acceleration, braking and lateral manoeuvres. All these measures are influenced by the vertical force or load on the tyres.

The lateral and longitudinal slip stiffness of tyres affect the performance of the vehicle. When the vehicle is accelerating, braking or cornering the tyre properties along with the slip stiffness come into play. Forces from the tyres are transferred to the road the execution of manoeuvres such as cornering, braking etc. largely depend on the capability of the tyres.

1.2 Purpose and Goal

This project will investigate the influence of vertical load on truck tyres, with respect to the rolling resistance and the lateral slip stiffness. There will also be an investigation to see if the vertical load influences the rolling resistance coefficient or other parameters that could affect the rolling resistance. The goal is to select a suitable tyre model for modelling of truck tyres.

1.3 Delimitations

In order to make this project feasible with regards to the scope of the work and the project timeline as well as man hours available, certain delimitations have been considered. They are as follows:

1. Dry paved roads are considered with friction utilisation within peak limits.
2. Variation of tyre normal force in the range of $0.5 \cdot F_{Z,Nom} < F_Z < 1.2 \cdot F_{Z,Nom}$.
3. Effects of variation in tyre inflation pressure and tyre temperature have been neglected.

1.4 Limitations

There have been a few limitations to the project, mostly related to the availability of test data. This makes it hard to verify theories and calculations, since there is a limited amount of data to use for verification. No physical tests can be conducted.

2

Theory

In order to investigate different tyre models and their scope of usability for this project, a literature review of the different types of tyre models used in the industry and the physics of each model was conducted. For the variation of rolling resistance, a study of roll-out tests was performed of trucks with lifted and non-lifted axles was reviewed. The findings of these reviews are presented in this section.

2.1 Lateral slip stiffness models

The models studied and considered for the lateral slip stiffness modelling are discussed in this section. For the sake of a simplistic comparison and to prevent scope creep w.r.t this project, this discussion mainly includes only two types of models, physical tyre models and curve-fit tyre models.

2.1.1 Physical Model

Physical models are based on the physics of the tyre-road interaction and available physical data for the tyre. The brush tyre model is a type of a physical tyre model in which the tyre treads are represented as bristles which deform over the longitudinal, lateral and vertical direction [4].
Based on the pressure distribution of the contact patch, the brush models can be classified into the following categories:

1. Uniform pressure distribution brush model.
2. Parabolic pressure distribution brush model.

Additionally, to explain the lateral dynamics of the tyre-road interaction, the brush model is classified based on the deformation of the bristles into the following categories:

1. Brush model with independent bristles.
2. Brush model with dependent bristles (String model).

The parabolic pressure distribution model is considered for the scope of this project for the physical tyre model, as it is a more accurate representation of the actual contact dynamics of a tyre.

Parabolic pressure distribution model:
Using the equation 2.30 from the Vehicle Dynamics compendium [3] for the lateral

dynamics, the lateral force F_y generated by the tyre is defined as:

$$-F_y \cdot sign(s_y) \;=\; C_y \cdot |s_y| - \left(2 - \frac{\mu_{slip}}{\mu_{stick}}\right) \cdot \frac{(C_y \cdot |s_y|)^2}{3 \cdot \mu_{stick} \cdot F_z} + \left(3 - 2 \cdot \frac{\mu_{slip}}{\mu_{stick}}\right) \cdot \frac{(C_y \cdot |s_y|)^3}{27 \cdot (\mu_{stick} \cdot F_z)^2} \tag{2.1}$$

else if $\omega \cdot v_x \leq 0$ or $|F_y| \leq \mu \cdot F_z$

$$-F_y \cdot sign(s_y) \;=\; \mu \cdot F_z \tag{2.2}$$

Where,

$$\mu_{stick} \;=\; \mu_{peak} \cdot \frac{(\frac{3-2}{k_\mu})^2}{\frac{4-3}{k_\mu}} \tag{2.3}$$

$$k_\mu \;=\; \frac{\mu_{stick}}{\mu_{slip}} \tag{2.4}$$

k_μ = Frictional ratio constant, μ_{peak} = Peak frictional coefficient.
C_y is the lateral slip stiffness of the tyre, which is evaluated using the expression shown below:

$$C_y = \frac{G_y \cdot W \cdot L^2}{2 \cdot H_y^2} \tag{2.5}$$

Where, G_y = Shear modulus of rubber, W = Tyre width [m], L = Contact patch length [m], H_y = Tread height [m]. The lateral slip of the tyre s_y is defined as:

$$s_y = \frac{v_y}{R \cdot \omega} \tag{2.6}$$

Where, R = Tyre radius [m], v_y = Lateral velocity of the tyre [m/s] and ω = Rotational speed of tyre [rad/sec].
In equation 2.5, the definition of the lateral slip stiffness includes parameters such as G_y, W, L and H which are intrinsic physical properties of a tyre with L being load varying as well. This makes it difficult to estimate them accurately for each tyre being modelled, since these are measured and experimentally determined quantities. For this reason, we use an alternate, more general method of calculating the lateral slip stiffness C_y referring to equation 2.48 from the Vehicle Dynamics Compendium [3] which is as given below:

$$C_y \;=\; \frac{F_z}{F_{zNom} + \frac{k_{Nom}-1}{F_{zNom}} \cdot {F_z}^2} \cdot C_{x,Nom} \tag{2.7}$$

$$C_x \;=\; \frac{F_z}{F_{zNom}} \cdot C_{x,Nom} \;\; and \;\; k_{Nom} = \frac{C_{x,Nom}}{C_{y,Nom}} \tag{2.8}$$

This way of evaluating the lateral slip stiffness C_y depends only on the nominal longitudinal and lateral slip stiffness C_{xNom} and C_{yNom} respectively.

2.1.2 Curve Fit Model

With the availability of multiple tyre models used for modelling tyre characteristics and performance, the preferment towards curve-fit models such as the Pacejka

(Magic Formula) has increased due to their more accurate response to tyre behaviour approximation compared to physical models, however over limited operating ranges. These models best approximate tyre behaviour over steady state operating conditions, whereas the physical models conversely seem to have a larger validity range. Given that the scope of this project follows certain de-limitations, the investigation of the curve-fit models is constrained to the *openPBS* curve-fit model, which is derived using Pacejka's Magic Formula. The comparison of this curve-fit model with the physical models discussed in the previous section serves as a good investigative and comparative approach for choosing an optimal tyre model.

The *openPBS* tyre model is developed to approximate tire behaviour in both linear and non-linear regions. The tyre lateral force calculated using the model in the linear region of operation of the tyre is evaluated as [1]:

$$F_y = -C_{st} \cdot \alpha \tag{2.9}$$

where C_{st} represents the tyre's cornering stiffness or lateral slip stiffness.

The approximation of the non-linear region of operation of the tyre by the OpenPBS tool is based on a simplified version of Pacejka's Magic Formula, which simplifies the number of parameters involved in evaluating the tyre's lateral forces. The steady state lateral force evaluated by the *openPBS* model in the non-linear region of operation is as shown below [2]:

$$F_{yt} = F_{ZT} \cdot u_y \cdot sin\left(C \cdot atan\left(\frac{CC_y}{C \cdot u_y} \cdot \alpha_y\right)\right) \tag{2.10}$$

where F_{ZT} is the tyre vertical force, u_y is the maximum lateral force coefficient (peak friction coefficient), C is a shape factor, CC_y is a tyre cornering coefficient and α_y is the slip angle. Tyre shape factor C is defined as:

$$C = 2\left(1 + \frac{asin(u_2)}{\pi}\right) \tag{2.11}$$

where u_2 is the ratio between infinite slide friction and peak friction and has a value of 0.8. The maximum lateral force gradient is modified by the vertical force:

$$u_y = u_{y0} \cdot \left(1 + ug_y \cdot \frac{F_{ZT} - F_{ZT0}}{F_{ZT0}}\right) \tag{2.12}$$

where u_{y0} is the tyre maximum lateral force coefficient at nominal load with a typical value of 0.8 [1] and ug_y is the maximum lateral force gradient at actual load that has a typical value between -0.1 and -0.3 [1].
Tyre cornering coefficient CC_y can be defined from the cornering coefficient at nominal tyre force CC_{y0} and the maximum cornering coefficient gradient ccg_y as follows:

$$CC_y = CC_{y0} \cdot \left(1 + ccg_y \cdot \frac{F_{ZT} - F_{ZT0}}{F_{ZT0}}\right) \tag{2.13}$$

Both the linear and non-linear *openPBS* tyre models include tyre relaxation, which determines how fast the tyre reaches steady state conditions. However, for the scope of this project, we consider the study of steady state operating conditions and so we do not include the transient effects of relaxation length.

2.2 Rolling resistance model

2.2.1 Constant rolling resistance

The fundamental theory of this report is based on the forces acting on the wheel illustrated below in figure 2.1.

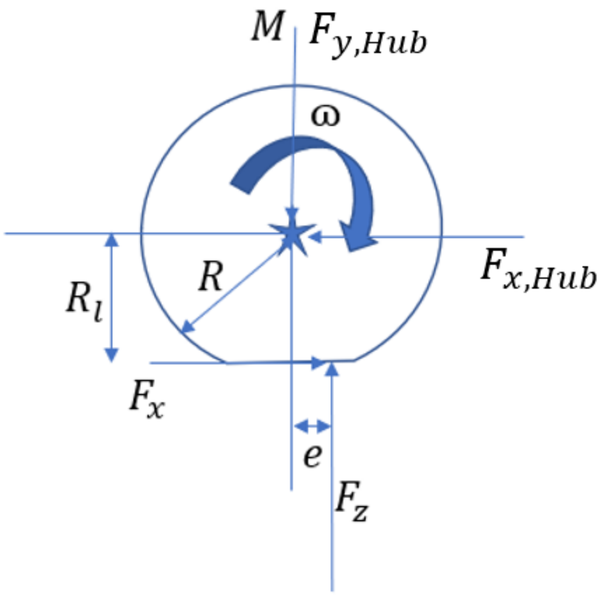

Figure 2.1: Forces acting on the wheel

In figure 2.1, M is the mass acting on the wheel, R is the radius of the wheel, R_l is the loaded radius of the wheel, e is the distance at which the forces act on the wheel.
ω is the angular velocity of the wheel, F_x is the longitudinal force, and F_z is the vertical force acting on the wheel.
The longitudinal force acting on the wheel is found by the expression,

$$F_x = \frac{T}{R_l} - \frac{e}{R_l} F_z \tag{2.14}$$

Where, T is the torque applied.
The rolling resistance co-efficient is further found by the expression,

$$rrc = \frac{F_x}{F_z} \tag{2.15}$$

Initial data is taken from the SAE paper where size of tyre used is 445/50R22.5 on the dry surface is used.
The variation of rolling resistance co-efficient is found by changing the various parameters like vertical load, tyre pressure (which varies the contact patch length of

tyre with road), which is further used to reduce the co2 emissions using the energy equations.
1. Variation of rolling resistance for different tyre measurements vs tyre speed

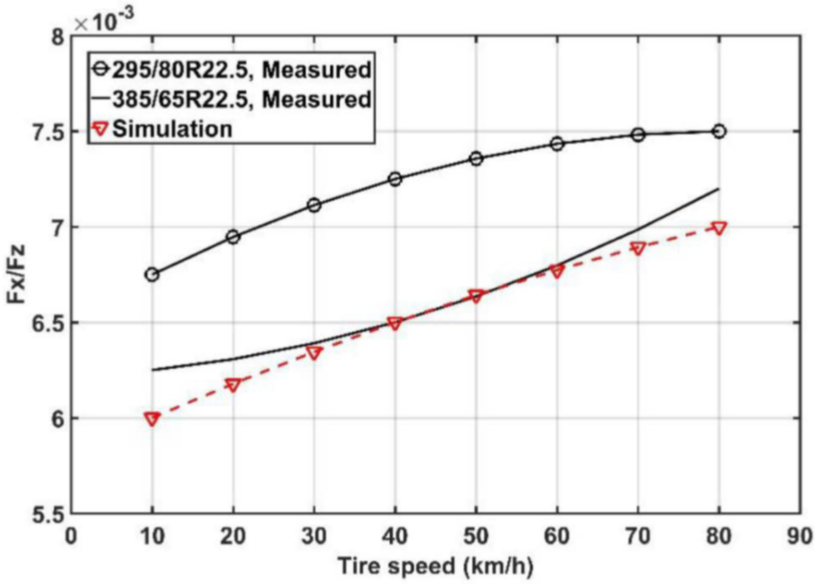

Figure 2.2: Rolling resistance vs tyre speed for different tyres[5].

We see that rolling resistance co-efficient increases with increase in speed, and deceases with increase in the width of the tyre.
2. Variation of rolling resistance for different tyre measurements vs tyre speed

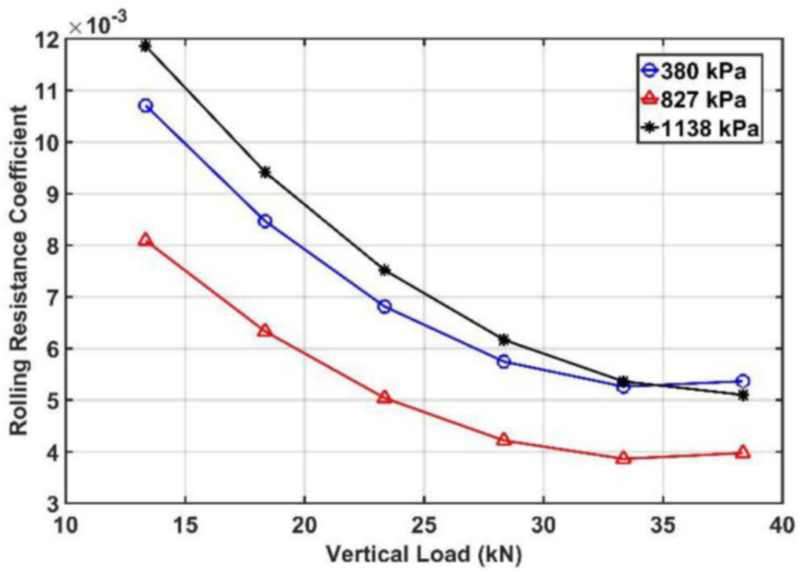

Figure 2.3: Rolling resistance vs vertical load for different tyre pressure.[5]

Rolling resistance co-efficient shows the decreasing nature with increase in vertical load. We also see that rrc is high when the pressure in too high or too low, for nominal load lower rrc is observed.

2.2.2 Varying rolling resistance coefficient

The theory of variable rolling resistance with respect to vertical load is commonly accepted and mentioned in several reports and studies. One study that evaluated the change in rolling resistance coefficient on trucks is a roll out test conducted by Lennart Cider at Volvo trucks [7]. The test procedure was to roll out a truck with a trailer at 80km/h and let it roll until a complete stop, then the same thing was done again but now with lifted axles on the truck and the trailer to increase the vertical load on the wheels still in contact with the road surface. During this procedure, measurements on the longitudinal forces acting on the truck was conducted. The different truck combinations used in the test is presented below in figure 2.4, the letter T stands for trailers and A stand for axles.

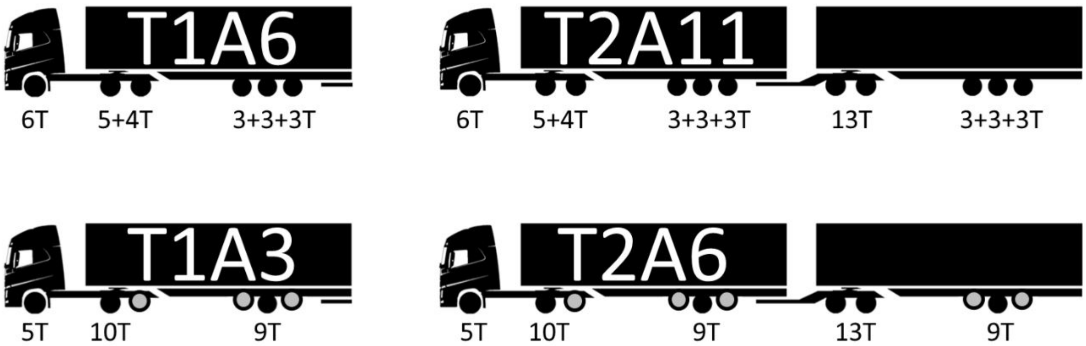

Figure 2.4: Vehicle fleet for roll out test

The result from the Ciders report regarding the change in rolling resistance coefficient is tabulated below in table 2.1.

Table 2.1: Rolling resistance coefficient results from Lennar Cider

	T1A6	T1A3	T2A11	T2A6
RRC	0,007	0,006	0,007	0,055

3
Methods

The methodology used to develop the tyre models included a literature review of the existing tyre models along with their merits, limitations and evaluation of use for the purpose of this project. The efficacy of each model was tested with the limited test data available, and then the suitable reference model was tuned to best match the available test data of the truck tyre. The testing and tuning of the model outputs was performed using MATLAB as it served the purpose for computations and graphical comparisons. In addition, the test data available from the roll-out tests [7] was tested for an alternate hypothesis to try and explain the varying RRC, which was done using MATLAB as well.

3.1 Lateral slip stiffness model

The general methodology followed for the physical and curve-fit model is the same. The process flow can be summarised in the following steps below:

1. Gather the test data from the VTI experiments [6].
2. Creating the reference model in MATLAB.
3. Identify the operating conditions, fixed parameters, tuning parameters and intermediate parameters.
4. Tuning of the model to best match the reference test data available.
5. Checking validity of the tuned model for larger operating ranges.

The reference data extracted from the VTI experiments [6] is shown in the image below:

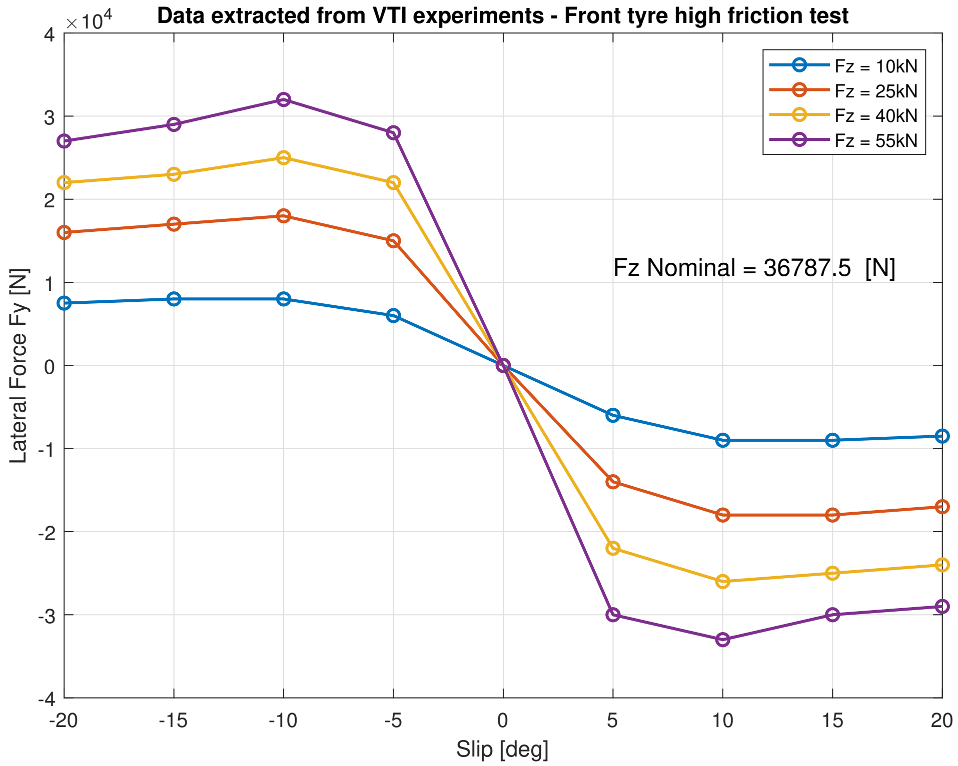

Figure 3.1: VTI experiment data for varying vertical loads

This plot shows the tyre lateral force vs. slip angle curves for four different normal loads, for tests performed on a high-friction surface. This data set was used as the reference for tuning both the physical and curve fit models. A point to note is that the data for the VTI experiments was extracted manually using a grid superimposition on the lateral force vs. slip plots available in the report [6], rather than generating the plot using explicit data points of slip and F_y. This results in the less "smooth" data plot as seen in figure 3.1 above.

Following this step, the MATLAB script for both models is developed using the respective reference models. The structure of the MATLAB script is similar for both the models, which includes the following components:

- **Initialisation function**: Defines the operating conditions, fixed parameters, tuning and intermediate parameters.
- **Main function**: Defines the physics of the lateral force generation by the tyre for each tyre model.
- **Plotting function**: Generates the relevant plots comparing the test data vs. the function output.

3.1.1 Physical Model

For the physical model, the brush model with parabolic pressure distribution was considered [3]. The *initialisation function* is defined using the following set of parameters:

Parameter Classification	Parameter	Description	Tuned/Used Value
Operating condition	F_z	Normal load on tyre	10,25,40,55 kN
Fixed	$F_{z,Nom}$	Normal load on tyre (nominal)	36.7 kN
	$C_{x,Nom}$	Nominal longitudinal stiffness	506265 N/slip
	μ_{peak}	Peak friction coefficient	0.65
Tuning	k_{Nom}	Slip stiffness constant	1.315
	k_μ	Frictional constant tuning factor	1.6
Intermediate	μ_{stick}	Stick friction coefficient	0.9368
	μ_{slip}	Slip friction coefficient	0.5855

Table 3.1: Parameter classification for the physical model

Using this parameter set, the lateral force generation function (*main function*) utilising the physics of the model was defined using the equation 2.1 presented in the theory section. Subsequently, the plotting function is defined to plot and compare the lateral force vs. slip plots for the varying normal loads.

Tuning the model:
Once we have the model generating output for all the vertical load cases, we tune the model using the tuning parameters k_{Nom} and k_μ to match the model curve trends with the test data. Since we have four distinct normal load cases of 10kN, 25kN, 40kN and 55kN, it is difficult to match all the curves simultaneously, and requires some numerical optimisation technique. Since the nominal vertical load of the test tyre for which the test data has been obtained is 36.7 kN [6], we aim to tune the model such that the best possible fit of the model output and the test data is obtained around the nominal vertical load. The closest available test data normal load case w.r.t this is the 40 kN case.

The k_{Nom} is defined as the ratio of the longitudinal and lateral slip stiffness of the tyre (ref: equation 2.8). We obtain the nominal longitudinal slip stiffness for the 40 kN load case from the VTI experiments data [6], which is **506265 N/slip**. Using this, we tune the model by performing a parameter sweep for k_{Nom} and k_μ. A representation of the parameter sweep performed for k_{Nom} is shown the figure 3.2 below. The solid red line with the circle markers represents the 40 kN vertical load test case from the VTI experiments [6].

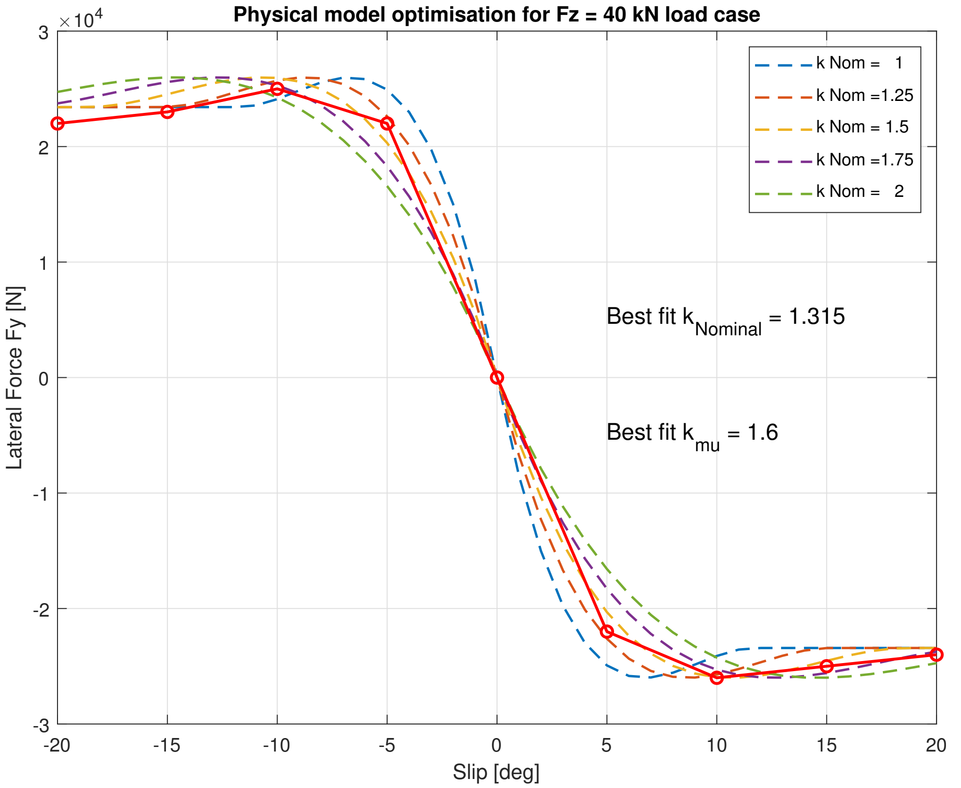

Figure 3.2: Physical model optimisation for 40 kN load case

The "best fit" curve for a given value of the tuning parameter (say for example, $k_{Nom} = 1.25$) is evaluated by visual comparison with the reference test data of the 40 kN load case. The range of the parameter sweep is started at a broad interval and narrowed down to inspect a closer fit to the reference data curve. The closest fit in the approximately linear range of the tyre (slip angle of -5 to 5 deg.) passing through the test data points at -5 and 5 deg of slip is chosen for the best fit criteria.

3.1.2 Curve-Fit Model

The process for modelling the curve-fit model is similar to the general structure of tuning the physical model as described in the previous section, which includes creating the model in Matlab based on the physics of the model, identify the *fixed parameters*, *tuning parameters*, *intermediate parameters* and *operating conditions*. Once this is done, we tune the model to match the VTI experiments data by manipulating the *tuning parameters*. Since we have the test data for four different normal load cases, we tune the model to match the test data for the nominal tyre load ($\approx$ 36.7 kN). The test data load case available that is closest to this value is the 40 kN load case [6], and hence we do tuning of the model to match that curve of the test data. The physics of the curve-fit model is based on the openPBS non-linear curve-fit model [2], which is based on Pacejka's Magic Formula.

The parameter set identified and classified for the curve-fit model is as shown below:

Parameter Classification	Parameter	Description	Tuned/Used Value
Operating condition	F_z	Normal load on tyre	10,25,40,55 kN
Fixed	$F_{z,Nom}$	Normal load on tyre (nominal)	36.7 kN
Tuning	ug_y	Maximum lateral force gradient	-0.2
	CC_{y0}	Cornering coefficient at $F_{z,Nom}$	11.648
	u_{y0}	Max. lateral force coefficient at $F_{z,Nom}$	0.8
	ccg_y	Max. cornering coefficient gradient	0
	u_2	Slide friction ratio	0.8
Intermediate	C	Shape factor	2.5903
	u_y	Max. lateral force coefficient	F_z dependent
	CC_y	Cornering coefficient	F_z dependent

Table 3.2: Parameter classification for the curve-fit model

Using this parameter set, the lateral force generation function (*main function*) utilising the physics of the model was defined using the equation 2.10 presented in the theory section. Subsequently, the plotting function is defined to plot and compare the lateral force vs. slip plots for the varying normal loads.

Tuning the model:

The tuning process for the curve-fit is the same as followed for the physical model, with the difference that the number of tuning parameters are five instead of two. This compounds the issue of tuning all of the parameters and cannot be done without the use of numerical optimisation, which risks scope creep for this project. For this reason, we proceed with considering conservative estimate values for the parameters u_{y0}, ccg_y and u_2 listed in the table 3.2 above to limit the complexity of the tuning and focus on CC_{y0} and ug_y as the primary tuning parameters. Also similar to the physical model, the tuning for the curve-fit model is done around the normal load case closest to the nominal normal load of the tyre, which is the 40 kN load case.

The parameter ug_y is the maximum lateral force gradient at actual load [2], which is 40 kN for the tuning case. The tuning is done with an initial value of zero and a parameter sweep on both sides of zero to check for best fit. Similarly, the parameter CC_{y0}, which is the cornering coefficient at nominal tyre force is tuned with a parameter sweep from 8 to 12, The range of values for the parameter sweep was considered based on the final values referred to from [2], and then the range was accordingly adjusted based on the trends of the plot for the 40 kN load case observed. A representation of the parameter sweep performed for the tuning parameter CC_{y0} is shown the figure 3.3 below. The solid red line with the circle markers represents the 40 kN vertical load test case from the VTI experiments [6].

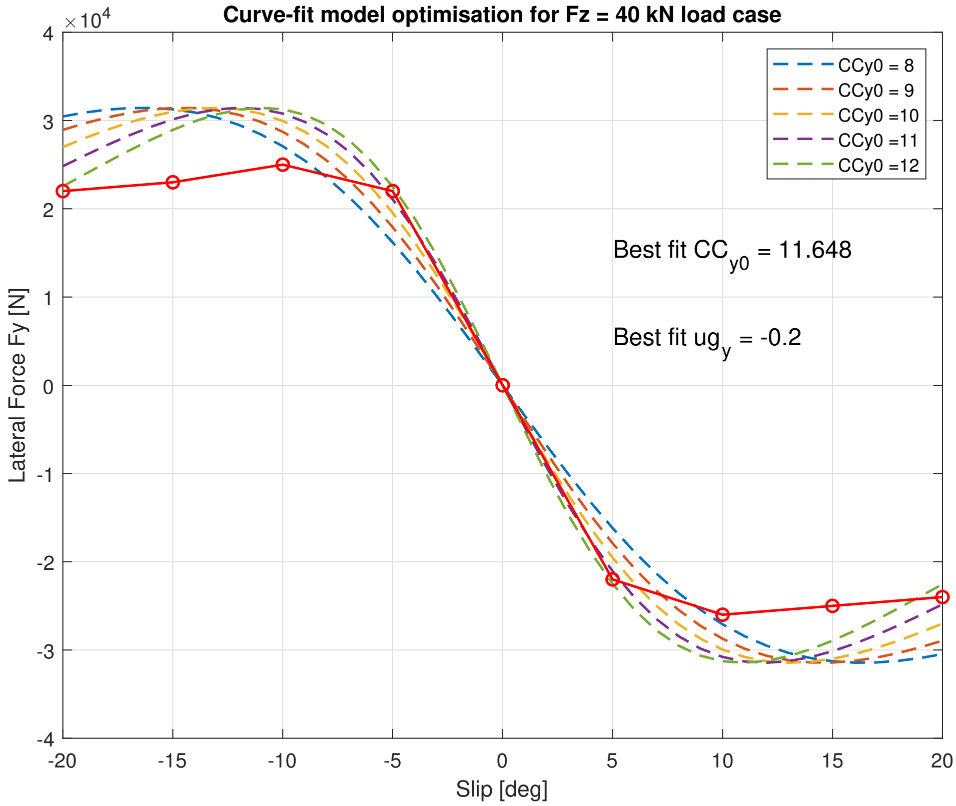

Figure 3.3: Curve-fit model optimisation for 40 kN load case

The "best fit" curve for a given value of the tuning parameter (say for example, $CC_{y0} = 11$) is evaluated by visual comparison with the reference test data of the 40 kN load case. The range of the parameter sweep is started at a broad interval and narrowed down to inspect a closer fit to the reference data curve. The closest fit in the approximately linear range of the tyre (slip angle of -5 to 5 deg.) passing through the test data points at -5 and 5 deg of slip is chosen for the best fit criteria.

3.1.3 Lateral Slip Stiffness

The goal for this part of the project is ultimately to determine to an optimal type of tyre model to calculate the lateral slip stiffness of a test tyre. To compare the lateral slip stiffness of the test tyre [6] evaluated by both type of the tyre models discussed above, we plot the same and compare them in a plot. The plot includes the data of the longitudinal slip stiffness acquired from the VTI experiments [6] and the lateral slip stiffness calculated using the physical model, the curve-fit model and the reference from the VTI experiments [6]. The data for the longitudinal and lateral slip stiffness from the VTI experiments includes only four distinct values corresponding to four distinct vertical load test cases, however the the outputs of the physical model and the curve-fit model is a continuous curve for varying vertical load cases.

3.2 Rolling resistance model

After reviewing several reports on rolling resistance and the related variables and components, a new theory was introduced. The theory was that the rolling resistance coefficient could in fact be constant, not varying with the vertical load as the theory presented in the report by Lennart Cider. Further the change in rolling resistance could be explained as a result of minimising the mechanical friction losses in the wheel bearings by lifting the axles, this quantity is referred to as ΔT. How this was investigated and evaluated will be presented below.
The first step was to backtrack the longitudinal force acting on the truck in order to get a better understanding of the forces acting on the system. The available data from the report related to the test that was investigated is presented below.

Truck	Mass (Kg)	RRCv	R (m)	g (m/s^2)
T1-6 axle	24000 (Kg)	0.007	0.537	9.81
T1-3 axle	24000 (Kg)	0.006		
T2-11 axle	46000 (Kg)	0.007		
T2-6 axle	46000 (Kg)	0.0055		

Table 3.3: Data from Cider report

Using the data from cider report, longitudinal force and vertical force are calculated initially using the following equations,

$$F_z = Mass * g \tag{3.1}$$

$$F_{x1} = RRC_{v1} * F_z \tag{3.2}$$

Where, F_{x1} is the longitudinal force of truck with all the axles performing.

$$F_{x2} = RRC_{v2} * F_z \tag{3.3}$$

F_{x2} is the longitudinal force of the truck with the lifted axle.
RRCc and ΔT are further calculated by solving the equations for trucks carrying same load with and without lifted axles.
The formula that is used to compute is:

$$F_{x1} = F_z * RRCc + x * \Delta T/R \tag{3.4}$$

$$F_{x2} = F_z * RRCc + y * \Delta T/R \tag{3.5}$$

Where, x is the number of axles in action on the truck with all axles performing.
y is the number of axles in action of the lifted axle truck.

-	Truck with all axles performing (x)	Truck with lifted axles (y)
Case 1	6	3
Case 2	11	6

Table 3.4: Values of x and y for two different cases

This resulted in a linear equation system with two equations and two unknowns for each truck, the constant rolling resistance coefficient and the delta torque. This was solved using MATLAB, the code can be found in appendix A.2.2 and the results are tabulated in the results section.

3.2.1 Investigating the Frictional losses in wheel bearings

By using SKFs bearing calculation tools the frictional losses of the wheel bearing could be estimated [8]. Since there was no available data on the specific bearings mounted on the trucks for the specific tests, a standard bearing for a Volvo FH16 was used as a reference [9]. Unfortunately the specific bearing was not available on SKFs website, a bearing with similar size was used instead. The friction losses where then calculated for the given values of Fz for the roll out test, the results are presented in the next chapter.

4

Results

In this section, we present the results obtained for lateral slip stiffness and rolling resistance investigations, their inference and the validity of these results.

4.1 Lateral slip stiffness model

4.1.1 Physical Model

The figure 4.1 below shows the comparison between the tuned physical model output and VTI test data for the different vertical load cases.

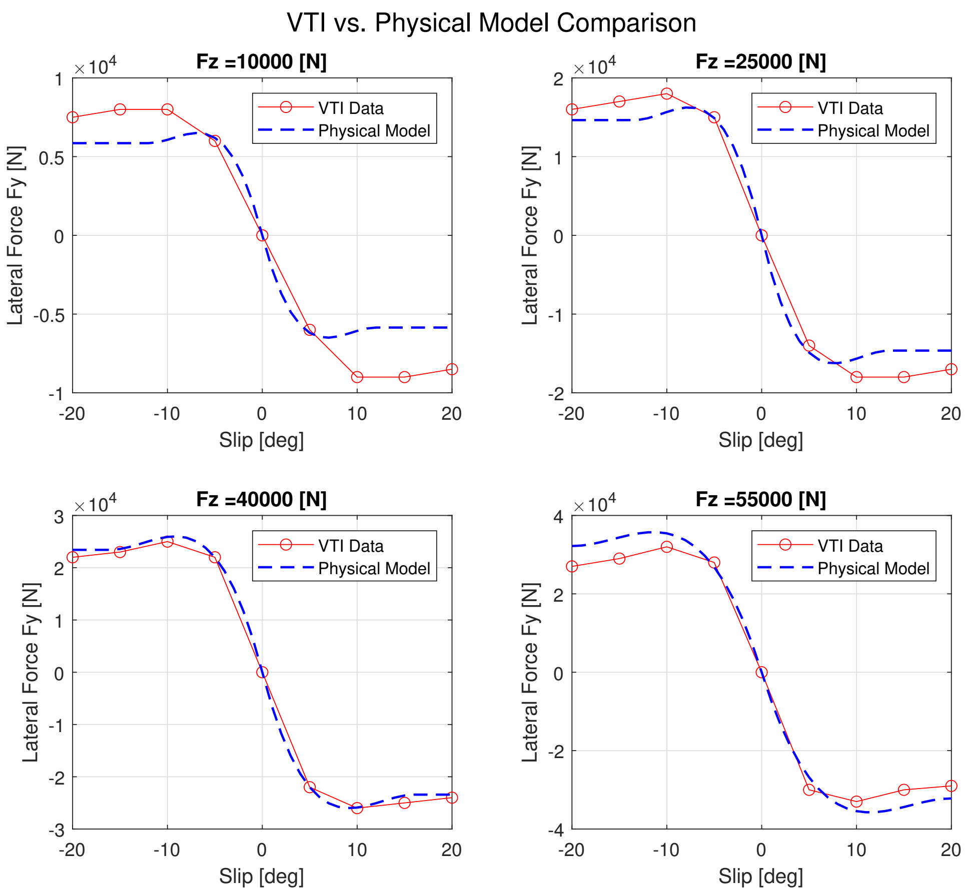

Figure 4.1: Physical model vs.VTI test data for different vertical loads

4.1.2 Curve-Fit Model

The figure 4.2 below shows the output of the tuned curve-fit model in comparison with the VTI test data [6] for the varying vertical load cases.

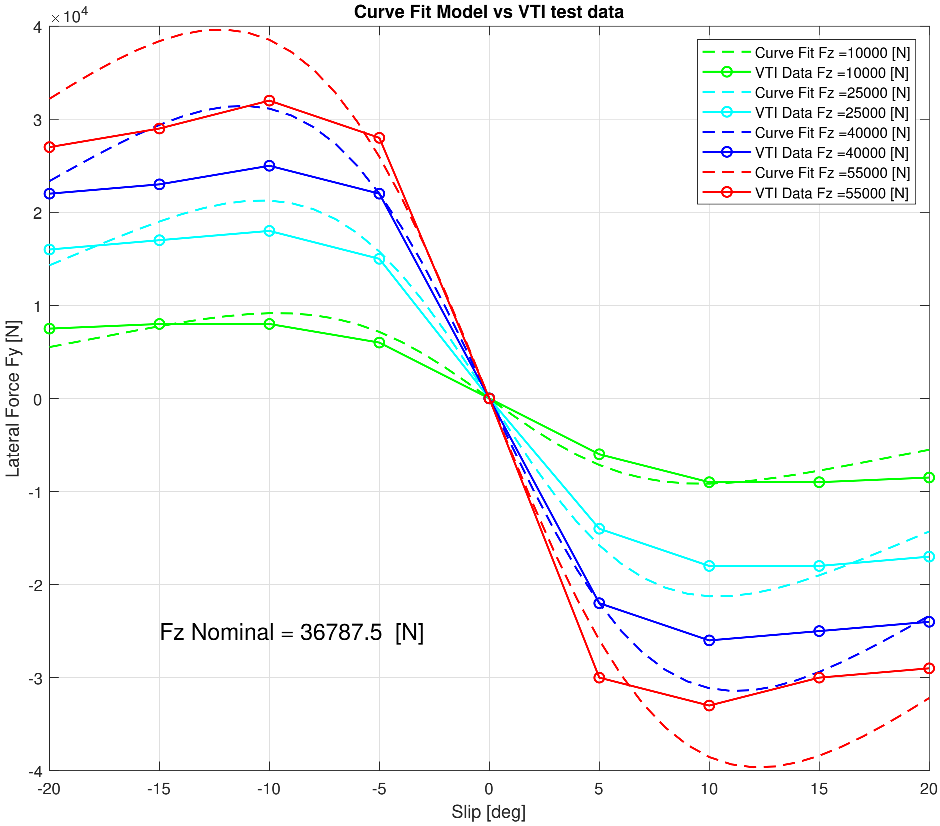

Figure 4.2: Curve-fit model output vs. VTI test data

4.1.3 Lateral Slip Stiffness

The figure 4.3 below shows the lateral slip stiffness output for the physical model and the curve-fit model and the comparison of these with the reference test data from the VTI experiments. As a measure of comparison, the longitudinal slip stiffness is also included in the plot to show the trend of the C_y w.r.t C_x. The curve-fit model shows a better conformance to the test data as compared to the physical model.

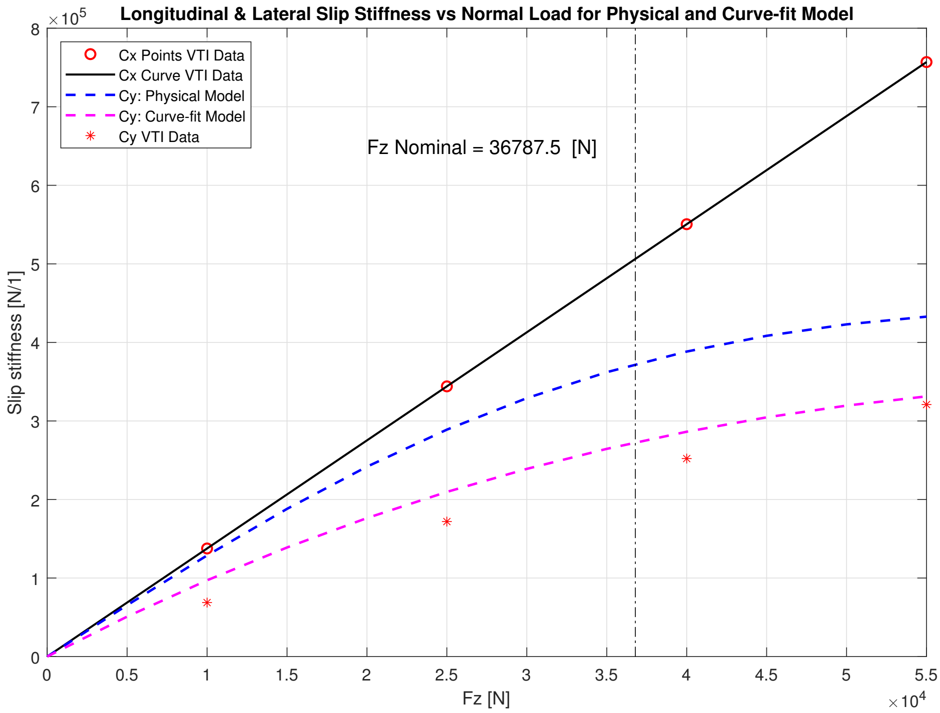

Figure 4.3: Lateral slip stiffness

4.2 Rolling resistance

The results fro RRCc and ΔT are as shown below

Truck	RRC	Fz total (N)	Fx total (N)	ΔT (N-m)	RRCc
T1-6 axles	0.007	235440	1648.1	42.15	0.00499
T1-3 axles	0.006	235440	1412.6		
T2-11 axles	0.007	451260	3158.8	72.42	0.00371
T2-6 axles	0.0055	451260	2481.9		

Table 4.1: Results for ΔT and RRCc

When the first truck runs with all of the axles in the lowered position there is a torque loss of 42.15 Nm per axle compared to when three of the axles are lifted. The rolling resistance was calculated to be 0.00499.
For the second truck with 11 axles the torque savings when the axles was in the lifted position was 72.42 Nm per axle. In this case the rolling resistance coefficient was calculated to 0.00371.

4.2.1 Bearing losses

The calculations from SKFs website is presented below in figure 4.4.

Designation	Bearing type	Principal dimensions			Basic load ratings		Fatigue load limit	Speed ratings	
		Bore	Outer diameter	Width	Dynamic	Static		Reference	Limiting
		d	D	B	C	C_0	P_u	n_{ref}	n_{lim}
		mm			kN			r/min	
HM 218248/210	Tapered roller bearing	89.975	146.9...	40	280	355	39	3400	4300

Designation	Frictional moment		Friction sources				Power loss
	Total	At start 20-30°C and zero speed	Rolling	Sliding	Seals	Drag loss	
	M	M_{start}	M_{rr}	M_{sl}	M_{seal}	M_{drag}	P_{loss}
	Nmm						W
HM 218248/210	2460	5710	2350	108	0	0	227

Figure 4.4: Bearing data and friction losses

The results above is from the calculations done for the second truck with 11 and 6 axles respectively. The nominal friction losses given from the SKF calculations gives a result of nearly 2.5 Nm per bearing it becomes almost 10 Nm per axle, which is around 13.5% of the total ΔT per axle.

5

Discussion & Conclusion

5.1 Lateral Slip Stiffness

The physical model (figure 4.1) and the curve-fit model (figure 4.2) have both been tuned for the test data available by tuning parameters specific to each model. The curve-fit model provides a better estimation of the lateral slip stiffness w.r.t the test data available as shown by figure 4.3, however more dense measurements in terms of slip angles and vertical load cases are needed to cement this finding.

5.2 Rolling Resistance

Some of the changes in rolling resistance coefficient could be explained by friction losses from the wheel bearings, however not nearly as much as anticipated. The remaining change in rolling resistance still needs to be accounted for, there could be other forces that could explain this phenomena. But for now the theory about a varying rolling resistance coefficient with respect to vertical load is not opposed, although some of the change in rolling resistance could be explained by losses in the bearings, it is not enough.

6
Future work

6.1 Lateral Slip Stiffness

The accuracy of the physical and curve-fit models can be improved by using optimisation techniques for tuning the parameters, to get a better fit for the available test data and increase the validity of the models. Since the test data available from the experiments was sparse, more dense measurements could be used, representing different types of truck tyres and the results could be analysed to check the trends for each model. A commonised parameter structure for both models could be created to implement modular modelling.

6.2 Rolling Resistance

For future work regarding the rolling resistance, it would be interesting to conduct a study on a real truck tyre and measure actual friction losses on a wheel bearing. In contrast to the nominal values provided by the bearing manufacturer. A finite element study of the tyre contact patch and the vertical load offset could increase the understanding of the phenomena why the pressure distribution is offset towards the tyre rolling direction. Further, the parameter identified in 3.2 could be extended to treat each individual axle, i.e. assuming RRC is linearly varying with F_z for each axle, as opposed to for the whole vehicle as done in present work.

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[6] Hjort Mattias, Banek Romuald and Bruzelius Fredrik. Documentation of Magic Formula parameterization of two truck tires on low friction (ice) and high friction (wet safety walk tape) surfaces, VTI.

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[8] Wheel bearing for Volvo FH16, `https://lastbilar.autodoc.se/skf/15725?gshp=1&gclid=CjwKCAiAt9z-BRBCEiwA_bWv-P3s7WtUS_4HKfuO22Nt7QSTuNsjRL15sNtkG6exrSDJUytgMpcgLRoCqtEQAvD_BwE`

[9] SKF wheel bearing calculation tool, `https://www.skfbearingselect.com/#/bearing-selection-start`

A

Appendix

A.1 Lateral slip stiffness model Code

A.1.1 Physical Model (Parabolic pressure distribution)

A.1.1.1 Parameterisation file

```
%% Instructions

% 1. Run this initialization script to load all parameters
% 2. Run Plotting_Function script to get the Fy vs slip plots
     for all four load cases
% 3. Run LateralSlipStiffness script to get comparison
     between Cx, Cy (physical model) and Cy (VTI data)

%% Initializing tyre parameters and operating conditions
clear; close all;
clc;

% Tyre Specification - 315/70 R22.5 (571.5mm) - Front Tire
% Nominal inflation pressure - 8 bar
% Nominal axle load = 7.5 ton (front axle)
% Nominal Fz = (7.5/2)*9.81 = 36.7875 kN
  (Closest available load case = 40 kN)

%% VTI Data

load VTI_front_tire_high_friction_data.mat
slip_dt = table2array(VTIFrontTireHighFrictionData(:,1));
LatFor = table2array([VTIFrontTireHighFrictionData(:,2)...
VTIFrontTireHighFrictionData(:,3)...
VTIFrontTireHighFrictionData(:,4)...
VTIFrontTireHighFrictionData(:,5)]);

%% ----------Operating conditions---------- %%

% Normal load on single tire [N]
```

```
StringModel.F_z = [10000 25000 40000 55000];

%% ------------Fixed parameters------------ %%

% Nominal tire load [N]
StringModel.Fz_Nom = 36787.5;
% Tire-road friction coefficient
StringModel.mu = 0.8;
% Longitudinal stiffness at nominal load
StringModel.C_xNom = 5.06265e+05;
% Peak friction coefficient
StringModel.mu_peak = max(abs(LatFor(:,3)))/...
StringModel.F_z(3);

%% ------------Tuning parameters------------ %%

% Slip Stiffness constant (Tuned manually for 40kN load)
StringModel.k_Nom = 1.315;
% Frictional constant tuning factor
StringModel.k_mu = 1.6;

%% ------------Intermediate parameters------------ %%

StringModel.mu_stick = StringModel.mu_peak*...
((3-2/StringModel.k_mu)^2)/(4-3/StringModel.k_mu);
StringModel.mu_slip = StringModel.mu_stick/StringModel.k_mu;
```

A.1.1.2 Function definition

```
%% Lateral Force Calculation - Brush Model with Parabolic
Pressure Distribution %%

function[Fy] = TyreSlipModel(Slip,F_z,StringModel,~)

C_y = StringModel.C_xNom.*F_z./(StringModel.Fz_Nom+...
(((StringModel.k_Nom-1)./StringModel.Fz_Nom).*F_z.^2));

if abs(Slip) < (3*StringModel.mu_stick.*F_z)/C_y

Fy = -sign(Slip).*(C_y.*abs(Slip)-(2-StringModel.mu_slip/...
StringModel.mu_stick)*((C_y.*abs(Slip)).^2./...
(3*StringModel.mu_stick*F_z))+(3-2*StringModel.mu_slip/...
StringModel.mu_stick)*((C_y.*abs(Slip)).^3./...
(27*(StringModel.mu_stick*F_z).^2)));

else
```

```
Fy = −sign(Slip).*StringModel.mu_slip.*F_z;

end

end
```

A.1.2 Curve-Fit Model

A.1.2.1 Parameterisation file

```
%% Instructions

% 1. Run this initialization script to load all parameters
% 2. Run PlottingFunction_CF script to get the Fy vs slip
plots for all four load cases

%% Initializing tyre parameters and operating conditions
clear; close all;
clc;

% Tyre Specification − 315/70 R22.5 (571.5mm) − Front Tire
% Nominal inflation pressure − 8 bar
% Nominal axle load = 7.5 ton (front axle)
% Nominal Fz = (7.5/2)*9.81 = 36.7875 kN

%% −−−−−−−−−−−−−−−−−−−−Operating Conditions−−−−−−−−−−−−−−−−−−−%%

TuningPar.Fz = [10000 25000 40000 55000];

%% −−−−−−−−−−−−−−−−−−−−Fixed Parameters−−−−−−−−−−−−−−−−−−−%%

TuningPar.Fz_Nom = 36787.5;  % Nominal tyre normal load

%% −−−−−−−−−−−−−−−−−−−−Tuning Parameters−−−−−−−−−−−−−−−−−−−%%

% Primarily tuned

% Maximum lateral force gradient
TuningPar.ugy = −0.2;
% Cornering coefficient at nominal load
TuningPar.CCy0 = 11.648;

% Conservatively approximated values

% Maximum lateral force coefficient at nominal load
```

```
TuningPar.uy0 = 0.8;
% Maximum cornering coefficient gradient
TuningPar.ccgy = 0;
% Slide friction ratio / mu_slip
TuningPar.u2 = 0.8;

%% ---------------Intermediate Parameters---------------%%

% Shape factor
TuningPar.C = 2*(1+asin(TuningPar.u2)/pi);
```

A.1.2.2 Function definition

```
function [Fy] = CurveFitModel(Fz,alpha,TuningPar,~)

uy = TuningPar.uy0.*(1+TuningPar.ugy.*...
(Fz-TuningPar.Fz_Nom)./TuningPar.Fz_Nom);

CCy = TuningPar.CCy0.*(1+TuningPar.ccgy.*...
(Fz-TuningPar.Fz_Nom)./TuningPar.Fz_Nom);

Fy = -Fz.*uy.*sin(TuningPar.C.*atan((CCy./...
TuningPar.C.*uy).*alpha));

end
```

A.2 Rolling resistance model code

A.2.1 RRC model

A.2.2 For truck with single trailer carrying load of 24 tonnes

```

clc
clear
close all

%% Rolling resistance model of the truck with trailer

% Initial parameters:
mass = 24e3;       % Weight of the truck (Kg)
g=9.81;            % Acceleration due to gravity (m/s^2)
R = 0.537;         % Radius of the wheel (meters)

```

```
RRC1 = 0.007;     % rrc of truck.
RRC2 = 0.006;   % rrc of truck with lifted axles.


% formulation for calculating vertical and longitudinal
 forces:

Fz1 = mass*g;     % vertical force acting on the truck (N)
Fz2 = Fz1-1;

Fx1 = Fz1*RRC1;  % Longitudinal force of the truck (N)
Fx2 = Fz1*RRC2;  % Longitudinal force of the truck with
                                          lifted axles (N)


e1 =Fx1*R/Fz1;    % The point at which the load acts
                                              on the wheel
e2 =Fx2*R/Fz2;    % The diantce at which the load acts
                          on the whel for lifted axle truck.


% To slolve for torque loss and constant
                    rolling resistance co-effiecint

syms delta_T RRCc

eq1 = Fz1*RRCc + (6*delta_T)/R == Fx1;
eq2 = Fz2*RRCc + (3*delta_T)/R == Fx2;

sol = solve([eq1,eq2],[RRCc,delta_T]);

% results for constant rolling resistance and torque loss

RRCc_out = double(sol.RRCc)  % constant rrc
delta_T_out = double(sol.delta_T) % Toque loss [N-m]
```

A.2.3 For truck with single trailer carrying load of 46 tonnes

```
clc
clear
close all

%% Rolling resistance model of the truck with trailer

```

```
% Initial parameters:
mass = 46e3;      % Weight of the truck (Kg)
g=9.81;           % Acceleration due to gravity (m/s^2)
R = 0.537;        % Radius of the wheel (meters)

RRC1 = 0.007;     % rrc of truck.
RRC2 = 0.0055;    % rrc of truck with lifted axles.


% formulation for calculating vertical and longitudinal
 forces:

Fz1 = mass*g;     % vertical force acting on the truck (N)
Fz2 = Fz1-1;

Fx1 = Fz1*RRC1;   % Longitudinal force of the truck (N)
Fx2 = Fz1*RRC2;   % Longitudinal force of the truck with
                                        lifted axles (N)


e1 =Fx1*R/Fz1;    % The point at which the load acts
                                             on the wheel
e2 =Fx2*R/Fz2;    % The diantce at which the load acts
                         on the whel for lifted axle truck.


% To slolve for torque loss and constant
                    rolling resistance co-effiecint

syms delta_T RRCc

eq1 = Fz1*RRCc + (11*delta_T)/R == Fx1;
eq2 = Fz2*RRCc + (6*delta_T)/R == Fx2;

sol = solve([eq1,eq2],[RRCc,delta_T]);

% results for constant rolling resistance and torque loss

RRCc_out = double(sol.RRCc)  % constant rrc
delta_T_out = double(sol.delta_T) % Toque loss [N-m]
```