CHALMERS

Modeling of Tyre Parameters' Influence on Transport Productivity for Heavy Trucks

Master's Thesis in Automotive engineering

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ABSTRACT

Today in Volvo Group Trucks Technology, in order to increase the accuracy and reliability of tyre selection, as well as to minimize the rolling resistance to lower the cost of operation, a PhD project 'TyreOpt – Fuel consumption reduction by tyre drag optimization' was started in 2012. As the phase one of this project, this master thesis was to construct a simple but accurate enough model to calculate the cost of operation for heavy trucks, specifically, fuel cost and tyre cost.

With the help of Matlab and Simulink, a mathematical model was built and it enables to select the best suitable tyre for a particular vehicle specification and operating environment. Besides, it can also preliminary demonstrate the relationship between each of the tyre parameters and cost of operation.

In order to ensure that other tyre dependent features are still on acceptable levels for specific vehicle and operating environment, three constraints equations were also conducted as a supplement to the model. They are startability, handling and ride comfort models, representing three basic vehicle dynamics in three directions.

Finally, a Simulink model was built to model fuel cost and tyre wear, and constraints equations were also created for basic dynamics performance. But future work is needed to improve the result accuracy due to the models are simplified.

Key words:
TyreOpt, tyre selection, fuel cost, tyre wear, cost of operation, startability, handling, ride comfort

CONTENTS

PREFACE

A dynamically excellent truck which is expensive to run is basically like a white elephant, it is not going to bring the necessary revenue to any company which develops it. Therefore it is essential to strike a good balance between the dynamic abilities of the truck and its running cost. The running cost of the truck can be broken down into the two key aspects which are the fuel cost and the tyre cost. This report focuses on developing a simple but complete model to help select the right tyre for the right job, i.e. to select the right tyre which offers the lowest fuel cost and tyre wear while being able to fulfill the dynamic requirements of the given task.

This thesis is a part of the 'TyreOpt' project being run at Volvo GTT. The project is funded by Swedish Energy Agency, and its main aim being to develop a method to select the right tyre for the given task rather than just pick it based on the information supplied by the tyre manufacturer. The thesis is done in collaboration with the Vehicle Dynamics group at Chalmers University of Technology under the guidance of Prof. Bengt Jacobson. The thesis is also guided by project manager Dr. Peter Lindroth of Volvo GTT and Zuzana Šabartová, a PhD student at Chalmers University of Technology. Zuzana Šabartová is working with the PhD Project titled 'TyreOpt – Fuel consumption reduction by tyre drag optimization' and the work done in this thesis will be carried forward by her during the course of her PhD.

This thesis was conducted between February 2013 and October 2013 as a requirement for the master program titled 'Masters of Science in Automotive Engineering'. During the course of this thesis we were guided by numerous individuals from both Volvo GTT and Chalmers University of Technology, apart from the aforementioned guides. This supervision is highly appreciated as it helped greatly to ensure the work was on the right track and of high quality.

NOTATIONS

Abbreviations

Volvo GTT	Volvo Group Trucks Technology
GTA	Global Transport Application
RRC	Rolling resistance coefficient
GVW	Gross vehicle weight
FEA	Finite Element Analysis
UOIT	University of Ontario Institute of Technology
VTM	Virtual transport model
CVM	Complete vehicle model
FEM	Finite element method
RMS	Root mean square
QSS	QuasiStatic Simulation
SEK	Swedish krona

Roman upper case letters

F_x	[N], Rolling resistance force
F_z	[N], Vertical (normal) force at the tyre-ground contact patch
F_G	[N], Gradient resistance
T_wheel	[Nm], Torque on the rear wheel
ACC	[m/s^2], Calculated acceleration from the speed information of the road
V	[m/s], Maximum limited Speed on the road
F_roll_f	[N], Rolling resistance forces on front tyres
F_roll_r	[N], Rolling resistance forces on rear tyres
Fzf	[N], Front normal force of the truck
Fzr	[N], Rear normal force of the truck

F_$aero$	[N], Aerodynamics driving force
C_d	[-], Aerodynamics coefficient
Air	[kg/m^3], Air density
A_f	[m^2], Truck frontal area
F_xf	[N], Total longitudinal force on front axle
F_xr	[N], Total longitudinal force on rear axle
C_x_f	[N/rad], Cornering stiffness on front axle
C_x_r	[N/rad], Cornering stiffness on rear axle
T_gb	[Nm], Torque of the gearbox
T_ice	[N], Torque of the internal combustion engine
T_∞	[°C], Ambient temperature
A	[mm/m], Wear rate
T_s	[°C], Tyre surface temperature
Q	[g], Total volume of wear debris produced
W	[N], Total normal load
H	[-], Hardness of the softest contacting surfaces
K	[-], Dimensionless contact
L	[m], Sliding distance
T_max	[Nm], Maximum engine torque from the engine map
K_{us}	[1/N], Understeer tyre gradient
L_f	[m], Distance between front axle and center of gravity of the truck
L_r	[m], Distance between rear axle and center of gravity of the truck
Fi0	[Hz], Road frequency
$Total_{price}$	[SEK], Total cost of operation
$Price_{fuel}$	[SEK], Fuel price

$Price_{tire}$ [SEK], Tyre price

Roman lower case letters

f_r [-], Coefficient of rolling resistance

c_slip_f [rad], Combined slip on front wheel

c_slip_r [rad], Combined slip on rear wheel

dv_wheel [m/s^2], Acceleration on the rear wheel

rrc_f [-], Rolling resistance coefficient on front wheel

rrc_r [-], Rolling resistance coefficient on rear wheel

rrc_f [-], Constant rolling resistance coefficient on front wheel used in startability

rrc_r [-], Constant rolling resistance coefficient on front wheel used in startability

$alpha$ [rad], Road inclination

s_x_f [rad], Front tyre slip in longitudinal direction

s_x_r [rad], Rear tyre slip in longitudinal direction

s_y_f [rad], Front tyre slip in lateral direction

s_y_r [rad], Rear tyre slip in lateral direction

toe_f [rad], Toe angle in front axle

toe_r [rad], Toe angle in rear axle

$upshift$ [-], Gear ratio upshifting vector

$downshift$ [-], Gear ratio downshift vector

w_{map} [rpm], Rotational speed vector of the engine

$gear\_ratio$ [-], Gear ratio vector

$gear\_final$ [-], Ratio of the final gear

dv_gb [m/s^2], Acceleration of the gearbox

fc [mg], Fuel consumption with an unit of ‘mg’

f [N/m], Wheel stiffness

r_0	[mm], Abradibility
c	[-], A material constant taken to be independent of temperature
t_0	[°C], Reference temperature
d	[mm], Spacing between abrasion patterns
r	[m], Radius of wheel rim
$wear_f$	[mm/m], Front tyre wear rate
nog_f	[-], Number of tyres on front axle
nog_r	[-], Number of tyres on rear axle

Greek lower case letters

ω_wheel	[rpm], Rotational speed on rear wheel
ω_gb	[rpm], Rotational speed of the gearbox
ω_ice	[rpm], Rotational speed of the engine
α_max	[rad], Maximum slope for startability calculation
θ	[rad], slip angle
ρ	[N/m], Resilience of the wheel
$\propto$	[-], Temperature coefficient
γ	[-], Hysteresis coefficient
β	[-], Correction coefficient
P	[psi], Standard tyre pressure
μ	[-], Maximum friction coefficient

1. Introduction

1.1 Background

The transportation industry is increasingly characterized by specialization and customization. Volvo Group Trucks Technology (Volvo GTT) therefore created a platform called 'Global Transport Application' (GTA) to have a common design language across different companies within Volvo Group and eventually to choose 'the right vehicle for the mission'. GTA defines a number of parameters that specify differences in driving and transport conditions for vehicle operations worldwide [1]. It helps to establish the optimum vehicle specification including selection of tyres in a more systematic way while fulfilling customer needs and lowering costs.

Currently at Volvo GTT, the selection of tyres is done by a methodology which greatly involves the information supplied by the tyre manufacturers. The required information is requested by Volvo GTT and provided by the tyre manufacturers. This does not offer complete control to Volvo GTT in helping the customer pick the right tyre as the accuracy of the data provided by the manufacturer is uncertain. Therefore a PhD project 'TyreOpt – Fuel consumption reduction by tyre drag optimization' was started in 2012. This master thesis is a part of the project and its main aim is to build models needed to pick the best tyre for the assigned conditions.

1.2 Aim

The aim of this thesis is to produce a model which can be used for optimization of tyre design and optimization of tyre selection. The model shall be simple enough to execute fast, but complete enough to cover all relevant objectives, i.e. costs and constraints. Specifically, the model shall reflect the influence of tyre design and selection on the cost of operation. And the cost of operation has two main components, which are fuel cost and tyre wear cost.

This principle can be expressed into a mathematical formula as shown in equation 1:

$$\min \quad f(x,p) = f_{fuelcost}(x,p) + f_{tyrewear}(x,p) \tag{1}$$

Where $'x'$ represents the tyre and $'p'$ represents the vehicle and the operating environment. Therefore, it can be understood that once the tyre is selected and the vehicle and operating environment is specified, the result of $f_{fuelcost}(x,p)$ as well as $f_{tyrewear}(x,p)$ can be calculated, and the sum of these factors is the tyre running cost of operation, which is aimed to be minimized as much as possible.

This function is subject to three constraints equations which limit the arguments of $f(x,p)$ to a reasonable range and as well fulfill all vehicle dynamics requirement. More specifically,

$$g_{startability}(x, p\ ) \leq c_{startability} \tag{2}$$

$$c_{handling\_min} \leq g_{handling}(x, p) \leq c_{handling\_max} \tag{3}$$

$$g_{ridecomfort}(x, p) \leq c_{ridecomfort} \tag{4}$$

These formulas representing three constraints in three directions, they can be calculated as a supplement from the main mathematical formula equation (1) above and the result will be compared in the later phase of the project with some existing models in Volvo to check the accuracy and reliability.

The fuel cost can be predicted by using a mathematical function of various parameters like rolling resistance coefficient (RRC), gross vehicle weight (GVW), operating cycle, topography, vehicle properties, tyre properties, etc. [2]. In this thesis, a RRC surrogate model is adopted to help to calculate rolling resistance coefficient (describe in details in section 2.2). Overall, the fuel cost calculation can be achieved by a Simulink model using reverse vehicle dynamics which will directly estimate the fuel consumption [2].

The tyre wear is quite complex to predict accurately but there have been various papers written by people over the years that give a good overview of the same [3] [4] [5]. Using these papers and other resources available at Volvo GTT a prediction of tyre wear can be obtained. The factors affecting tyre wear are mainly the slip angle, tyre stiffness, damping, road conditions, operating cycle; tread surface temperature, ambient temperature, etc. [6]. From these various parameters a simple enough model needs to be developed which uses the least parameters possible.

Besides, in order to ensure the optimum tyre is selected for specific driving environment and vehicle specification, it needs to satisfy various requirements under different circumstances. This is done by constraining the overall model to three very important parts of vehicle dynamics i.e. longitudinal, lateral and vertical dynamics. These are basically equations for startability, handling and ride comfort which will ensure that the vehicle can handle the task assigned to it. The reason for selecting these constraints is to simplify the overall model which can be evaluated in a reasonable time and they represent important boundaries such that the optimal solution will be acceptable. This simple model will be used in the next phase of the project where optimization algorithms requiring many evaluations of model will be implemented.

1.3 Delimitation

As the phase one of the 'TyreOpt' program, this thesis work is to create a simple model involving some basic tyre parameters to calculate fuel consumption and tyre wear. The reason for simplification is since the optimization will require a large number of simulations and there are many different customers with different vehicles and operating environments, leading many unique problems needed to be solved. Hence the accuracy level of the result is not high mainly due to:

(1) For the particular selection of vehicle and roads, not all the correct values can be obtained so some assumptions of the input data have to be made;

(2) A simplified model leads to some important input variables or relationships are missing.

For example,

- Only one truck is used and some components of it are not modeled precisely, like the engine map does not fit the engine adopted in this thesis, it obviously affects the fuel consumption prediction. In principle the result can still be accurate by increasing the input values accuracy.
- Two different sample roads are used, but data for fuel cost calculation and ride comfort model are obtained from different roads. So the result accuracy can also be raised by improving the road specification.
- Only three tyres are selected in this thesis, several basic parameters of tyres are used but some others are not covered here, such as material related parameters. Besides, the tyres are considered as non-tread which leads to an unreasonable tyre life.
- Constraints are only one example in each direction, they guarantee the basic dynamics performance in these directions but more important factors such as brake performance in longitudinal and durability in vertical direction are not covered. Besides, these models are simplified and tyres independent, it leads some constant results in tyre parameters' validation, see section 4.3.
- RRC surrogate model for the rolling resistance coefficient is used but only adopted in the Simulink model for the fuel consumption calculation. This model should be related to constraints equations to improve the result accuracy.

2. Information review

2.1 Tyre

In 1845, Robert William Thomson invented and patented the first pneumatic tyres in the world that consisting of a steel rim and an inflated rubber toroid [7]. As of today, tyres have been well optimized and widely adopted on cars, trucks, bicycles, etc. Generally, a tyre is a ring-shaped covering that fits around a wheel's rim to protect it and enable better vehicle performance. Most tyres, such as those for automobiles and bicycles, provide traction between the vehicle and the road while providing a flexible cushion that absorbs shock.

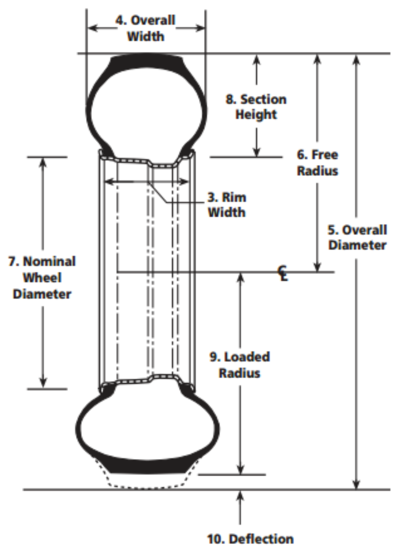

Figure 1 Illustration of tyre structure

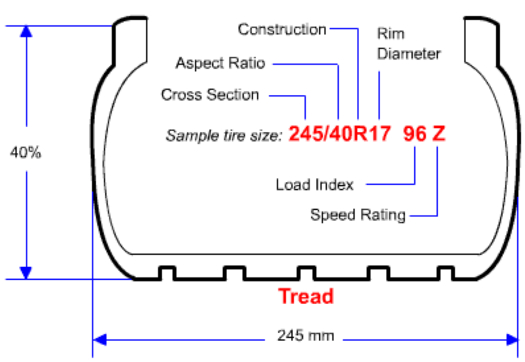

Figure 2 Illustration of tyre identification

The figure 1 above shows different specifications on a typical vehicle tyre and figure 2 explains the meaning of tyre identification. For a normal truck tyre, a series of tyre marking can be found on the tyre side wall, for instance, 245/40R17 96Z. The tyre marking determines the tyre basic characters and properties and it is also one of the references to help to tyre buyers to select the best suitable tyre. In other words, people can select tyres for different road conditions according to tyre marking, but at the same time a trade-off must be achieved between handling performance and comfort, between acceleration and wear, as well as between rolling resistance and desired friction for generation forces in ground plane [7]. For example, Volvo GTT has many tyre suppliers and customers often have unique transport missions and operating environment, therefore the tyre marking is probably not enough to be able to select the most 'suitable' tyres, some other factors such the tread depth, tyre materials and groove patterns need to be taken under consideration.

2.2 RRC surrogate model

In this thesis, a mathematical model to calculate the rolling resistance coefficient (RRC) was adopted to model the fuel consumption. Generally, the rolling resistance can be defined as the effort required keeping a given tyre rolling. It includes mechanical energy losses due to aerodynamic drag associated with rolling, friction between the tyre and the road and between the tyre and the rim; it also includes energy losses within the structure of the tyre. The rolling resistance is the mechanical energy that is converted into heat resulting from a tyre rolling a unit distance on the road.

The rolling resistance on a free-rolling tyre when there is no applied wheel torque and no side slip can be characterized through the RRC [8] as equation 5 below

$$f_r = \frac{F_x}{F_z} \tag{5}$$

Where f_r represents the coefficient of rolling resistance, F_x is the rolling resistance force and F_z is the vertical (normal) force at the tyre-ground contact patch, as shown in figure 3.

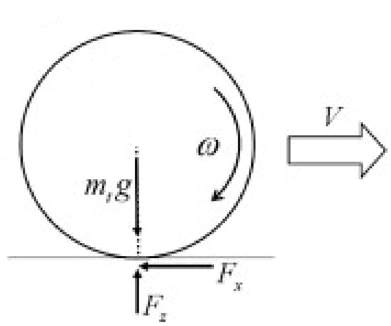

Figure 3 Rolling resistance analysis

Here, a Finite Element Analysis (FEA) model is introduced and ran to define both forces. As a result of cooperation between University of Ontario Institute of Technology (UOIT) and Volvo GTT we have access to a 3-Dimensional FEA truck tyre model [9]. This tyre model is detailed enough to differ between different tyre parameters settings even so it has many limitations, e.g., a free-rolling tyre when there is no applied torque was modeled.

In this model, there are four tyre design parameters (tyre inflation pressure, tyre width, tyre diameter, groove depth) and two operating parameters (vehicle speed, spindle load) get involved. Investigations done by UOIT shows how the RRC is influenced by them.

FEA models are usually very complex and require a long computation time to be evaluated; each evaluation of the rolling resistance coefficient function considering hard surfaces takes four to five hours of computing time. Considering this, a simple, compact and relatively fast model connecting vehicle, tyre and road is needed for further research. For a further research we need a simple, compact and relatively fast model connecting vehicle, tyre and road. The rolling resistance coefficient function will be part of this model and it has to be evaluated many times. Therefore, we need to create a model of the function describing the rolling resistance coefficient as function of design and operating parameters which can be evaluated in a reasonable time; we cannot use the FEA model directly. It will be used only to generate a set of sampled points and this set will be used further to generate a simple surrogate model, and it should be noticed that this surrogate model was conducted under a specific speed or a static load on the tyre model, there is no torque applied on the tyre which lowers the result accuracy when doing a real test on the vehicle.

The FEA model was run many times to receive a set of data points at which the rolling resistance coefficient varies due to changes in the parameters. Each parameter was varied between its upper and lower bounds determining a validity region of the model. A surrogate model of the rolling resistance coefficient is then based on the data set generated by FEA and can be trusted only within these bounds determining a validity region of the model, see the Table 1. The validity region was subsequently extended for the needs of complex joint vehicle, tyre and road model by using one dimensional extrapolation but the accuracy of the model is much lower in the added regions.

Table 1 Parameters and validity regions in RRC surrogate model

Parameter	Validity region	Extended validity region
Load [lbs.]	[3000; 9000]	[1000; 9000]
Speed [km/h]	[10; 100]	[10; 120]
Pressure [psi]	[55; 165]	-
Diameter [mm]	[917; 1120]	-
Width [mm]	[228; 455]	-
Groove depth [mm]	[0; 24]	-

The surrogate model was obtained using a radial basis functions interpolation of the simulated points. The resulting model of the unknown function is then a linear combination of linear radial functions [10]. The implementation of radial basis functions interpolation was done in MATLAB, and this RRC surrogate model can be then incorporated into the complex joint vehicle, tyre and road model.

2.3 Fuel cost

As one of the most important aspects in vehicle performance, fuel cost has always been paid attention to in modern automotive industry. Today, a connection between fuel cost and the vehicle itself has been correctly established and well modeled [11]. In the simplest terms, the fuel is burned inside engine cylinders to produce power, and eventually transferred to the driven wheels in order to overcome all the resistance forces that trying to retard or hold back the truck [12]. Therefore, almost all the working components on a vehicle have influences on overall fuel cost to different extent, like engine operating condition, aerodynamics force and tyre properties. As mentioned in RRC surrogate model, tyre rolling resistance results from the internal friction of a tyre as it deflects during motion, and it is the second most significant contributor to vehicle power requirement [12]. Therefore, one of the objectives of this thesis is to measure fuel cost for a particular truck by analyzing different influences on vehicle parameters, especially tyre properties.

Factors affecting the overall fuel cost can be categorized into two main parts: internal factors & external factors. More specifically, internal factors include all vehicle components and tyre properties, like vehicle speed, vehicle weight, tyre diameter, etc.; while all the influential factors ‘outside’ the vehicle can be treated as external factors, like aerodynamics force, road condition. In this thesis work, a RRC surrogate model [13] and a simplified GTA will be adopted to represent ‘internal factors’ and ‘external factors’ respectively. And eventually the overall fuel cost can be predicted and calculated by gathering these factors and this output can also be compared to an internal Volvo program 'PERF' to verify its accuracy.

From the GTA program, three blocks will be introduced namely (1) Transport mission (2) Vehicle utilization (3) Operating environment. One parameter from each of these three blocks will be studied in this fuel cost prediction model for simplification. Regarding internal factors, Vehicle and tyre blocks will be modeled but they would be combined and analyzed by adopting a RRC surrogate model which has been implemented. The figure 4 below shows the basic overview how the overall fuel cost can be calculated and figure 9 in the next chapter shows a diagram of Simulink model how the fuel cost is modeled as well as the tyre wear.

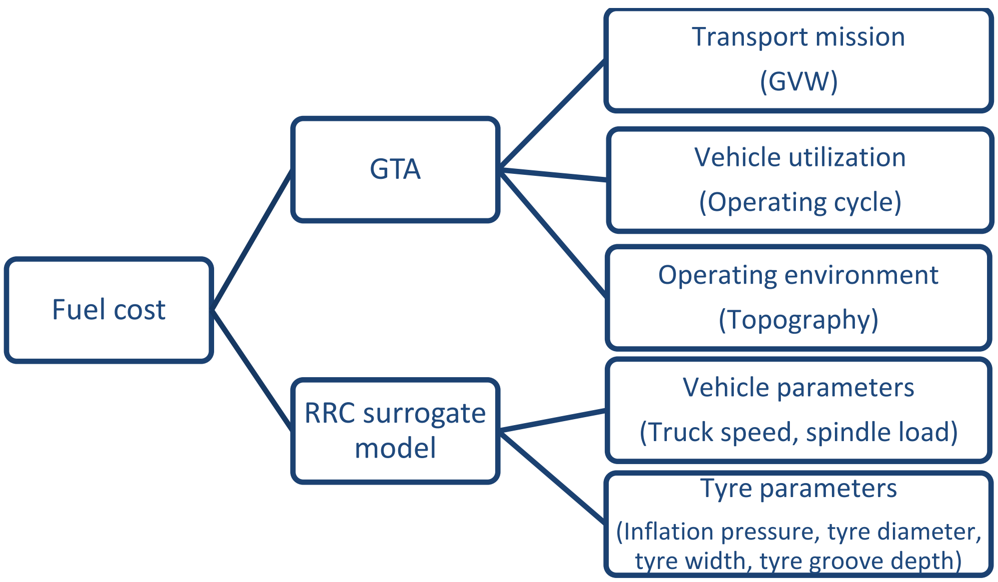

Figure 4 Diagram of fuel cost calculation

2.3.1 Transport mission (Gross vehicle weight)

The nature of the transport mission determines the vehicle type, size, capacity, gross vehicle/combination weight [1]. Among these specifications, the gross vehicle weight (GVW) has the greatest influence on fuel cost because an optimized GVW will enable high productivity, high operating reliability as well as low average costs [1].

It is quite obvious that a heavier truck will burn more fuel. Therefore the gross vehicle weight, which represents the total weight of a particular truck and influences other vehicle specifications such as weight distribution and tyre deflection, needs to be taken into account. With advancements in manufacturing techniques, reducing vehicle weight using lightweight materials leads to the possibility of downsizing the engine, which consumes less fuel while delivering the same level of performance. Verified assumptions have been made that 20% reduction of vehicle weight is responsible for about a third of the maximum fuel consumption reduction [14].

2.3.2 Vehicle utilization (Operating cycle)

The operating cycle basically describes how often the vehicle stops to load or unload goods or passengers [1]. This factor has a great impact on fuel cost as it directly affects the vehicle operating performance i.e., all the parameters like tyre pressure, engine speed, vertical load, etc., which are interconnected while measuring the fuel cost.

2.3.3 Operating environment (Topography)

In GTA, operating environment includes the 'external' parameters, such as road parameters and weather conditions. Among these parameters, topography is considered as the most important one since the vehicle must have enough power to overcome the earth's gravity when tackling hills and other difficult terrain [1]. For a road with a constant road slope α, the gradient resistance can be calculated by

$$F_G = mg \sin \alpha, \tag{6}$$

where m is GVW, therefore, it means with the increase of gradient slope, more and more fuel needed to be consumed to overcome this gradient resistance force to climb slope.

2.3.4 Tyre

This thesis mainly focuses on investigating tyre parameters' influences. Some tyre parameters such as inflation pressure, tyre geometry, tread pattern and depth, directly affect the fuel cost to different extents. In the RRC surrogate model, four tyre important parameters, namely tyre width, tyre diameter, tyre groove depth and inflation pressure have been taken into consideration [13].

Rolling resistance is the force resisting the motion when a vehicle is driving on the road. A previous study shows tyre rolling resistance is the second most significant contributor to vehicle power requirements, followed after aerodynamics drag [12]. In the RRC surrogate model, the output, rolling resistance coefficient is influenced by the tyre design (tyre inflation pressure, tyre width, tyre diameter, groove depth) and the operating parameters, truck speed, spindle load), as shown in figure 4 above.

2.3.5 Vehicle

For vehicle and tyre themselves, this thesis work will connect them together by using RRC surrogate model. This surrogate model has been represented by creating relationship between rolling resistance coefficient and several vehicle or tyre specifications. Also, the main component through the driveline such as gearbox and engine are added and modeled to calculate the fuel consumption.

Despite all the external factors, the vehicle itself has a great influence on fuel cost. In the RRC surrogate model, vehicle speed and GVW have been taken into consideration to calculate RRC. Also, they affect the fuel consumption since they affect the aerodynamics force, vehicle weight distribution, suspension performance and so on.

2.4 Tyre Wear

The modeling of tyre wear has been in debate for many decades now. It is obvious that there is a direct effect on the running cost of the vehicle because of the wear of the tyres. This wear has been modeled by various people with numerous different approaches using different parameters [5] [6] [15]. The various parameters include rolling resistance, temperature, slip angle, wheel stiffness, contact patch dimensions, etc. Therefore it is clear that defining a single tyre wear formula which can be justified using the data available at Volvo can be intricate.

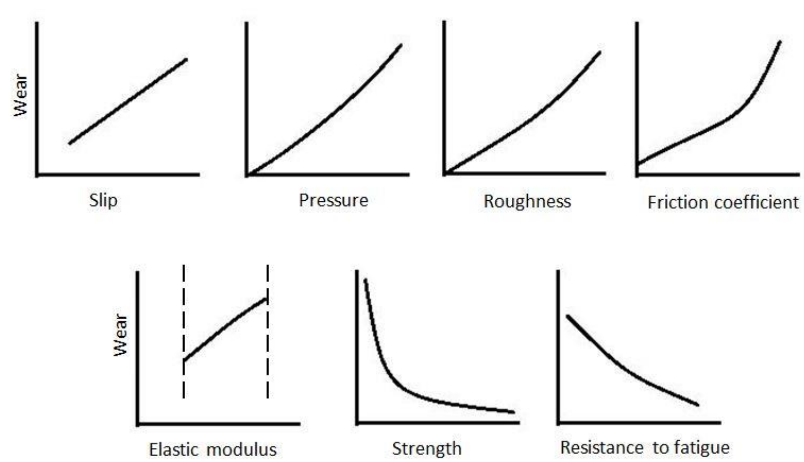

Figure 5 The dependence of wear on the main parameters

Figure 5 shows a summary and verification of how the various factors affect the wear properties of a given tyre [6]. This is also a good reference to use when modeling, to select the right parameters for the wear equation. It must be remembered that these parameters also depend upon each other.

Three main approaches for wear have been modeled in the past which give a different insight on how wear depends on different parameters. All these models have their own advantages and disadvantages.

2.4.1 The Schallamach Approach

This paper focus on modeling of tyre wears using slip [5]. According to Grosch and Schallamach the wear could be determined by using the abrasion pattern of the individual tyre and also the temperature [5]. This model corresponds with the conclusions drawn from the multiple interviews conducted at Volvo GTT and also the Volvo GTA [1]. The origin of forces that cause slip in the wheels can be rolling resistance, cross-winds, the inertial forces accompanying acceleration and braking, cornering, etc. [5].

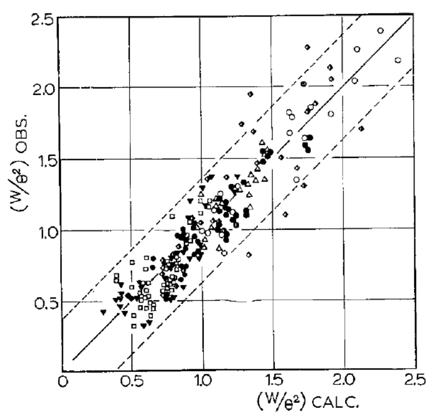

Figure 6: Observed to measured values of wear

The figure 6 represents a comparison of the wear calculations to the experimental values and the dotted lines are the 95% confidence lines [5]. Thus it can be stated that this approach is quite accurate. But, there are a few constants used by Schallamach which need to be provided by the tyre manufacturer and also some other information like the distance between the abrasion pattern and the temperature that need to be directly measured for each tyre. All these unknowns make this equation have a lot of assumptions when modeled mathematically.

2.4.2 The Kraghelsky & Nepomnyashchi Approach

This concept is quite different from the approach taken on wear by the other authors. The formula derived for wear in this method takes into account the effect of strength, fatigue factors, modulus of elasticity, coefficient of friction, load and roughness upon wear under sliding conditions [3] to deduce an equation of wear for wheels rolling with slip. The main kind of wear of tyres under normal conditions is caused by fatigue due to interactions with the rough surfaces [3]. The drawback of this equation is that, apart from the fact that it does not include temperature modeled in it, it is necessary to have all the above mentioned material characteristics measured during some independent experiments. To validate the formula for sliding conditions the results were compared by the authors to experimental data from existing literature. For the case of rolling with slip lab experiments were conducted on a range of rubbers which mainly differed in modulus of elasticity.

2.4.3 The Bulgin & Walters Approach

Bulgin & Walters use a combination of fatigue and abrasive wear and setup an empirical relationship of tyre wear [6]. This approach therefore can be considered to the better than the above two as it is obvious that wear is caused under both these conditions. This approach also has the advantage of taking care of change in slip angle but this also acts as a limitation as it is the only important parameter which affects the outcome. This is better represented by the figure 7.

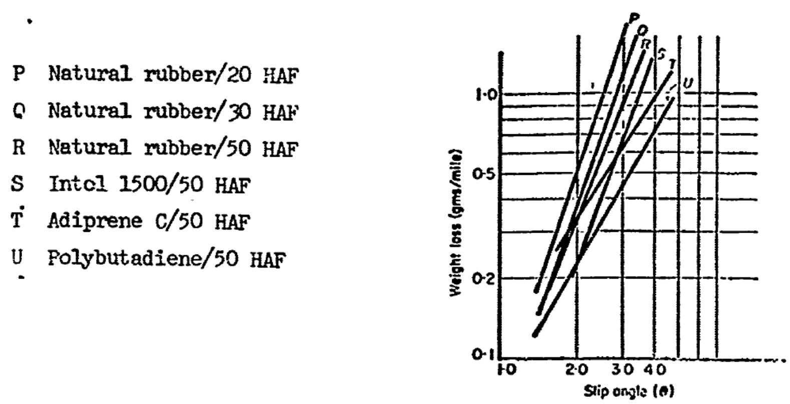

Figure 7: Logarithmic graph showing wear increases as power of slip angle (Different lines represent different tyre types)

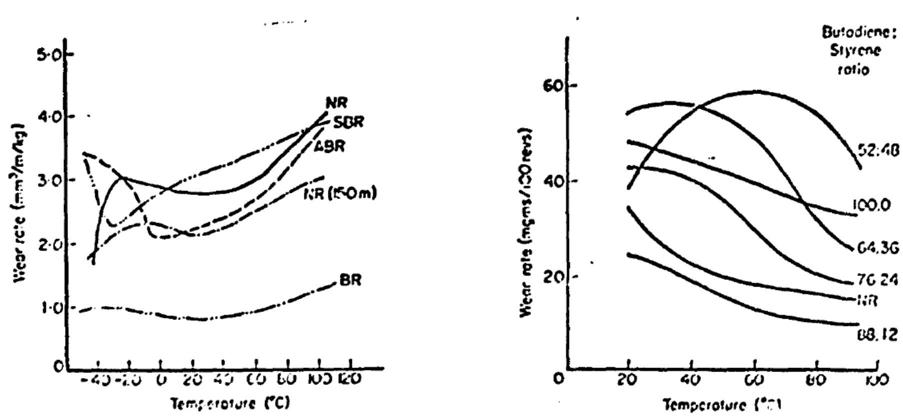

Figure 8: Wear Rates on Rough (left) and Smooth (right) surfaces (Different curves represent different tyre types)

The approach taken by Bulgin & Walters studied the wear rate on two extreme surfaces as show by figure 7. They also found that the wear rate in case of smooth surfaces is proportional to the power of the applied stress and inversely proportional to the carbon content and temperature. However the proportionality between the wear and the factors mentioned in the previous sentence are only true in a certain regime as seen in the above figure 8 [6].

2.5 Constraints equations

After the Simulink model to predict fuel consumption and tyre wear is established, a series of constraints equations needed to be set in order to fulfill the vehicle dynamics requirement. This makes sure the truck can run in a required operating environment in this assigned job. For simplification, the model in this thesis work will be constrained by examples from one of each of the three vehicle directions.

(1) Startability in longitudinal direction
(2) Handling in lateral direction
(3) Ride comfort in vertical direction

During the pre-study phase, a series of existing models (see appendix 7.2) which involve different specified vehicle components have been found and studied for each of the constraints equations mentioned above. But in order to keep this model simple, these constraints models have also been compared then the most suitable components will be adopted into our own model.

2.5.1 Startability

Startability is one of the most critical longitudinal performances which indicate the ability to start from stand-still and maintain steady forward motion on specified grade when operating at maximum laden mass. It results from the inclined road which creates a gradient resistance force along the vehicle motion direction. For a particular truck and road friction, a maximum slope angle can be calculated.

During the pre-study phase, two existing models have been studied for startability constraints equations:

(1) Virtual transport model (VTM) which is the best available model at Volvo GTT
(2) Startability model which uses a half vehicle model to calculate the maximum slope by setting a group of equilibrium equations in terms of normal force, momentum, etc.

The table 7 in appendix 7.1.2 shows how different components specified in different models. It also shows the operating conditions, quality and computational resources for the same. In this thesis work, Startability model is simplified by using two formulas representing the total resistance force and total tractive force [16], and then they are adopted to calculate the maximum slope. But RRC is assumed as constant instead of using RRC surrogate model due to the invalid speed range in the RRC surrogate model. This will be introduced in details in the next chapter.

2.5.2 Handling

For handling, three existing models which both being used at Volvo GTT have been studied in pre-study phase, i.e. VTM, CVM (complete vehicle model) and theoretical calculation.

(1) VTM uses quite a typical representation of the vehicle; i.e. point mass for the body, cab and the axle, frame is a 2 point mass connected by a spring. It can be used for various operating environments but is accurate for road conditions of only up to 5Hz.

(2) CVM uses a finite element method (FEM) model for the vehicle whose complexity is fixed depending on accuracy of the result required. It can be used also for high frequencies of the road disturbances, but with a fixed constant speed. It is mostly used for on-center handling analysis without large sweeping motions within the linear tyre range.

(3) The theoretical calculation, which was an outcome of a discussion with a Volvo GTT engineer [17]. It basically uses the difference in slip gradients of the front and back wheels respectively to find out if the vehicle handles within required limits.

The table 8 in appendix 7.1.2 gives more details of these models.

After carefully studying these models and discussion with the engineer at Volvo GTT it was decided, for the reason of simplicity to stick to theoretical calculation which has the least factors and has been proved accurate [17] .

2.5.3 Ride comfort

Ride comfort is a very important dynamics factor in the vertical direction. The selection of the wrong tyre can affect the ride comfort of the cab and similarly the selection of the right tyres can completely change the ride characteristics of a truck from harsh to smooth and vice versa.

The main models compared in this case are:

(1) VTM in this case is rarely used as its results cannot be considered for road frequencies of above 5Hz.

(2) The Virtual approach uses a 3-D vehicle model which can be disassembled into six parts to measure ride comfort. This model can minimize the cost for design which can be passed on to customers. The disadvantage is that it takes a lot of computation time as it runs on ADAMS, a multi-body dynamics simulation tool [18].
(3) Half vehicle model is simple model which can be seen as an extension of the quarter car model but with more degrees of freedom. This model is of good accuracy and can be used for road disturbances up to 15Hz [19].
(4) CVM is the most accurate model used at Volvo GTT. It can be used for high road frequencies and also for speeds of 110 km/h with high accuracy. The disadvantage is its high computation time [20].

The table 9 in appendix 7.1.2 gives more details of these models.

The half vehicle model [19] is used in this master thesis as it is mathematically simple to compute at the same time being well modeled by Volvo GTT. By using different road profiles parameters like velocity, road spectra density and inclination, the acceleration in vertical direction can be calculated by measuring output spectra and root mean square (RMS) value.

So a series of constraints equations representing vehicle dynamics performance in three directions were decided. A better comparison with existing tools should ideally have been done which could involve more parameters and came out with better result; therefore some further studies are needed in this area in the later phase of TyreOpt project.

3. Methodology

The whole point of this thesis is to develop a simple way to minimize the overall cost of operation of the truck while not compromising on its dynamic properties. The methodology gives an overview of how the models for fuel consumption and tyre wear were finalized, which will be studied in section 3.2 and 3.3. Then section 3.4 introduces how the constraints equations for the models were decided upon and how these equations were implemented.

3.1 Objectives definition

Before running the Simulink model (see figure 9 below) and the constraints equations, all the necessary parameters need to be predefined, i.e. truck, road and tyre parameters. This step is to save the parameters in the workspace. Then the Simulink model and constraints equations will share the same workspace to calculate results, like fuel consumption and tyre wear, which secures that all costs and all constraints equations are evaluated for the same specification.

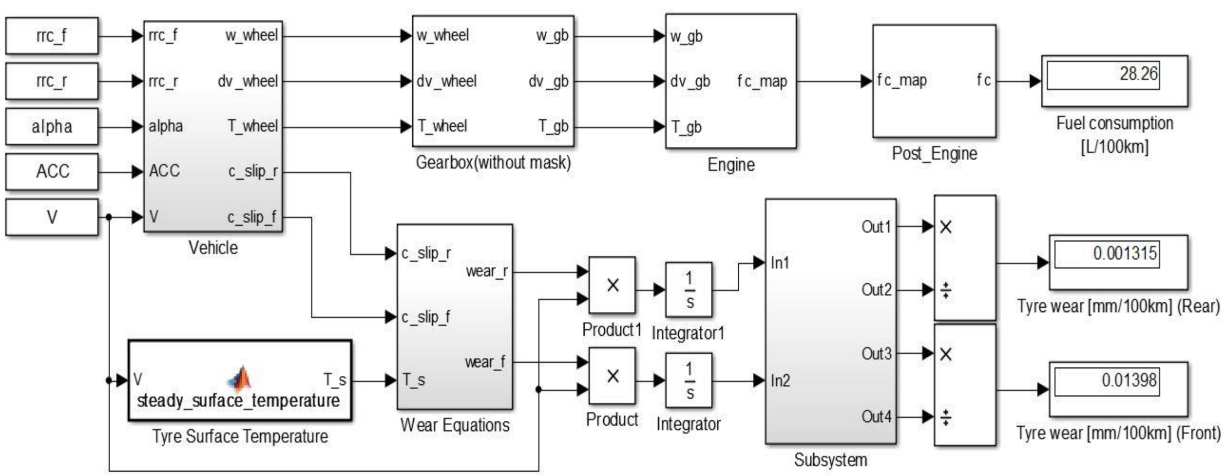

Figure 9 Diagram of Simulink model

Figure 9 shows the overall diagram of Simulink model which is created to calculate fuel consumption and tyre wear. It can be seen that after defining all the necessary specifications, the model will go through the vehicle block which is used to calculate the 'external' forces then it will break down into two separated paths to model fuel consumption and tyre wear respectively. The first path is a basic reverse driveline while the second path adopts the tyre wear equations as well as the tyre surface temperature calculation to obtain the tyre wear. Sub-blocks in this Simulink model will be studied in details in this chapter later.

3.1.1 Truck definition

A Volvo FH 4*2 rigid truck was introduced in this project as a reference. In the overall model, this truck has been separated into several parts with corresponding names, including gearbox, engine, center of gravity calculation, external parameters. All truck data come from reality. This truck can carry about eleven tons of goods. Here, two kinds of different goods have been defined: liquid and solid. Specifically, the liquid goods is distributed from bottom to top in the loading platform which means position of the center of gravity for the load does not change; in contrast, the solid good is distributed from the beginning to the end of the loading platform, thus the position of the center of gravity for the load will move backward with increasing weight of goods, see the figure 10 below. Besides, the cornering stiffness, which will be used to model tyre wear, is also defined here with the help of vertical load on front and rear axle.

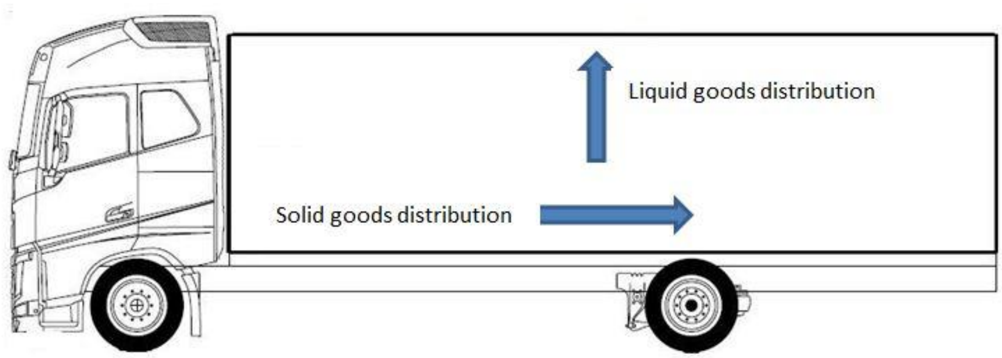

Figure 10 Diagram of Volvo FH goods distribution

3.1.2 Road definition

Once the truck is featured, it is also necessary to introduce some sample roads. In this thesis, two sample roads (Road 365 and Road 521 from Volvo program PERF) with road height and road maximum limited speed as a function of road longitudinal position information are used. Road 365 represents a hilly road which road 521 is a sample of flat highway. With these data, the road inclination and speed profile as a function of time can be calculated. It is interesting to notice that the acceleration is calculated from speed profile, so it would include several acceleration/deceleration points with very high values. However, these two roads are used only for fuel consumption purposes. They does not include any road curvature information and the sample points for collecting speed and height data are quite far apart, so they do not include frequency inputs or road waviness which are important for ride comfort model. Therefore some typical frequency data [7] are superimposed to these two roads to calculate the RMS value, more details will be studied later in the section 3.4.3. Figures representing Road 365 and 521 can be found in appendix B.

3.1.3 Tyre definition

As the main object in this thesis, tyre is defined as a separate file in the model. According to the suggestion from Volvo GTT, three different truck tyres have been introduced. Firstly, six tyre design parameters (normal load, tyre inflation pressure, truck speed, tyre diameter, tyre width and tyre groove depth) are defined to calculate rolling resistance coefficient using RRC surrogate model. Then other parameters such as wheel dynamics radius, wheel rim radius are listed or assumed to model tyre wear. After setting up the truck, tyre, road specification, all the variables will be saved in the workspace before running the Simulink model to calculate the fuel consumption and tyre wear.

3.2 Fuel consumption

As mentioned in previous chapter, this thesis focuses on constructing functions which would enable minimize tyre running cost by optimization of various parameters. Thus, as the biggest part of the cost operation at least for many transport applications, fuel consumption needs to be calculated and modeled in this project. According to vehicle dynamics principle, a reverse vehicle driveline has been created with the aid of Simulink [2]. Each block will be introduced below.

3.2.1 Vehicle block

First of all, as a common block for both fuel consumption and tyre life path, five outputs, namely ω_wheel, dv_wheel, T_wheel, c_slip_f, c_slip_r are calculated from corresponding inputs (rrc_f, rrc_r, $alpha$, ACC and V) by vehicle dynamics equations.

F_roll_f and F_roll_r are rolling resistance forces on front and rear tyres respectively, which can be calculated using equation 7 and 8 below.

$$F\_roll\_f = rrc\_f * Fzf * \cos(alpha) \tag{7}$$

$$F\_roll\_r = rrc\_r * Fzr * cos(alpha) \tag{8}$$

rrc_f and rrc_r are the outputs from RRC surrogate model, representing the rolling resistance coefficient on front and rear tyre respectively, Fzf and Fzr are the corresponding front and rear normal forces; $alpha$ is the road inclination obtained from the road profile.

The aerodynamic driving force F_aero which is proportional to the square of vehicle speed can be calculated by equation 9.

$$\mathrm{F}_{aero} = 0.5 * C_d * Air * A_f * V^2 \tag{9}$$

The parameters C_d, Air, A_f and V represent the aerodynamics coefficient, the air density, the truck frontal area and the truck speed, respectively.

The gravitational resistance force F_grade can be obtained by equation 10.

$F\_grade = m * g * sin(alpha)$ (10)

Here, m is the total mass of the truck, including the weight of the load. g is gravitational acceleration (9.81m/s^2), $alpha$ is the road inclination angle.

Due to the truck is rear wheel driven; the total force on front axle is considered only the rolling force.

$F\_xf = F\_roll\_f$ (11)

The tractive force on the rear axle is equal to the sum of all resistance forces, which is given by equation 12 below:

$F\_xr = m * ACC + F\_aero + F\_grade + F\_roll\_r$ (12)

ACC is the truck acceleration.

Regarding the tyre life calculation, the combined slips at front and rear tyres are calculated here as well.

For longitudinal direction,

$s\_x\_f = F\_xf/C\_x\_f$ and $s\_x\_r = F\_xr/C\_x\_r$ (13)

Where C_x_f and C_x_r are the cornering stiffness on front and rear axle respectively.

For lateral direction,

$s\_y\_f = \tan(toe\_f * \pi/180)$ and $s\_y\_r = \tan(toe\_r * \pi/180)$ (14)

toe_f and toe_r are the toe angle in front and rear axle. In this model, toe_f is given from Volvo GTT as constant value and the value of toe_r is assumed to one fourth of the toe_f.

Hence, the combined slip can be obtained by equation 14 and 15.

$c\_slip\_f = \sqrt{s\_x\_f^2 + s\_y\_f^2}$ (15)

$c\_slip\_r = \sqrt{s\_x\_r^2 + s\_y\_\mathrm{r}^2}$ (16)

Once the combined slip value is obtained, the torque, rotational speed and acceleration of the wheel can be calculated using the following equations:

$T\_wheel = (F\_xf + F\_xr) * R$ (17)

$\omega\_wheel = V/(R * (1 - s\_x\_r))$ (18)

$dv\_wheel = ACC/(R * (1 - s\_x\_r))$ (19)

Here R is the tyre radius, V is the truck speed from the road definition.

Thus, five outputs (ω_wheel, dv_wheel, T_wheel, c_slip_f, c_slip_r) have been obtained in this vehicle block and from here the Simulink model will break down into two path to calculate the fuel consumption and tyre wear.

3.2.2 Gearbox block

The gearbox block is modeled by using an existing block in QuasiStatic Simulation Toolbox (QSS) as a reference [2], which is a Simulink-based simulation tool. After predefining the gearbox's parameters in the truck file, three outputs can be calculated (i.e. the torque, acceleration and rotational speed of the gearbox). The gearbox block design can be found below.

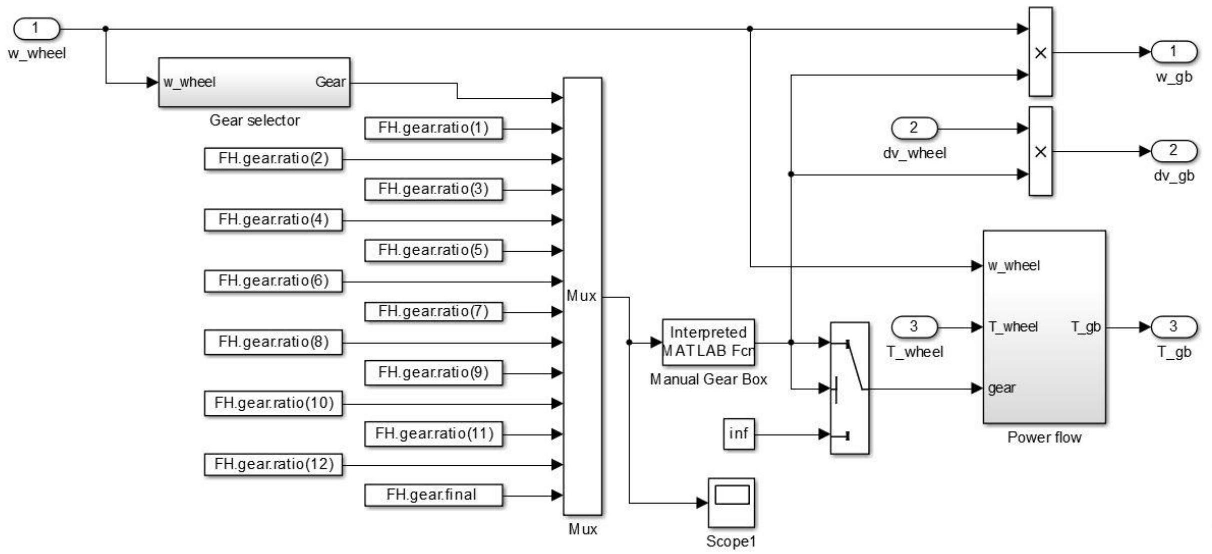

Figure 11 Diagram of the gearbox block

The gear selector sub-block uses gear upshift and downshift vector. According to the range of the engine rotational speed and the gear ratio vector, which can be obtained from the truck specification, the gear upshift and downshift can be roughly calculated using equation 20 and 21.

$$upshift = 0.8 * (w_{map(end)} * 3.6 * \pi * R/30)/(gear\_ratio(2: end) * gear\_final) \quad (20)$$

$$downshift = 0.8 * (w_{map(end-1)} * 3.6 * \pi * R/30)/(gear\_ratio(1: end - 1) * gear\_final) \quad (21)$$

The unit of $upshift$ and $downshift$ vector is km/h. Here, w_{map} has been defined as a vector from 800 to 2100 rpm with unequal steps. Specifically, $w_{map(end)}$ is 2100 rpm and $w_{map(end-1)}$ is 1900 rpm. Also, $gear\_ratio$ is a gear ratio vector with 12 elements and $gear\_final$ represents the ratio of the final gear. In the Simulink model, once a certain value of is ω_wheel is input, the switch point with a unit of 'km/h' can be calculated by the following equation:

$$Switch\ point = \omega\_wheel * R/3.6 \quad (22)$$

Then use this switch point to compare with $upshift$ or $downshift$ vector according acceleration or deceleration. For example,

If $upshift(5) \leq switch\ point < upshift(6)$ (23)

Then $upshift(5)$ will be chosen to get the corresponding gear ratio by using equation 20 and 21 above.

Thus, the proper gear ratio can be obtained to calculate three gearbox's outputs namely ω_gb, dv_gb, T_gb, representing the rotational speed, acceleration and torque of the gearbox, respectively.

3.2.3 Engine & Fuel tank blocks

By using the three outputs from the gearbox block as inputs to the engine block, this engine block can predict initial fuel consumption with the aid of corresponding engine map. Here an old engine map is introduced here due to confidentiality at Volvo GTT, but this does not have considerable influence on the model as the objective here is to prove the model works and gives a sufficiently accurate results. The engine map can be easily switched in later by whatever the situation calls for.

In the Simulink dynamics system, at each time signal the engine block will get one ω_gb and one T_gb value respectively. After going through a 'pre-engine' sub-block which is introduced to scale these two inputs into a proper range, ω_gb and T_gb become w_ice and T_ice, in order to locate the corresponding fuel consumption point at this vehicle operation moment.

Therefore the fuel consumption depending on the engine speed ω_ice (rpm) can be calculated by using equation below:

$$fc = 3 * A * \omega\_ice/60 \tag{24}$$

Here, A represents any fuel consumption point for one cylinder in the engine map(mg/stroke), so for each engine revolution, the fuel consumption of this six-cylinders engine is 6 A/2=3 A,this is because for a typical four-stroke engine, it needs two engine revolution to complete one operating cycle. So the overall fuel consumption with the unit 'mg' can be integrated through the whole road profile.

Regarding the fuel tank block, it is used to transfer units from 'mg' to eventually 'L/100km'. Finally, the overall fuel consumption can be calculated according to different user inputs. It also varies by changing different roads and tyres specifications.

3.3 Tyre wear

3.3.1 Calculation principle

The aim was to develop a simple tyre wear equation which is fairly accurate and helps to provide a good estimation of overall tyre cost. This arises the question of how simple is simple enough while providing results that can be justifiable. Mainly three different models were chosen to be studied during the initial phase of the thesis of which the Schallamach model was chosen as it included temperature and this factor was concluded as one of the major factors affecting the wear from the various discussions with experts at Volvo. The model was based around the Archard wear equation and the A. Schallamach and K.A. Grosch tyre wear equation at controlled slip [15]. The combination of Archard's and Schallamach's equation was chosen as for the overall model as it was noticed during modeling that the Schallamach equation missed important wear parameters which were present in the basic wear equation of Archard. The Schallamach wear equation can be represented as:

$$A = \theta^2 * \rho * f * r_0 * [1 + \propto * (t_s - t_0) + c * d] \tag{25}$$

Here A is wear rate with unit mm/m, it shows that the wear rate can be affected by a series of parameters, such as slip angle θ, wheel stiffness f, temperature and so on.

And the Archard wear equation is given as:

$$Q = \frac{K*W*L}{H} \tag{26}$$

Here Q is the total volume of wear debris produced, it is proportional to normal load W and sliding distance L, while reversely proportional to hardness of the softest contacting surfaces.

The similarities between the two equations are quite apparent when they are compared above so they were combined to create the wear equation used in this model. The value for surface temperature was found out by the equation provided in the paper 'Analysis of impact factors of tyre wear [4] where the necessary validation is done to prove the temperature achieved is very similar to the real surface temperature. The slip was assumed to be combined slip and found out with the basic slip equations [7].

The tyre temperature equation is stated as:

$$T_s = \frac{0.0447*\gamma*V^{0.16}*F*d^{-0.5}*[\frac{\alpha_1*F}{2*P} + \sqrt{\left(\frac{\alpha_1*F}{2*P}\right)^2 + \alpha_2*F}]^{0.5}}{\pi*\beta*[2*d*c+0.4*(d^2-r^2)+0.4*(d^2+d*r+r^2)-0.6r*(d+r)]} + T_\infty \tag{27}$$

The equation is rather complicated; all the variables are explained in the notations. It can be noticed that the main factors which affect the surface temperature are vehicle speed and the size of the tyre.

The Overall outcome used in the code is:

$$wear_f = \frac{(K*{c_{slip_f}}^2*C\_x\_f*(1+\left(alpha_x*\left(T_s-T_{inf}\right)\right)+(\beta*\sigma)))}{1000*1000} \tag{28}$$

In this equation, $wear_f$ is the wear rate of front tyre with unit mm/m, c_{slip_f} is combined slip on front tyre, which was calculated in the vehicle block. T_s is tyre surface temperature, calculated by equation 27 above.

The whole concept described in the above equation was stored as a Simulink model. The whole model had mainly three input structs in which all the common data is input, the data possessed in these structs are then transferred to the workspace, here they can be used by the different equations as required.

The c_{slip_f} was calculated in the vehicle block and was provided to the block consisting of the wear equation. The surface temperature was also independently calculated and provided as an input. But the combined slip and the surface temperature are directly dependent on the vehicle load, speed and the rolling resistance coefficient. The wear rate is converted from mm/m to mm/100 km to match the output of fuel consumption from the Simulink model. In the overall model the tyre cost and tread depth is added which helps us to achieve the running cost of the tyre/100 km and also the overall running cost of the tuck per 100 km.

3.3.2 Assumptions and limitations

There are a lot of factors like resilience of the wheel, wear rate constant, spacing constant between the abrasion pattern and temperature coefficient which changes with the change in material of the tyre, these factors are assumed in model developed here. The reason behind this is that the tyre companies do not provide this information publicly and it is hard to convince the company to provide it, therefore these values are picked up from old wear papers which written in the 1960's because these papers seem to have the values for a few tyres. Requesting and getting this information from the tyre companies can decrease the number of assumptions made in the equation and provide a more accurate result, also because it could be hard to measure these values. It is also hard to measure these parameters as they can change from tyre to tyre and it is also highly dependent on the manufacturing process of the rubber and the tyre itself - same tyre from different factories can give different values of these material parameters. At least one material parameter should be added in a further research.

Though the Simulink model had both the front and rear wear modeled into it for the final result only the front tyre wear value is used. This is due to the fact that when Schallamach developed the wear equation it represented the wear of a tyre which is just rotating with a specific speed and a point load. This also applies to the rolling resistance model used in the overall model where there is no torque considered. The truck is rear wheel driven and there is an added factor of torque at the rear wheels which is more complex to implement at this stage, it is definitely an area which can be worked upon in the future. This may not have such a big impact however because after speaking to many experts at Volvo it was concluded that the front wear is always greater than the rear wear because of the higher slip angles up front, during turning the slip angles at the rear maybe greater but the time period for which these slip angles are experienced because of long haul operation are very small when compared to the overall running of the truck and hence have negligible effect on the wear. This can however be different in the case of other operational cycles and needs further looking into.

3.4 Constraints equations

The constraints equations help to ensure that the dynamic abilities of truck stay acceptable when the running cost is minimized. They are equations of Startability, Handling and Ride comfort which justify the truck's dynamic ability in the X, Y and Z directions. These equations are dependent on much less factors than in the model developed.

3.4.1 Startability

The aim of the startability constraints equations is to model the vehicle longitudinal dynamics then further define a maximum slope which this truck can climb under a particular tyre specification. By setting up a series of equations of motion, the maximum slope as well as some other vehicle parameters can be obtained. However, the rolling resistance coefficient is not adopted from the RRC surrogate model because the range does not cover the low speed, i.e. 0-10 km/h. In this case, constant values for rolling resistance coefficient have been assumed for front and rear axles respectively.

To obtain the maximum slope, two main equations are used here, representing the total resistant force and the total tractive force respectively.

$$F_x = rrc_f * Fzf * \cos(\alpha\_max) + rrc_r * Fzr * \cos(\alpha\_max) + m * g * \sin(\alpha\_max) \quad (29)$$

$$F_x = \min([\mathrm{T\_max} * gear\_ratio(1) * gear\_final/R, Fzr * \mu * \cos(\alpha\_max)]) \quad (30)$$

The symbol rrc_f and rrc_r are the constant rolling resistance coefficient on front and rear axles respectively, differ from rrc_f and rrc_r which used to model fuel consumption.

In equation 29, the sum of the resistance force is equal to rolling resistance force at both front and rear axles and the gravitational force.

In equation 30, the tractive force is chosen from the smaller value between the force produced from engine maximum torque $'\mathrm{T\_max} * gear\_ratio(1) * gear\_final/R'$ and the friction force in longitudinal direction $'Fzr * \mu * \cos(\alpha\_max)'$. Here, μ represents the maximum friction coefficient using magic tyre formula [21]. Thus, once the weight of load is known, Fzf and Fzr can be calculated so the maximum slope α_max then can be obtained by solving the system of equations 29 and 30.

3.4.2 Handling

The tyre understeer gradient is calculated here to predict the handling performance using equation 31 even though there is no steering information in the model [7], it is a prediction how the vehicle will behave in the steering situation.

$$K_{us} = \frac{C_r * L_r - C_f * L_f}{C_f * C_r * L} \quad (31)$$

It can be noticed that there is no relationship between tyre parameters and understeer gradient. This parameter is only affected by the cornering stiffness and the center of gravity, which both depend on how much weight of loads are carried on the truck. Thus, this is one part of the future work which will involve some tyre parameters into the understeer coefficient calculation, so then a clear comparison about how understeer gradient influenced by different tyres can be seen.

3.4.3 Ride comfort

The ride comfort model is based on the simple four degree of freedom model where the various components of the truck are represented as masses which are connected by a spring and damper between each of them [7] represented by figure 12 below. Here the different representations of Z are the distances moved in vertical direction of each of the present components in the model.

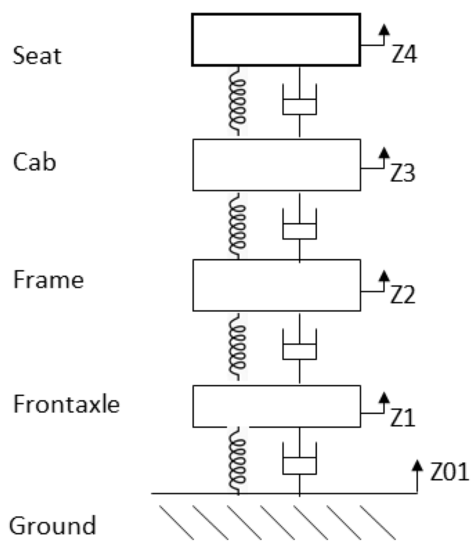

Figure 12 Four DOF Truck Model

These blocks are primarily the seat, cab, frame, front axle and then the road. The value of root mean square (RMS) acceleration of the seat will provide the information about the ride comfort of the truck. The weighted RMS value of the seat depends mainly on two main factors of the road which are the road spectra inclination and the road spectra density. It also depends on factors of the vehicle like the speed, CGW, length of the vehicle, etc.

Table 2 : Ride Comfort for diffrerent road frequencies

Road	Fi0 (Hz)	Road condition	RMS_wk Full Load Unloaded
Road 1	1e-6	Good road	0.2890 0.3030
Road 2	5e-6	Normal road	0.6462 0.6775
Road 3	10e-6	Bad road	0.9138 0.9581
Road 4	100e-6	Very bad road	2.8838 3.0299

Table 2 gives an insight into the actually effect of road roughness Fi0 (road spectral density) when the inclination is kept constant. The road spectral density is a measure of the frequency of the road that is experienced by the vehicle when it travels over a particular road. The road conditions described here as good, normal, bad and very bad are conditions as described by Volvo GTA [1] classified according to their different frequencies. The weighted RMS (RMS_wk) uses an ISO 2631 filter which weights together several time periods which have different vibrations spectra [7].

It is apparent from the table that as the roughness increase the ride comfort becomes worse and goes up to a value of 3 which is unacceptable conditions for the driver (the limit is in-between 0-1.5) [22].The two values of weighted RMS represent the two extreme conditions of the vehicle i.e., fully laden and unloaded. A point which can be noted from table 2 is also that the ride comfort remains in the acceptable range even for bad roads which covers most of the conditions the truck will be used in.

4. Model validation

This chapter intends to show the Simulink model result and validate that the models respond in a credible way to variation in three different respects:

(1) Tyre types

(2) Tyre design parameters

(3) Other parameters

Some tables and figures are attached in this chapter to illustrate which type of tyre is the 'best' under different operating conditions and how each of the tyre parameters affect the rolling resistance coefficient (RRC) and then change the fuel consumption and tyre wear. Moreover, the result of three constraints equations will be studied in this chapter.

4.1 Validation of cost response to tyre type variation

As the main object in this thesis model, the Simulink model will output fuel consumption in L/100km and tyre wear in mm/100km. Figure 13 shows the fuel consumption with increasing loads for the three different tyre types. Generally, the fuel consumption increases by roughly 10 liters from empty to fully loaded truck and also it can be noticed that tyre 1 and tyre 2 have very similar curves while tyre 3 saved about 2 liter of fuel in the same condition.

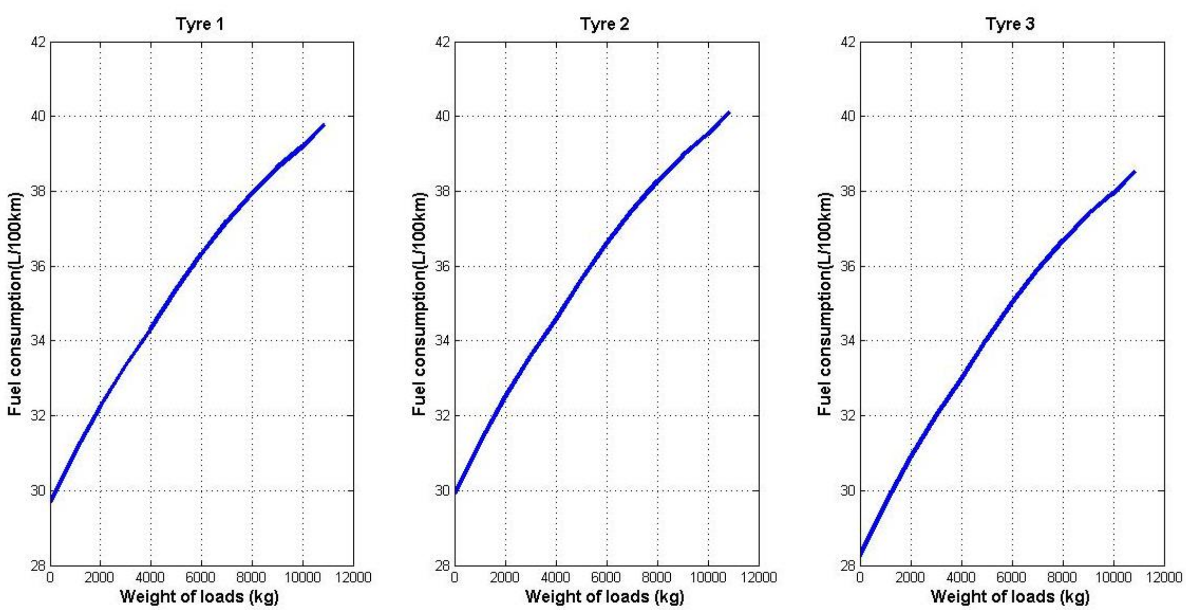

Figure 13 Fuel consumption VS tyres

As discussed earlier in section 3.2 the wear plotted in Figure 14 is the front wear. The unit used to measure tyre wear is mm of tread depth worn over a distance of 100km. The model works as it is expected to i.e. there is an increase in tyre wear as the load on the vehicle increases, the wear increase from around 0.014 mm to around 0.024 mm from completely unloaded to fully loaded. The reason why rear wear is not considered can be clearly understood table 3 below, it is close to 10 times smaller than the front wear which is unusual but the reason for the same is explained in section 3.3.

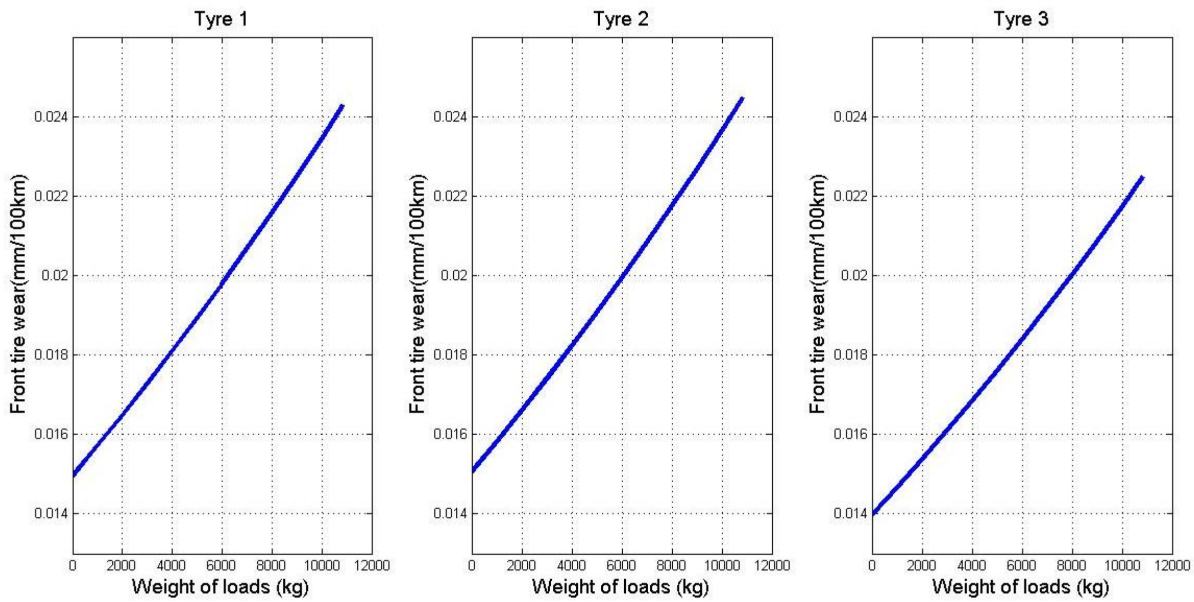

Figure 14 Front tyre wear VS tyres

Table 3 Tyre wears comparison Chart

	Empty(front)	Fully loaded (front)	Empty(rear)	Fully loaded(rear)
Tyre 1	0.01496	0.02427	0.001406	0.003689
Tyre 2	0.01507	0.02447	0.001418	0.003723
Tyre 3	0.01398	0.02248	0.001315	0.003419

4.2 Tyre types

In this section, tables will show the two outputs from the Simulink model (i.e. fuel consumption and tyre wear) and how they vary with different tyres. And the total cost can be calculated with respect to these two outputs.

Table 4 Tyre type validation 1

	Fuel consumption (L/100km)	Fuel Price(SEK)	Front wear(mm/100km)	Tyre tread depth(mm)	Tyre price(SEK)	Fuel cost(SEK)	Front wear cost(SEK)	Total cost(SEK)
Tyre 1	39.77	13.60	0.02427	15.8	2876	540.872	26.51	567.38
Tyre 2	40.11	13.60	0.02447	13.5	2095	545.496	22.78	568.28
Tyre 3	38.53	13.60	0.02248	16.2	2612	524.008	21.75	545.76

Table 4 above illustrates the differences among three model tyres under road 521. The operational case has been defined into fully liquid loaded truck (18 tons), which means the center of gravity of the loads in longitudinal direction is fixed. Total cost is calculated using equation below.

$$Total_{price} = Price_{fuel} * fc(end) + (tirewear(end)/Tire\_depth) * Price\_tire * (nog\_f + nog\_r) \quad (32)$$

Where, $Price_{fuel}$ is 13.6 SEK, $fc(end)$ and $tirewear(end)$ are the final outputs of the Simulink model after integration, representing the fuel consumption and front wear respectively. $Price\_tire$ is the price of tyre, and it varies with tyre types. $'nog\_f + nog\_r'$ is the number of tyres on this truck, 2+4=6 numerically. Hence, it can be concluded that the tyre 3 has the best performance regarding both fuel consumption and front tyre wear. While the tyre 2 consumes the highest amount of fuel and tyre wear, but the total cost is almost the same when compared with tyre 1. This is due to the price of tyre 2 is the cheapest among them.

Table 5 Tyre type validation 2

	Fuel consumption (L/100km)	Fuel Price(SEK)	Front wear(mm/100km)	Tyre tread depth(mm)	Tyre price(SEK)	Fuel cost(SEK)	Front wear cost(SEK)	Total cost(SEK)
Tyre 1	42.9	13.60	0.02405	15.8	2876	583.44	26.27	609.71
Tyre 2	43.20	13.60	0.02424	13.5	2095	587.52	22.57	610.09
Tyre 3	41.52	13.60	0.02230	16.2	2612	564.67	21.57	586.24

Once changing the road condition (road 365 in this case), similar performance can be noticed from table 5 above. The tyre 3 saves about 20 SEK per 100 km in total cost, still remain the top among three model tyres. Therefore from these two tables, it can be concluded that tyre 3 has the best performance under two road conditions.

4.3 Tyre parameters

Regarding the tyre parameters' validation, the road condition was stick to only road 521 and the tyre 1 was used as reference. Then each of the tyre parameters in the RRC surrogate model has been modified according to their valid ranges. Correspondingly, loops in overall model have been tested to see how these parameters affect the fuel consumption, tyre wear and total cost. In additional, three constraints dynamics figures are also plotted here to compare the tyre parameters' influences. In this section, a series of figures will be shown, each of them includes six different model responses, namely fuel consumption, tyre wear and total cost in the first row and startability, handling and ride comfort in the second row.

4.3.1 Weight of load

As the first tyre parameter to validate, the weight of loads is separated into two different types, see the truck definition previously. Different load distribution makes an influence on center of gravity position then change the fuel consumption and tyre wear indirectly.

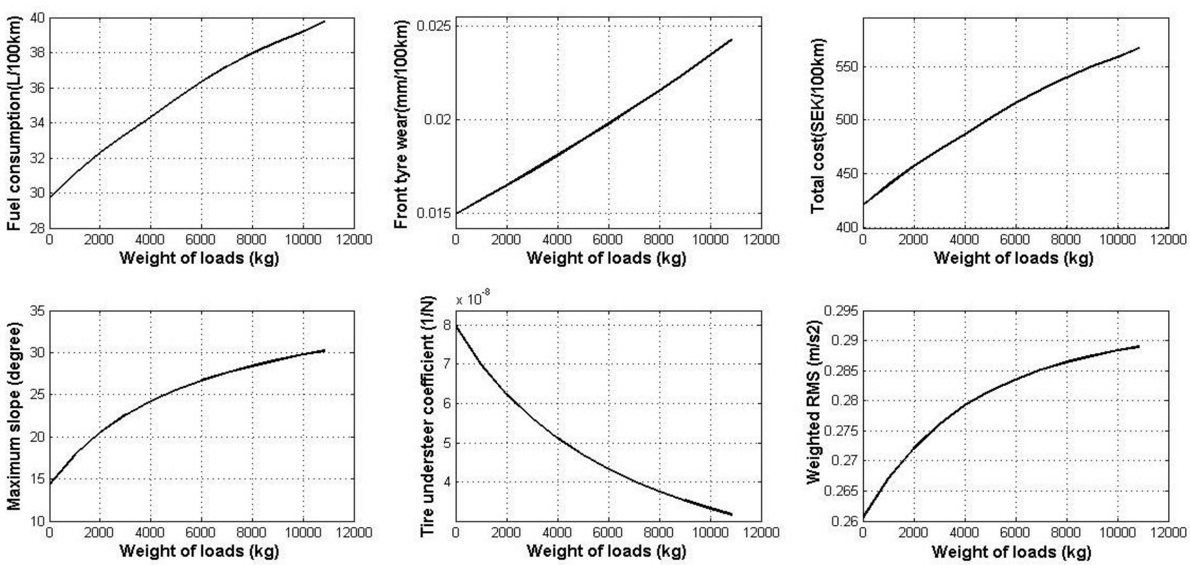

Figure 15 Weight of liquid load validation

Figure 15 shows the weight of liquid loads increases from zero to fully loaded. It can be noticed from the first row that the all these three variables ascend smoothly as expected. Specifically, the fuel consumption increases from 30 to about 40 L/100km. the similar trend occurs in front tyre wear and total cost, which go up from 0.015 to 0.024 mm/100km and from 425 to 570 SEK/100km respectively.

Regarding the constraints equations, the maximum slope increases from 15 to 30 degree with increasing load, this is due to vertical force on the driven axle increases faster than the longitudinal resistance forces and also the friction coefficient μ remains the same in this model, which means that the truck can climb higher with increasing weight of loads. Regarding the lateral dynamics, according to the equation 31, the tyre understeer coefficient drops from 8e-8 to around 3e-8. The last figure on the bottom right shows us the change in weighted RMS values when there is a change in GCW of the vehicle. This basically is the measure of ride comfort of the tuck, which can be seen is within 0.5 m/s^2 because if it increases over that then action must be taken [7]. (The road considered here is a good road from section 3.4.3)

The result is proved as valid as the fuel consumption and tyre wear will increase with increasing weight of load, but the accuracy of the result can be improved by some future studies.

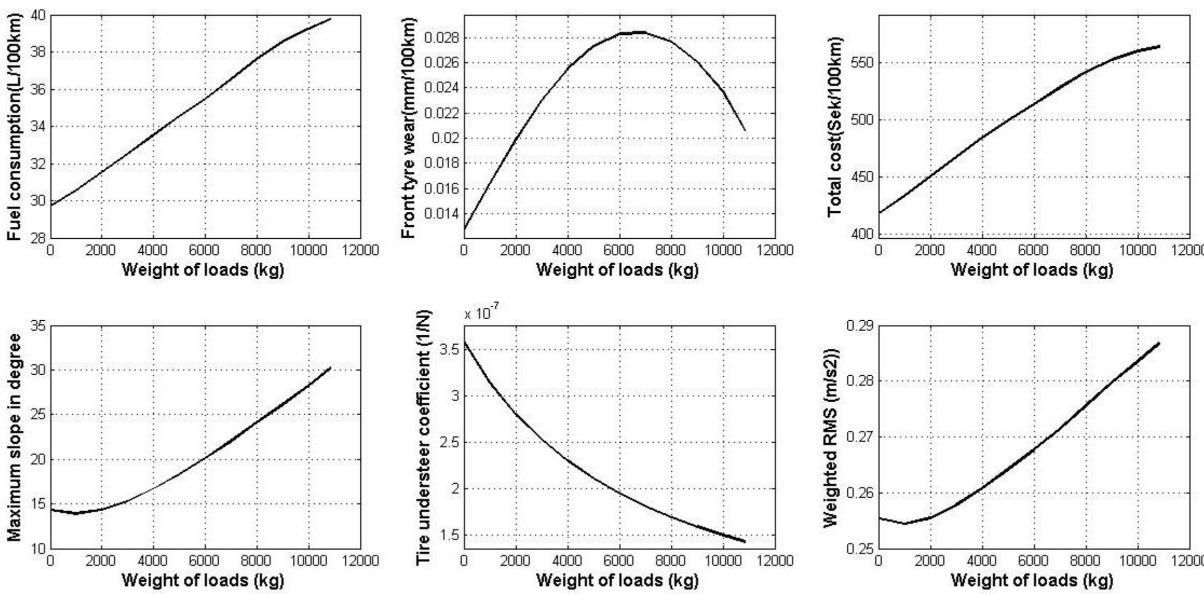

Figure 16 Weight of solid load validation

The situation is different if solid loads are carried, especially the front tyre wear. This is because the center of gravity moves backward with increasing loads, the load on front axle reaches its peak at about 60% loaded before it drops back with full loads. So the front tyre wear gets the similar trend, see figure 16 above.

About the constraints equations, the changing curves of the startability are a bit different compared with figure 15. The reason again is the center of gravity changes and the truck is rear axle driven. It is interesting to notice that the ride comfort in this condition is better at partially loaded condition than in figure 15. This is down to the fact that in this condition the loading is done from the front of the truck to the back which affects the center of gravity by a large extent and hence affecting the overall weight distribution.

4.3.2 Speed

The vehicle speed has a great influence on all these objectives. Apart from the invalid range, like 0-10 km/h, fuel consumption goes up greatly from about 22 to 52 L/100km, as well as the tyre wear from 0.0242 to 0.0244 mm/100km. this is because the truck is forced to run in a constant speed for the whole road condition without considering the proper torque and optimal driving speed. Moreover, it should be noted that the fourth subplot in figure 17 is 'meaningless' due to the truck speed does not affect the startability performance. Actually in this model, only the load has effect on these three constraints equations. Speed does not make influences on them which can be seen from the second row of figure 16, and same situations happen in tyre inflation pressure, tyre diameter, tyre width and tyre groove depth validations which will be mentioned later. This is due to the model has been simplified as well as the RRC surrogate model only has one output which used to calculate fuel consumption. Therefore, future studies are needed to introduce more tyre parameters into this model.

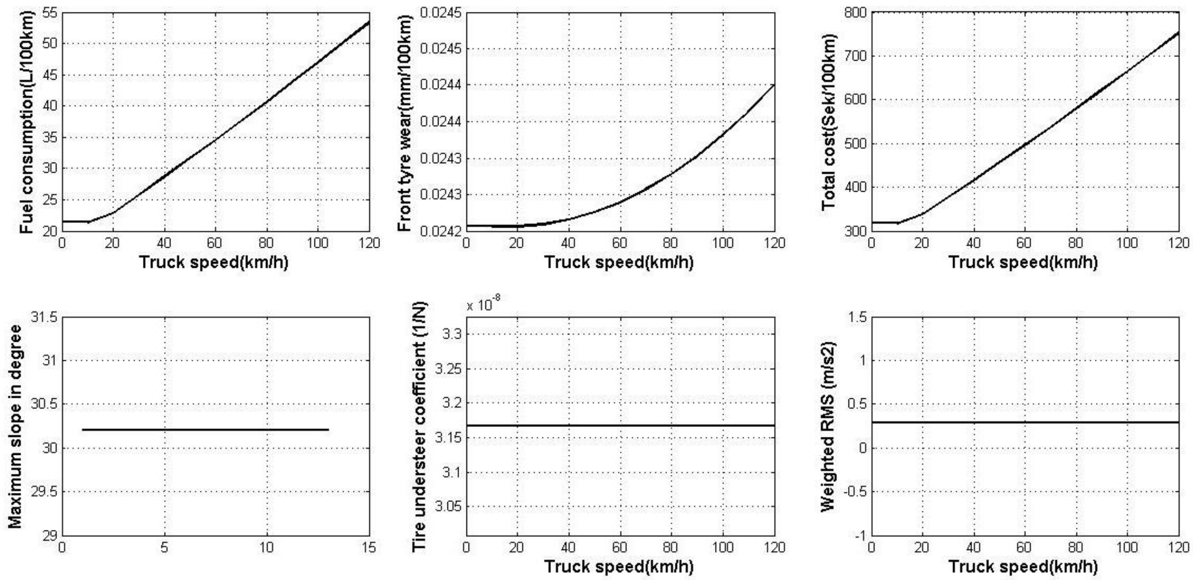

Figure 17 Speed validation

4.3.3 Tyre inflation pressure

In the RRC surrogate model, the range of the tyre inflation pressure is from 55 to 165 psi. Theoretically, the 'harder' the tyre is, the less fuel will be consumed. And it is confirmed by the figure 18 below. Front tyre wear decreases from around 0.027 to less than 0.024 mm/100km, which makes the total cost drops by 20 SEK in one hundred kilometer. Additionally, fuel consumption, tyre wear and total cost points for three tyres under their nominal inflation pressures are also showed in three different markers here. With the almost same pressure, it can be seen clearly that tyre 3 performs much better in fuel consumption, tyre wear and then total cost than the tyre 1 and 2. Here the reader might be misguided from this result to have an incorrect conclusion: The tyre inflation pressure should be set as high as possible as it will save both fuel consumption and tyre wear. This is due to so far the constraints equations are tyre-parameters independent, tyre pressure obviously affects the vehicle dynamics performance, such as ride comfort. With the continuous improvement of this model, a balance tyre pressure between minimizing the cost operation and dynamics performance will be found. Another point to be noted here is that the tyre wear decrease with the increase in inflation pressure which is contradicting figure 5, this is due to the fact that figure 5 is plotted after comparing three different models and in our case the inflation pressure is used to calculate the surface temperature of the model. Further validation is required to prove the results of figure 18 though.

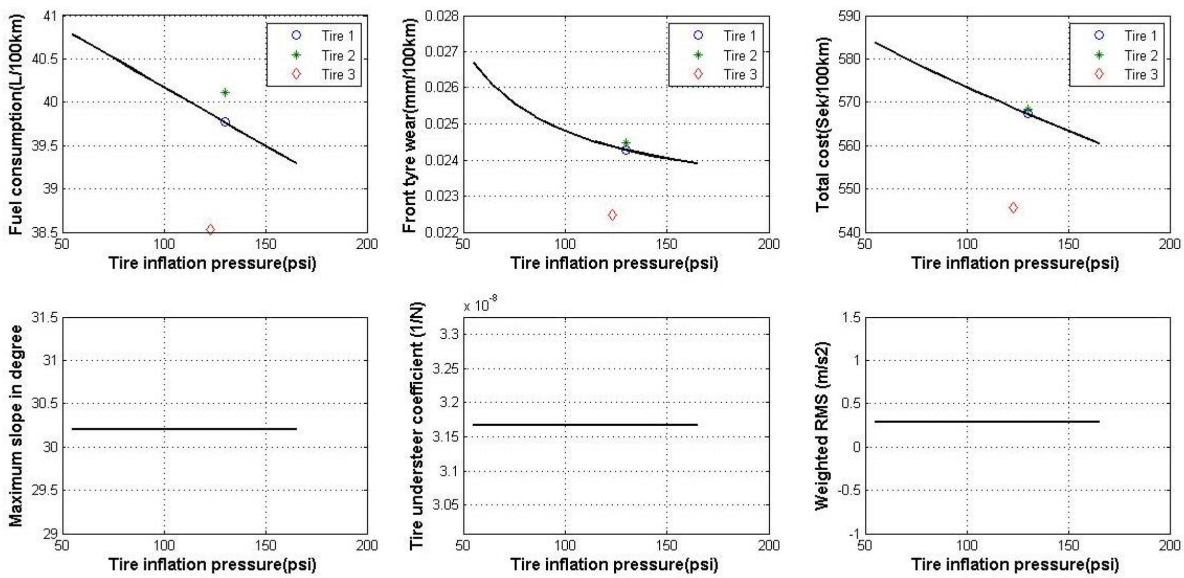

Figure 18 Tyre inflation pressure validation

4.3.4 Tyre diameter

In the RRC surrogate model, the rolling resistance coefficient is inversely proportional to the tyre diameter. This is due to the reduced tyre deflection as the diameter increases. Similarly, the fuel consumption, tyre wear and total cost fall down with the increasing tyre diameter, see the figure 19 below.

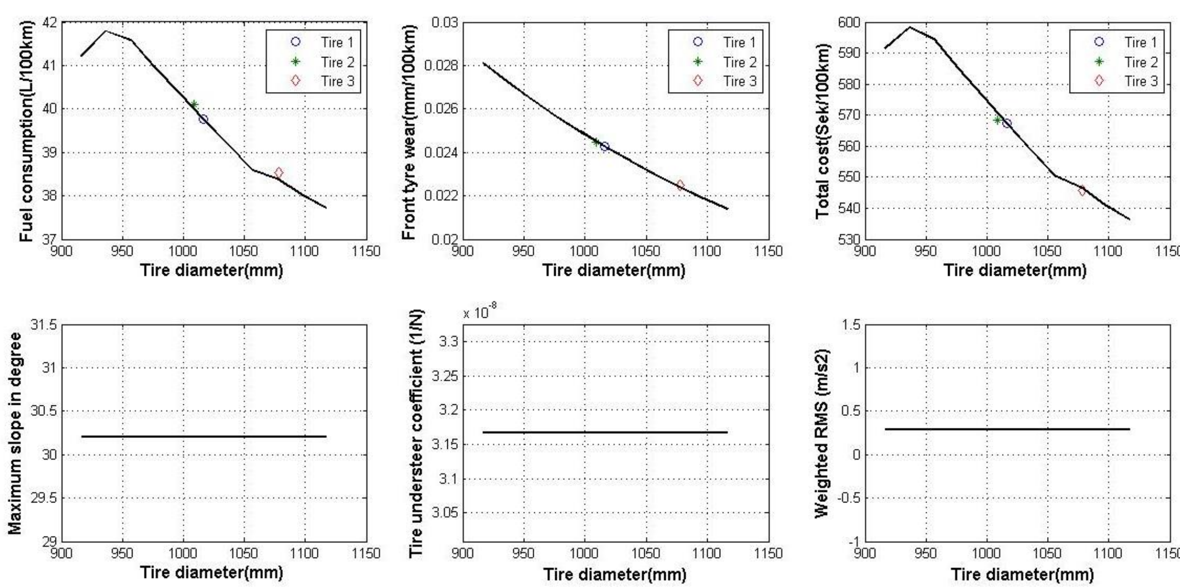

Figure 19 Tyre diameter validation

4.3.5 Tyre width

Regarding the tyre width, it has been simulated within the default range of the RRC surrogate model, i.e. from 227.66 to 455.31mm. It can be seen that the fuel consumption is not significantly increased as the tyre width increases. In contrast, the front tyre wear drops by about 0.006 mm/100km. the reason for this decreasing phenomena is due to the increased tyre width increases the tyre/road intersection and further reduce the tyre wear which is perpendicular to the direction of intersection. Also, it should be noticed that with almost the same tyre width, tyre 3 always performs better than tyre 1 and 2, see the three markers in figure 20.

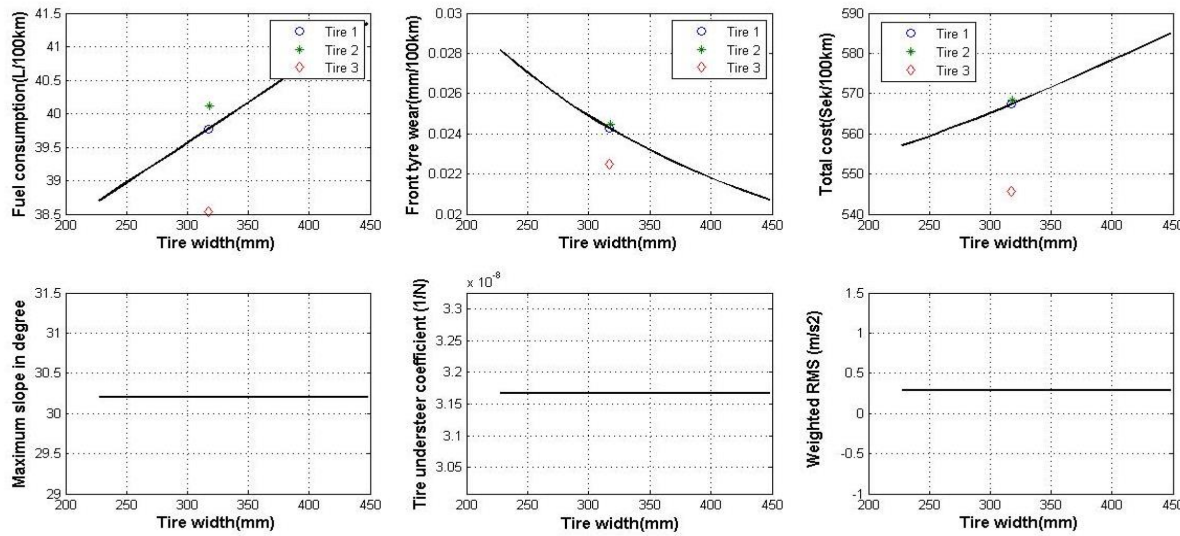

Figure 20 Tyre width validation

4.3.6 Tyre groove depth

The tyre groove depth figures show that these three main objectives have the same reducing trend as expected. This is due to the 'smooth' tyre generates tyre slip while a new tyre can produce good grip force, see figure 21 below.

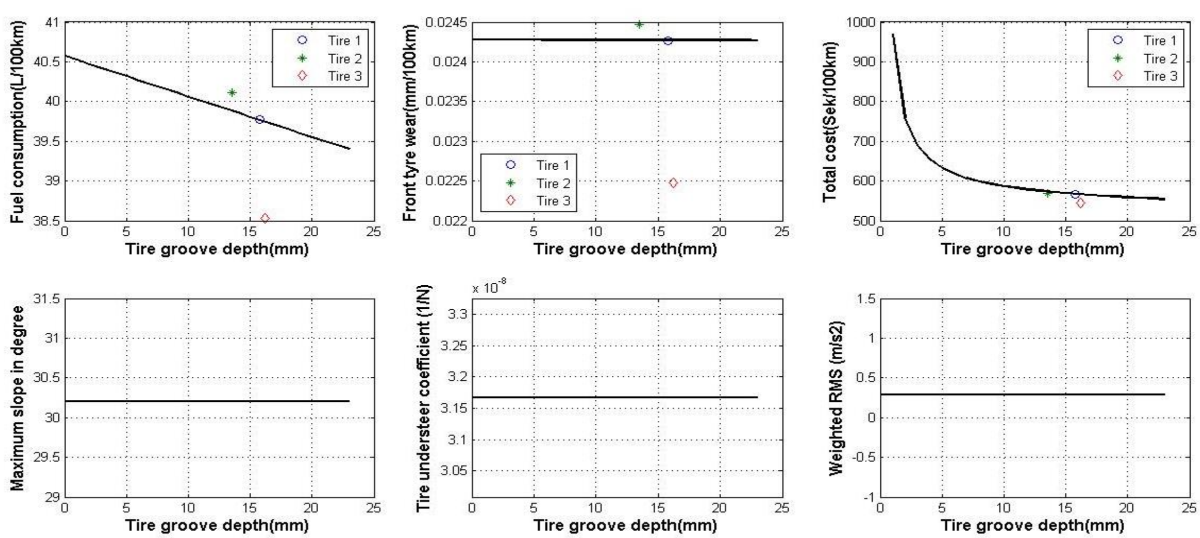

Figure 21 Tyre groove depth validation

The results of tyre parameters validations have been discussed with Volvo engineers and have been proved as valid. The constraints equations need to be improved later and then can be validated with some existing models which mentioned in the information review chapter above.

4.4 Other parameters

This section studies some other parameters' validations apart from those six tyre parameters. They mainly belong to the truck specification. The reasons for having this validation is to justify most of the important parameters in this model are changeable and also will not produce unreasonable results, and since model is created to minimize the function of $f(x,p)$, means that the $'p'$ factor which representing the vehicle and operating parameters should not be neglected. It is an essential step for the 'TyreOpt' project to expand this basic model and make more parameters involved.

4.4.1 Truck wheelbase

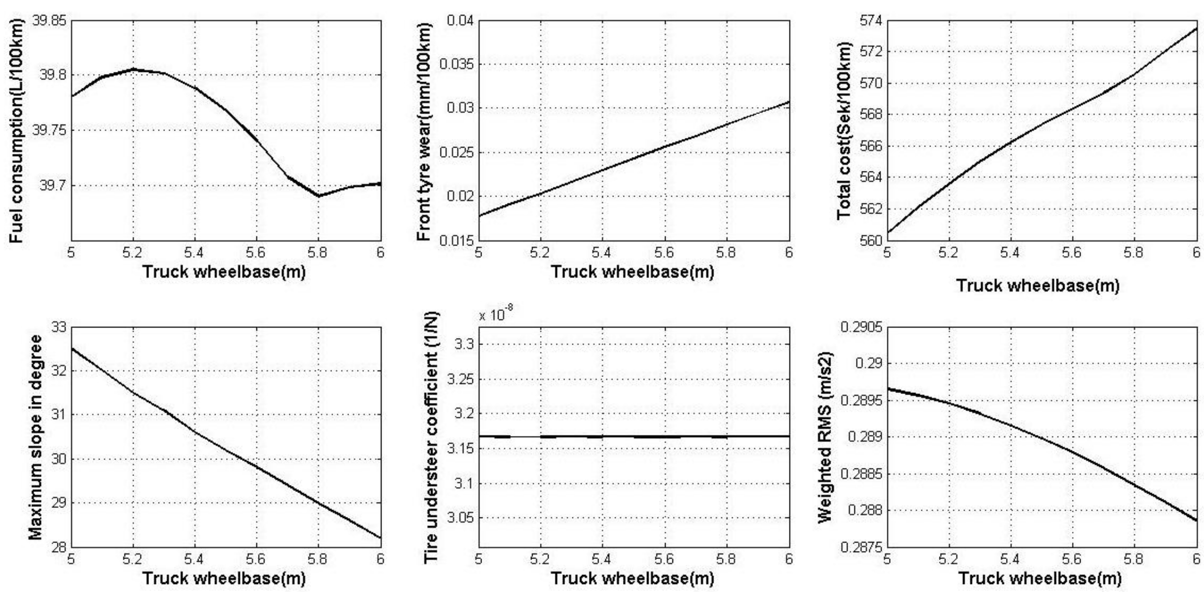

Figure 22 Wheelbase validation

Wheelbase is defined as the distance between the front and rear axles. In this thesis, the position of truck front axle is fixed and then the wheelbase can be varied by moving the rear axle. Here, figure 22 shows the wheelbase validation which changes from 5 to 6 m. When the truck is fully loaded, the distance between front axle and the center of gravity of the truck is about 5.2 m, which means the rear axle will locate exactly under the center of gravity point when the wheelbase is 5.2 m. In this case, the rear tyres will carry the maximum load and reach the fuel consumption peak, and it can be seen in figure 22.

4.4.2 Road height

In this thesis, the road data is changeable. Table 6 below shows different result when road height is alternated by multiplying coefficients. It can be seen that fuel consumption did not change a lot when the road became one fourth 'smoother', while obvious increasing of fuel consumption can be noticed when the height is multiplied by 2 and 4. The fuel consumption does not increase with corresponding coefficients, this is due to only the gravity resistance will change but some other slope-independent forces such as aerodynamics will remain the same. In addition, it is interesting to see that front tyre wear keeps constant in this validation. It is supposed to vary with coefficients because the road distance will change. But due to a simplified calculation principle is used, tyre wear will only be affected some factors, like truck speed, weight of loads, etc.

Table 6 Road height validation

	Fuel consumption (L/100km)	Front tyre wear (mm/100km)	Total cost (SEK/100km)
Height * 0.25	38.31	0.02427	547.53
Height * 1	39.77	0.02427	567.38
Height * 2	43.19	0.02427	613.89
Height * 4	52.62	0.02427	742.14

5. Conclusion & Future work

5.1 Conclusion

The aim of this thesis work is to create a complete and simplified model for 'TyreOpt' project, which identifies how different tyre parameters affect the cost of operation, while simultaneously balancing other tyre-dependent criteria such as handling properties and ride comfort [23]. Therefore, even though comparing which tyre is the best one under particular conditions has been successfully implemented from the model in this thesis, it is not the most important conclusion needed to be paid attention. The core of this thesis is to build up a framework which involves some tyre parameters and tyre-dependent models, for example RRC surrogate model, to demonstrate the relationship between each tyre parameters and total operation cost, specifically, fuel consumption and tyre wear.

Previous chapters have shown the corresponding results and model validations. Therefore, it can be concluded that the model created for this thesis is able to pick the best tyre for different operating conditions and preliminary clarify how each of the tyre basic parameters affect the fuel consumption and tyre wear. Additionally, three basic constraints models representing vehicle dynamics in different directions are also built as a supplement to the Simulink model.

5.2 Future work

However, the accuracy level of the result has not been properly discussed and completely proven. This is mainly due to the time limitation. It is also important to note that the model has been simplified to a large extent. The model simplification problem can be overcome by comparing the results to the well-established models available at Volvo GTT and relating those using mathematical methods to the model in this thesis, this will lead to a reduced analysis time. The best example of simplification can be the model of ride comfort which is a simple 4 DOF models when the accuracy can be increased by using a 35 DOF model, but this will also lead to and increased computation time. The tyre wear model is one where there has been a lot of data used from some old papers from 1960's and 1970's. Use of the aforementioned data was necessary as this data is not easily provided by tyre manufacturers now (more on tyre wear limitations can be found in section 3.3). One key improvement area can be to look into the overall wear model and see which of those assumed values can be implemented into the RRC surrogate model and the data can then be requested from the tyre manufacturers. This will help to increase the dependency of the wear model with the RRC surrogate model and also pin point the data which affects these models the most. As mentioned earlier, the constraints equations are not well related to tyre parameters; the outcome is constant results for tyre parameters' validation. Hence, one of the key areas further studies is to increase the dependence of tyre parameters in the constraints equation models which will only raise the accuracy and integrity of the whole model result.

6. Bibliography

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7. Appendices

7.1 Appendix A Models for main objectives and constraints equations

7.1.1 Fuel cost and tyre wear models

During the pre-study, several existing models which designed to calculate fuel cost were found.

- PERF, a Volvo GTT internal tool based in FORTRAN. PERF offers customers an opportunity to evaluate the truck's driveline and the criteria for low fuel consumption. It is possible for customers to define your truck specification by selecting engine, gearbox, tyres and so forth from the PERF main interface. Also, alternative specifications such as GVW, air resistance and road specification can be defined. As a result, PERF will sum up an overview of the performance of this specified truck, like economic speed range, maximum nominal speed, startability, etc.
- GSP, another Volvo GTT internal tool based in Simulink. It is a high-fidelity model. But due to the time limitation, we have not been able to find more information about it and used it to evaluate our model result.
- Simulink model from QSS toolbox, a fuel cost model based in Matlab/Simulink environment. QSS toolbox makes it possible for powertrain systems to be designed and improved quickly. It contains all the main components in powertrain driveline, including vehicle, gearbox, engine, fuel tank and battery (for hybrid powertrain). Each component has several elements for user to change easily and eventually calculates and compares the fuel consumption. This thesis work finally adopted this simulink model as a reference to design our fuel cost model.

Modeling and predicting tyre wear is a very intricate process. To the best of our knowledge, there is not tool available at Volvo GTT for tyre wear calculation. There are some approaches and tools studied during the pre-study stage which predicted tyre wear with good accuracy. Some approaches we found are from 1960's to 1970's, they are already studied in the information review and methodology chapters of this report.

7.1.2 Models for constraints equations

There are some models listed which obtained from papers or from Volvo GTT. They have been used or being used to simulate the vehicle dynamics.

Table 7 Startability reference models

No.	Model name	Tyre	Vehicle	Operating environment	Quality	Computational resource
1	VTM	MF-Tyre with relaxation, Valid up to 10 hz frequency	Point mass for the body, cab and the axle, Frame is a 2 point mass connected by a spring	Drive uphill or downhill till the truck flips over, Road disturbances up to 10hz, simple powertrain model, everything from simple brake model to a very realistic brake model	Best available model at Volvo, no traction control is included. Shouldn't consider results above 5 Hz (7/10)	Runs real time so runtime depends on the time you want to run the simulation (1 second takes one second)
2	Startability model	Magic tyre formula	one dimensional longitudinal movement and links engine torque, vehicle sped and road grade (only relatively low frequency dynamics of the vehicle itself is of interest)	Road model: α is road grade. For a constant grade road, der(α)=0, and the road grade is the derivative of the altitude, so the road altitude model becomes $\dot{z}$(s)=sin α (s) in spatial coordinates and $\dot{z}$ (t)=v(t)*sin α (t) when expressed indexed in time	Depends on the complexity of the model	Depends on the complexity of the model

Table 8 Handling reference models

No.	Model name	Tyre	Vehicle	Operating environment	Quality	Computational resource
1	VTM	MF-Tyre with relaxation , Valid up to 5 Hz frequency	Point mass for the body, cab and the axle, Frame is a 2 point mass connected by a spring	Drive uphill or downhill till the truck flips over, Road disturbances up to 10hz, simple powertrain model, everything from simple brake model to a very realistic brake model	Best available model at Volvo, no traction control is included. Shouldn't consider results above 5 hz (7/10)	Runs real time so runtime depends on the time you want to run the simulation (1 second takes one second)
2	CVM (Used for basic handling - Steady state cornering, Random steer input)	linear spring and damper, Lateral(Can be non-linear) and longitudinal also taken into consideration, slip equations also are important	FEM description of the vehicle. It can be very complex. Very detailed description of frame and cab for ride comfort. Same cab in different versions depend on the level of detail required	Road is specified as vertical profile height, can run with any speed, both measured road profiles and profiles generated in matlab, No elevation. Road disturbances of any magnitude. 0-110kmph. Curvature as a time signal, again both generated curvature	Not Always used for Handling, VTM has control systems it is preferred, but they complement each other, It's very close to reality, a fixed constant speed. It is mostly used for on-center handling analysis without large sweeping motions within the linear tyre range	Can be from a couple of minutes to a few hours. Use super elements. Takes time to reduce the model in to super elements. Once that is done it can run very fast. Can re-use the super elements need to be changed only when changes are made. Normally run with 2000 hz frequency (1 second takes 4 seconds)
3	Theoretical calculation	Tyre not included	Simple vehicle model, only wheelbase required	Static	Theoretical calculation, only depends on several parameters	Very fast direct computation

Table 9 Ride comfort reference models

No.	Model name	Tyre	Vehicle	Operating environment	Quality	Computational resource
1	VTM	Simple Spring damper vertical dynamics. Valid up to 10 hz frequency	Point mass for the body, cab and the axle, Frame is a 2 point mass connected by a spring	Drive uphill or downhill till the truck flips over, Road disturbances up to 10hz, simple powertrain model, everything from simple brake model to a very realistic brake model	Rarely used for ride comfort because of the low frequency range. Shouldn't consider results above 5 hz (2/10)	Runs real-time so runtime depends on the time you want to run the simulation (1 second takes one second) (Low cost as it only requires matlab)
2	Virtual Test Approac h	The front and rear tyres and the wheel ring are 6.00-14LT and 4.5J respectively	Three dimensiona l vehicle model consisting of all main parts	virtual test proving ground, consists of road, traffic signals and some external environment	Can minimize the cost, both designers and customers, even the users can gather and share the experience of vehicle ride comfort	A relatively long time required to build models by using ADAMS, might not be adopted in this thesis because it wasn't built in a mathematical method
3	Half Vehicle Model	point mass for front and rear tyres, spring and damping model	Half vehicle model with wheel base, body mass and pitch inertia. Anti-roll bar is included as part of the vehicle.	be described statistically by using a simple spectral density formulation, vehicle speed 20m/s, Road disturbances up to 15hz	Not mentioned, but it may be more accurate than a quarter car model because the best suspension and anti-roll bar stiffness can be obtained from half vehicle model	Not mentioned also. But it needs longer time than a quarter car model to evaluate ride comfort due to both front and rear wheels involved. Low assuming they are all simple

4	CVM	Non-linear spring and damper(Preferred) Handles loss of contact from the road properly, A rigid ring model - belt as a rigid ring up to 80 hz (Required above 50hz)	Fem description of the vehicle. It can be very complex. Very detailed description of frame and cab for ride comfort. Same cab in different versions depend on the level of detail required	Road is specified as vertical profile height, can run with any speed, both measured road profiles and profiles generated in matlab, No elevation. Road disturbances of any magnitude. 0-110kmph. Curvature as a time signal, again both generated curvature	Always used for ride comfort, Its very close to reality, no effect with vehicle speed (9/10)	Can be from a couple of minutes to a few hours. Use super elements. Takes time to reduce the model in to super elements. Once that is done it can run very fast. Can re-use the super elements need to be changed only when changes are made. Normally run with 2000 hz frequency (1 second takes 4 seconds)(Need to run for 5 min)

7.2 Appendix B Road profiles

Here shows the road profiles of road 365 and 521.

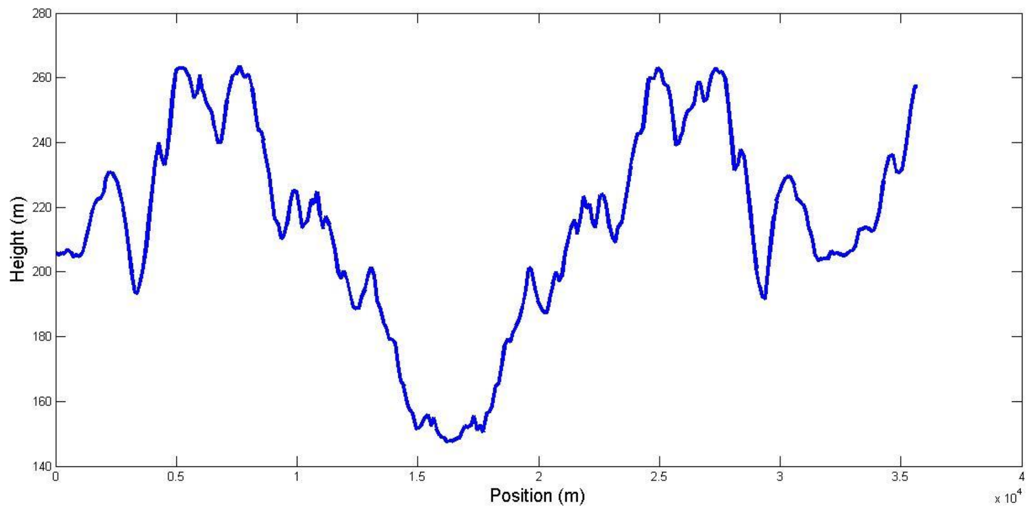

Figure 23 Road 365 height data VS longitudinal data

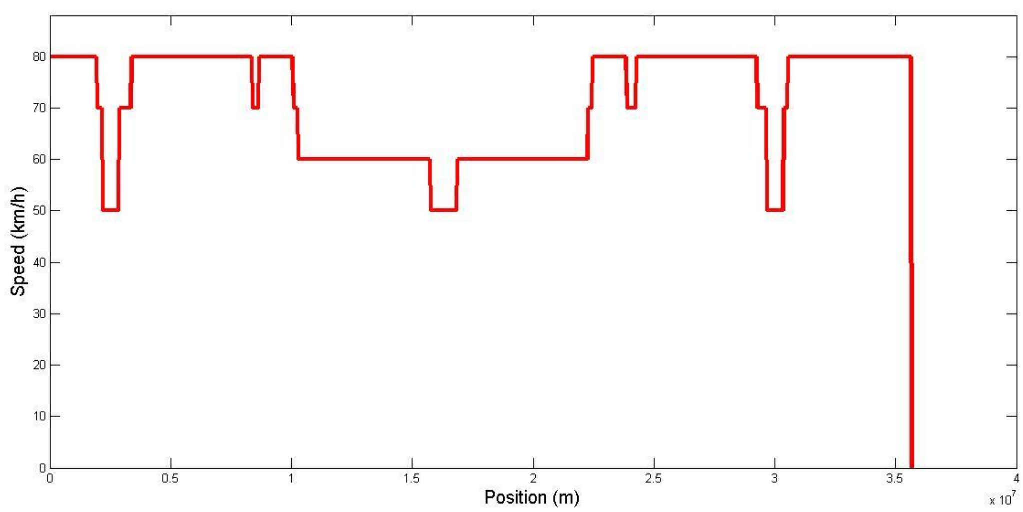

Figure 24 Road 365 speed data VS longitudinal data

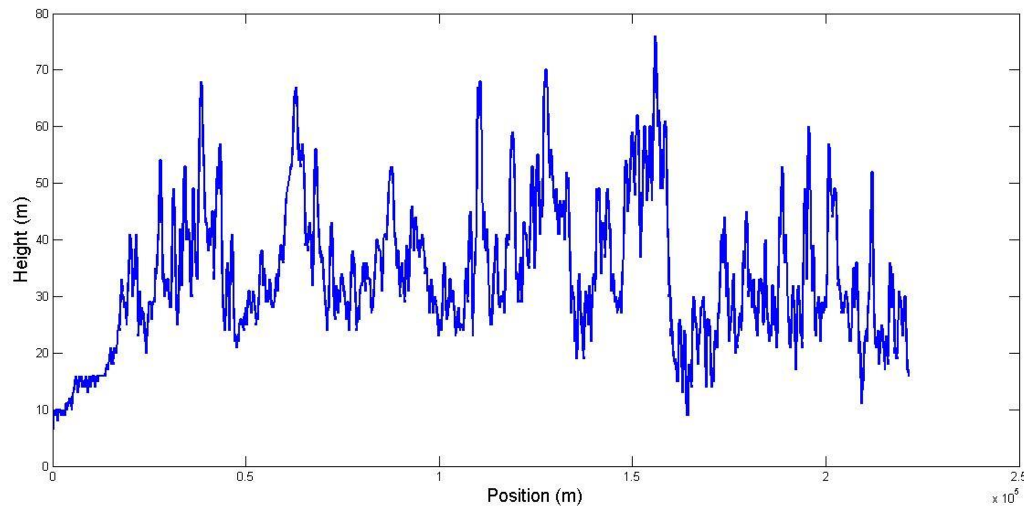

Figure 25 Road 521 height data VS longitudinal data

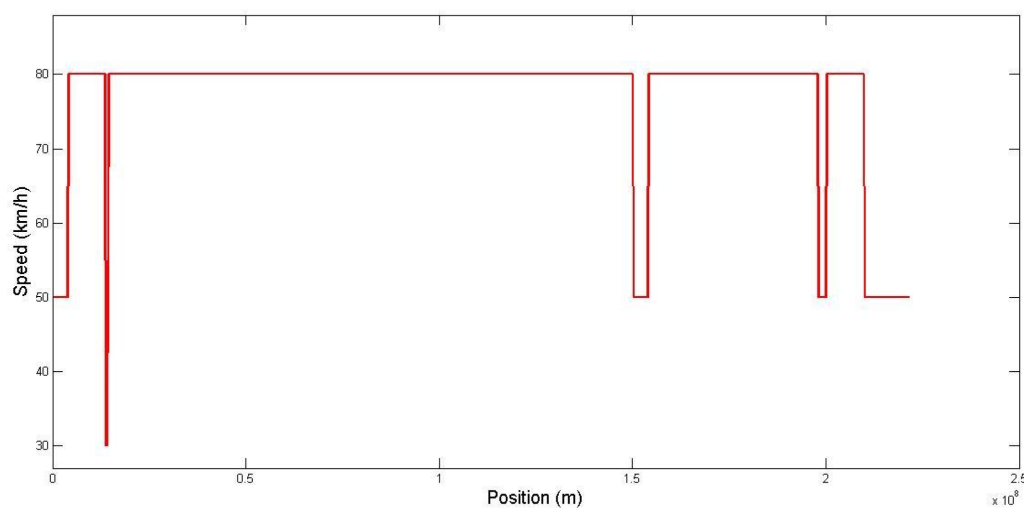

Figure 26 Road 521 speed data VS longitudinal data