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Estimation of Lateral Driving Resistance for Cornering with a Heavy-Duty Vehicle

Master's thesis in Master Programme Systems, Control and Mechatronics

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Cover: A cornering truck and semi-trailer. Photo by Gustav Lindh, 2016.

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Abstract

The driving resistance influences many aspects of a truck, such as the gear choice, the drivability and the fuel consumption. Therefore, it is possible to improve these elements by modelling the driving resistance more accurately. This master's thesis is based on two existing models for driving resistance, which are used together in order to get a final estimation. The two models have different strengths and weaknesses, which have been evaluated. A driving situation which neither of the models can handle properly has proven to be when the vehicle is cornering and braking at the same time. The pre-existing solution to this problem was to freeze the most recently known value, which is not an ideal solution. Consequently, this has been the main focus of improvement.

When cornering, an additional force, called the cornering force, acts on the truck. Therefore, in order to model the driving resistance accordingly, this force has been added to one of the pre-existing models. Since a rigid truck and a truck with a semi-trailer behaves differently when cornering, these cases have been considered separately. The modified model has then been fused together with the second pre-existing model by using a single exponential smoothing filter and by switching the filter update between the two models.

The new fused model of the driving resistance has been evaluated, both by simulation and in real life, to behave more accurately when cornering than the old model. In all other driving situations, the models' performances are equally good. It is shown that the increased driving resistance leads to a lower gear choice when cornering. This has no major effect on the drivability. Furthermore, there is no longer the need to freeze values when simultaneously braking and cornering, since the new fused model gives an accurate estimate in this driving situation.

Keywords: lateral forces, driving resistance, rolling resistance, cornering force, gear shift, drivability

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Nomenclature

Symbol	Description	Unit
α	Side slip angle	rad
β	Smoothing factor	-
γ	Articulation angle	rad
δ	Mean steering angle of vehicle	rad
δ_o	Steering angle of outer wheel	rad
δ_i	Steering angle of inner wheel	rad
θ	Road grade	rad
ρ_a	Density of air	kg/m^3
$\dot{\Psi}$	Yaw rate	rad/s
A	Frontal area of vehicle	m^2
a_x	Longitudinal acceleration	m/s^2
a_y	Lateral acceleration	m/s^2
C_a	Aerodynamic drag coefficient	-
C_r	Rolling resistance coefficient	-
C_{r1}	Another rolling resistance coefficient used together with C_{r2}	-
C_{r2}	Another rolling resistance coefficient used together with C_{r1}	-
F_{air}	Aerodynamic drag	N
F_d	Driving resistance	N
F_{d_new}	Driving resistance from new model	N
$F_{d\_traction}$	Driving resistance from traction based model	N
F_N	Normal force	N
F_{rr}	Rolling resistance	N
F_{rr_y}	Lateral rolling resistance	N
F_{slope}	Road grade force	N
F_t	Traction force	N
F_y	Lateral force	N
g	Gravitational constant	m/s^2
L	Wheelbase	m
L_a	Length between the back wheels of the truck and the turning point	m
L_t	Wheelbase of semi-trailer	m
L_w	Width between tires	m
m	Vehicle mass	kg
m_t	Semi-trailer mass	kg
r	Radius of curvature	m
r_a	Radius of curvature at turning point	m

r_t	Radius of curvature of semi-trailer	m
s_t	Smoothed average at time instance t	-
T_t	Tire temperature	$°C$
v	Vehicle velocity	m/s
v_l	Velocity at left front wheel	m/s
v_r	Velocity at right front wheel	m/s
x_t	Current observation at time instance t	-

1
Introduction

Air pollution is a large problem in today's world. Not only does it affect the temperature on the planet, but it also has an impact on the health of the population. For example, a health condition which is directly related to air pollution is asthma, see [1]. According to [2], heavy-duty vehicles are responsible for 30% of all on-road CO_2 emissions in the European Union, even though they only represent 4% of the vehicle stock. This needs to change in order to meet the European Union goal, see [3], to reduce transport greenhouse gas emissions by at least 60% from 1990 levels by 2050. One way of reducing the emissions is to reduce the fuel consumption of trucks. This can be done by always choosing an appropriate gear for the driving conditions at hand. When shifting gears there are many parameters involved, one of which is the driving resistance.

Other advantages of choosing the most suitable gear are presented in [4]. These are less wear on the clutch during heavy departures and improved drivability. Drivability is a term which according to [5] includes many aspects of a vehicle's performance, such as acceleration, engine noise, braking, automated shifting activity, and shift quality. One of the aspects of good drivability is a fast response to the driver's actions. The vehicle should, for example, brake right away when the driver applies the brake. The term is used when talking about fully or partly automotive vehicles, for instance a vehicle with automatic transmission. When talking about automated shifting activity and shift quality, drivability refers to the feeling when driving, which should be smooth and as close to what it would feel like if the driver did the gear shifting him- or herself. This means that a correct gear should always be chosen such that no engine failures occur and such that no unnecessary gear shifts are made which would make the driving uncomfortable. Good drivability also contributes to better ergonomics, both physical and physiological, and leads to a sustainable working place for the driver. A truck driver spends many hours in the truck, driving. Hence, ergonomics is important to make the ride as comfortable as possible. Physically, good drivability refers to a smooth and comfortable ride. Physiologically, if a strange gear is chosen, or if the ride is bumpy because of the gear choice, the driver might get annoyed. However, good drivability improves this.

1.1 Foundation

When modelling the resistive forces in a vehicle it is fairly common to neglect the lateral forces, since those are small compared to the longitudinal forces. In Figure

1.1.1 the longitudinal forces of a truck is presented. This thesis is based on two

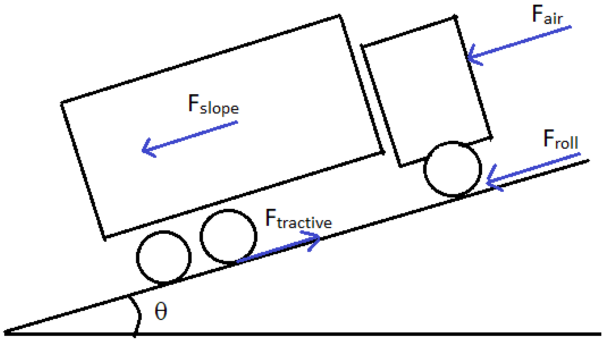

Figure 1.1.1: The longitudinal forces acting on a truck.

existing models for driving resistance, F_d, which are beneficial to use together in order to get a final estimate. The two models are preferable in different driving situations. The first model, the traction based model

$$F_d = F_t - ma_x, \tag{1.1}$$

where F_t is the traction force, m is the vehicle mass and a_x is the longitudinal acceleration, is the most straight forward one. The acceleration is differentiated from the speed, whereas the vehicle mass and traction force can be estimated. This model performs nicely on all occasions except for when the vehicle is braking. This is due to the added presence of a braking force, which is unknown. Other problems for this model are at very low velocities and when the vehicle is standing still. During these occasions it is not possible to estimate the traction force correctly since no velocity signal is available. To complement this model, and in order to have a working model for the driving resistance when braking, there exists a second model, the slope-, air- and rolling resistance based (SAR) model

$$F_d = F_{slope} + F_{air} + F_{rr}. \tag{1.2}$$

This model is based on the force induced by the road grade

$$F_{slope} = mg\sin(\theta) \tag{1.3}$$

where the road grade θ can be estimated with a longitudinal accelerometer, the aerodynamic drag

$$F_{air} = \frac{1}{2}\rho_a v^2 C_a A, \tag{1.4}$$

where ρ_a is the air density, v is the vehicle velocity, C_a is a coefficient and A is the front area of the truck, and the rolling resistance

$$F_{rr} = C_r F_N = C_r mg\cos(\theta) \approx C_r mg. \tag{1.5}$$

C_r is an estimated coefficient based on the current road surface conditions. The road grade θ will rarely be larger than corresponding 10% when driving on a public road in Sweden, see [6], and thus $\cos(\theta) \approx 1$. This is not always the driving situation at hand though, since there exist trucks which both drive and work off-road. However, even a slope of 20% will give a value that may be approximated to 1, which is considered to be good enough in this thesis.

The weakness of the SAR model is the longitudinal accelerometer, which does not always give an accurate measurement in certain driving situations. This is due to external longitudinal movements of the vehicle, wobbles, and additional lateral accelerations influencing the longitudinal result. The most critical driving situations for the longitudinal accelerometer are right before and after a stop, during gear switch when the road grade is more than 6%, and when the vehicle is cornering with a speed of 5 m/s or more, according to [7].

The best strategy in order to get a good approximation of the driving resistance when using these two models together, is to choose which model to trust depending on the driving situation. When braking, the SAR model should be used, with the exception of when the vehicle is cornering. In all situations when not braking, the traction based model should be used. In driving situations in which both models are to be trusted they could be weighed together, and when neither of them work it is profitable to set the driving resistance to the most recently known value, i.e momentarily freeze it. A driving situation in which neither of the models work is when the vehicle is braking and cornering at the same time. In this situation, the traction based model is not to be trusted, due to the unknown braking force, and the SAR model does not give an accurate result, due to the errors of the longitudinal accelerometer when cornering. Another situation in which neither model function, is when the vehicle is driving with very low velocity. The weaknesses and strengths of each model are presented in Table 1.1.

Traction based model, (1.1)		**SAR model, (1.2)**	
+	Curvature	+	Brake
−	Brake	+	Standstill
−	Low velocity	−	Curvature
−	Standstill	−	Low velocity

Table 1.1: The strengths and weaknesses of the traction based model and the SAR model.

The aim of this thesis is to investigate the possibility of improving the accuracy of the driving resistance estimate when using these two driving resistance models in combination. A better estimate for cornering when braking has been the main objective, since neither of the mentioned models perform nicely in this driving situation. Furthermore, the driving resistance when driving with low velocity is also investigated.

1.2 Contribution

The contribution of this thesis is an improved estimate of the driving resistance, in such a way that there is always a reliable estimate for every time instant. By these means, no freezing of values are necessary which improves the accuracy of the estimate. In addition to the scope of the previously existing estimate of the driving resistance, the new estimate can handle driving situations which require both cornering and braking. Since a rigid truck and a truck with a semi-trailer behave differently when cornering, these cases have been modelled separately. A truck with a trailer gives yet another case when cornering, but this case has not been considered in this thesis. The improved models are based on data which is possible to retrieve from a Scania vehicle.

1.3 Limitations

The main limitation to this project has been that no measured values of the signals estimated during the thesis have been available for comparison. Thus, every estimate is based on what is assumed to be correct, rather than what is actually correct.

Furthermore, when evaluating the new model estimate there have been some restrictions. The test used for analysing the drivability is very subjective, since only a few drivers have been involved in the evaluation. In order to get an objective judgement, more drivers should be allowed to test drive a vehicle with the new model and state their opinions.

Another restriction when evaluating the new model is related to the environmental sustainability. A good way of evaluating this is by analysing the fuel consumption. However, there is no available sensor for measuring this. Moreover, there is at the moment no way to calculate the fuel consumption from the available signals mentioned in Table 1.2. A model for simulation and calculation of the fuel consumption would take time to develop and since the fuel consumption has not been the main focus of this thesis, this has not been done.

1.3.1 Available Sensors and Signals

The available sensors and signals in a Scania vehicle which are of interest are presented in Table 1.2. Since the focus of this thesis has been to make a better estimation of the driving resistance based on sensors and signals already available, no sensors have been added to the truck during the work. Thus, this has been a limitation. Another limitation has been that the estimated signals listed in Table

1.2 are considered to be correctly estimated, and are therefore considered as known parameters in this thesis.

Traction	Engine torque, Status of clutch and gears
Transmission	Axle rotations, Gear ratios
IMU	Acceleration (X, Y, Z), Yaw rate
Wheels	Velocities of the wheels
Other	Air pressure, Temperature, GPS, Road surface temperature, Status of brake, Weight of semi-trailer/trailer
Estimated	Vehicle mass, Road grade, Radius of curvature

Table 1.2: Available sensors and signals in a truck.

1.4 Thesis Structure

The thesis is organised as follows. In Chapter 2, some background and theory needed in order to understand the concept of driving resistance are presented. Chapter 2 also contains the model equations for the driving resistance of a rigid truck and of a truck with a semi-trailer. In Chapter 3, these model equations are validated by simulation. In Chapter 4, a filter which smoothens and combines two models into a final estimation of driving resistance is developed. This chapter also contains the evaluation of the final models, done both with simulation and with real life testing. In Chapter 5, the improvements of the new models are discussed and lastly, Chapter 6 contains conclusions and some potential future work.

2

Modelling

In this chapter the SAR model will be remodelled. First, some theory needed in order to do the modelling are discussed, and then the models for a rigid truck and a truck with a semi-trailer are presented.

2.1 Coordinate System of a Vehicle

The coordinate system used in this thesis is presented in Figure 2.1.1. The x-axis is defined as longitudinal, the y-axis as lateral and the z-axis as vertical.

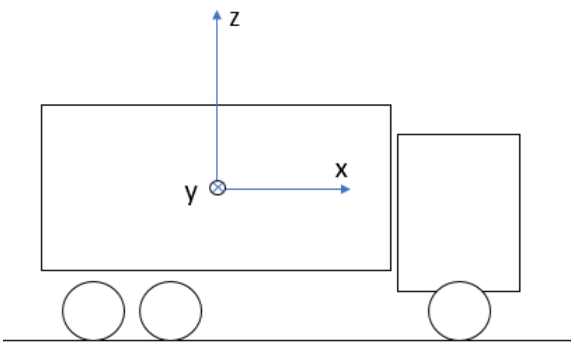

Figure 2.1.1: The vehicle's coordinate system.

2.2 Driving Resistance

Driving resistance is the sum of all external resistances currently acting on the vehicle, and thus depends on many things, for example aerodynamic drag, road grade, vehicle mass, rolling resistance, friction, gravity and the frontal area of truck, see [8]. When estimating the driving resistance the vehicle mass and the road grade are the two most important parameters, according to [9]. In [10] it is stated that the vehicle mass of heavy duty trucks can differ up to 400% depending on if the truck is fully loaded or empty. The vehicle mass does not change much once the truck starts driving. However, this can not be said about the road grade which varies continuously over time.

There exists different models for estimating the driving resistance. The models based on only longitudinal dynamics are quite straightforward, whereas more complex methods are not as established according to [11]. Since the exact parameters of a vehicle are rarely known, due to estimations, the accuracy of the system models is not perfect. The complexity of the model grows when looking at more than the longitudinal dynamics, taking more parameters into account. Thus, these systems are more sensitive to errors from estimated values. Based on these arguments, the traction based model (1.1) has not been further investigated since it is assumed to be straight forward. The SAR model (1.2), however, can be improved. Since both the road grade and the vehicle mass are considered to be known in this thesis, and since the aerodynamic drag equation is quite established, the rolling resistance remains as the parameter to investigate. There is, of course, the possibility to model the lateral aerodynamic drag when cornering, but this additional force is considered to be insignificant in relation to the other forces considered when estimating the driving resistance.

2.3 Rolling Resistance

When estimating the rolling resistance of a vehicle it is common to assume that the vehicle is driving straight forward on a dry surface. This results in the total rolling resistance being equal to the tire rolling resistance. However, in [12] there are other factors mentioned that should be taken into consideration such as the road resistance, the braking resistance and the resistance from the tire slip angle. These factors may be estimated in the rolling resistance coefficient. As with the driving resistance, there exist different models for the rolling resistance. One example is the model mentioned in equation (1.5), but there also exist models which take the velocity of the vehicle into consideration such as

$$F_{rr} = m(C_{r1} + C_{r2}v). \tag{2.1}$$

This model is presented in [13] and has two coefficients. In [14] however, the behaviour of this model has been estimated to be similar to model (1.5).

Another element which affects the rolling resistance coefficient is the tire temperature, T_t. Hence, a model which is dependent on both the tire temperature and the velocity exists, see [15],

$$C_r = f(v, T_t). \tag{2.2}$$

When the tire temperature increases, the rolling resistance coefficient decreases. The coefficient does not only depend on the tire temperature though, but also on other factors like how worn the tire is, the type of tire and the tire head. However, since a regular truck does not have a sensor for measuring the tire temperature, and since an estimation would be too uncertain to use according to [14], this has not been taken into consideration.

Yet another option is a rolling resistance model dependent on the resistance from the tire slip angle. The resistance from the tire slip angle is something which is related to how large the slip angle is. The slip angle, in turn, is dependent on the steering angle of the wheels. Generally, the heading direction of the wheels coincides with the travelling direction of the vehicle when the vehicle is driving straight forward. When cornering, however, the vehicle will have a lateral motion in addition to the longitudinal one. This means the heading direction of the wheels may not necessary be in line with the travelling direction anymore. Thus, a lateral component, the cornering force

$$F_{rr\_y} = F_y sin(\alpha) = ma_y sin(\alpha), \tag{2.3}$$

could be added when cornering in order to get a more accurate estimate of the rolling resistance. Here α is the side slip angle. By adding the cornering force to the rolling resistance, the accuracy of the SAR model when cornering might be improved. As mentioned, cornering is one of the weaknesses of this model. Therefore, this addition has been further investigated.

2.4 Ackermann Steering Geometry

Ackermann steering geometry is a solution to the right and left wheels turning with different radii when a vehicle is cornering. A right turn does, for example, require the right wheel to turn more than the left wheel. The different angles of the inner and outer wheels when cornering are illustrated in Figure 2.4.1.

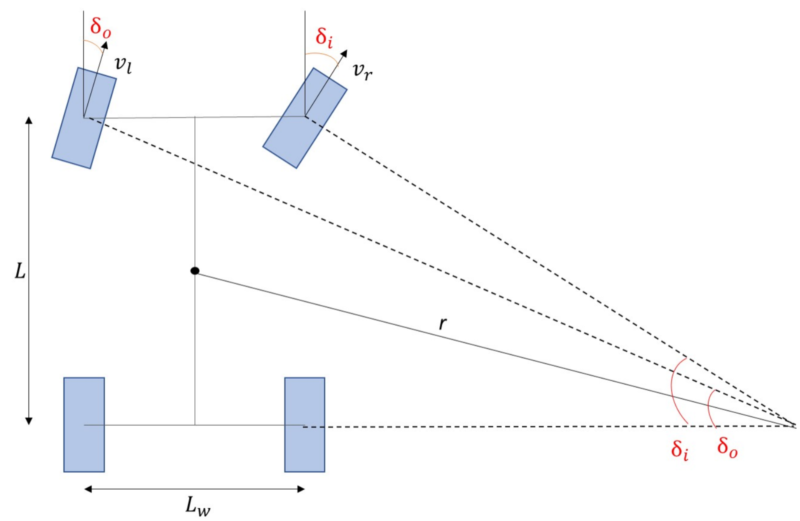

Figure 2.4.1: The wheels of a vehicle when cornering. Since it is a right turn, the right wheel is the inner one and the left is the outer.

As a consequence of the wheels turning with different radii, they also have different velocities v and different steering angles δ. The velocities will always be related to a certain wheel, left or right, but since it differs which wheel is the inner and which is the outer one, depending on the direction of the turn, it varies which steering angle is related to which wheel. The outer steering angle can be calculated as

$$\delta_o \approx \frac{L}{r + L_w/2}, \tag{2.4}$$

and the inner steering angle as

$$\delta_i \approx \frac{L}{r - L_w/2}. \tag{2.5}$$

Here, L is the wheelbase, L_w is the width between the tires, and r is the radius of curvature. By combining the inner and outer steering angle, it is possible to define the average steering angle at the front wheels as

$$\delta = \frac{L}{r} \tag{2.6}$$

which is called the Ackermann Angle, see [16]. However, this is only true for low speeds and for small steering angles. At higher speeds, the equation differs due to the presence of lateral acceleration, which acts on the tires as a lateral force and results in a side slip angle at each wheel. The side slip angle and its relation to the steering angle is presented in Figure 2.4.2.

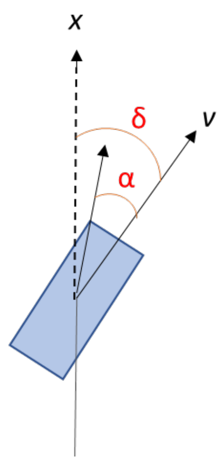

Figure 2.4.2: The steering angle and the side slip angle when making a right turn.

According to [12], the high speed steering angle when cornering is calculated as

$$\delta = \frac{L}{r} + \alpha, \tag{2.7}$$

where α is the average side slip angle of the vehicle. The side slip angle is defined as the angle between the true heading of the vehicle and the direction of steering at the wheels. This means that this angle is more correct to use when estimating the lateral position of the vehicle than the steering angle, which is defined as the angle between the front of the vehicle and the steering wheels direction. In order to solve α from (2.7), another expression for the steering angle is needed.

2.5 Steering Angle

The steering angle can be estimated with the relation

$$\delta = \frac{\dot{\Psi} L}{v}, \tag{2.8}$$

where $\dot{\Psi}$ is the yaw rate, v the velocity of the vehicle and L the wheelbase. The yaw rate is estimated by the change of velocity on the right respective the left wheel, i.e by how much the vehicle is turning. The equation of the yaw rate is

$$\dot{\Psi} = \frac{2}{L_w}(v_l - v_r), \tag{2.9}$$

where L_w is the width between the tires, v_l is the velocity of the left wheel and v_r is the velocity of the right wheel. The velocity of the vehicle is estimated as the mean of the wheel velocities,

$$v = \frac{1}{2}(v_l + v_r). \tag{2.10}$$

By combining (2.8), (2.9) and (2.10), the steering angle equation can be written as

$$\delta = \frac{4L(v_l - v_r)}{L_w(v_l + v_r)}. \tag{2.11}$$

By using (2.11) together with (2.7), it is hereby possible to get a complete expression for the side slip angle α.

2.6 Model of Truck

Since the model for a truck is an improvement of model (1.2), the SAR model, it is based on the road grade, the air resistance and the rolling resistance. The model equation is

$$F_d = F_{slope} + F_{air} + F_{rr}, \tag{2.12}$$

where the air resistance is kept as

$$F_{air} = \frac{1}{2}\rho_a v^2 C_a A. \tag{2.13}$$

The total rolling resistance with the added cornering force is modelled as

$$F_{rr} = C_r mg + m a_y \sin(\alpha), \tag{2.14}$$

where the side slip angle is estimated by combining (2.7) and (2.11) to

$$\alpha = \left| \frac{4L(v_l - v_r)}{L_w(v_l + v_r)} \right| - \frac{L}{r}. \tag{2.15}$$

The curve radius r is set to be a positive value greater than zero. When the vehicle is driving straight forward, the curve radius should be very large and in theory go towards infinity. Since a positive force should be applied during both a right turn and a left turn, the steering angle is set to be the absolute value of itself. Thus, the side slip angle will also always be positive, as well as the lateral force. This is reasonable, since the driving resistance is expected to increase when cornering, regardless of turning right or left. When the vehicle is not cornering, i.e. when it is driving straight forward, there will be no side slip angle acting on the wheels. Hence, the lateral rolling resistance will be nonexistent. This means that when the vehicle is driving straight ahead, only the longitudinal forces will be taken into consideration. Thus, the new model is identical to the old one in this precise driving situation.

The old F_{slope} is based on an estimation of the road grade made on measurements from a longitudinal accelerometer. Due to the problems of the longitudinal accelerometer mentioned in Section 1.1, it is desirable to make an adjustment. It is possible to retrieve the road grade of a public road from map data. This information is quite easy to use and therefore, an alteration of the road grade is made even though it is stated that the road grade is considered to be known beforehand. The map data is presented in % and the designed model uses radians. Hence, a conversion is needed:

$$\theta = \arctan\left(\frac{slope}{100}\right). \tag{2.16}$$

With this information, the force from the road grade can be calculated with the same equation as before,

$$F_{slope} = mg\sin(\theta), \tag{2.17}$$

which is used in all instances when there is map data available. When not, the road grade estimation from the longitudinal accelerometer is used. As a consequence of this alteration, the slope estimate becomes nearly, or at least partly, independent of the longitudinal accelerometer. Hence, the errors related to the accelerometer, such as not giving a correct estimate during low velocities, are not an issue for this new model. This makes the model much more reliable in driving situations where the accelerometer is not to be trusted.

In this new model of the driving resistance the wheelbase L, the length between the tires L_w, and the frontal area of the vehicle A are known parameters. The lateral acceleration a_y, the slope θ, the vehicle velocity v, as well as the right and left wheel velocities, v_r and v_l, are measured. The vehicle mass m, the rolling resistance and the air drag coefficients, C_r and C_a, the curvature radius r and the air density ρ_a are estimated. The gravitational constant g is set to be 9.81. Since this model is not dependent on any longitudinal accelerometer, there are no obvious driving situations in which this model might not perform as expected. The lateral acceleration is measured with an accelerometer but since lateral wobbles are not as common or large as longitudinal, and since it is easy to extinguish the lateral from the longitudinal acceleration, this signal is assumed to be sufficiently accurate at most times. Of course, there exist lateral wobbles due to lateral road grade, but these are considered to be small and not influence the accuracy of the lateral accelerometer too much.

2.7 Semi-Trailer Geometry

When adding a semi-trailer to a truck, the longitudinal and lateral forces are influenced. The longitudinal changes are not much of a concern, since the driving resistance is dependent on the vehicle mass which is increased accordingly when adding the semi-trailer. Therefore, the longitudinal model will remain valid. The additional lateral forces, however, needs more contemplation. When cornering with a semi-trailer, the truck and the semi-trailer will not turn with the same angle. Thus, the lateral force of the truck and the semi-trailer can not be estimated by the same equation. Hence, the angle between the truck and the semi-trailer, the articulation angle γ, needs to be known. Visualisations of the articulation angle are shown in Figure 2.7.1 and Figure 2.7.2.

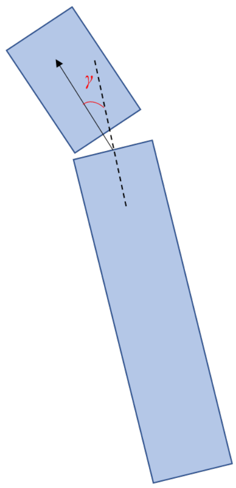

Figure 2.7.1: A visualisation of the articulation angle.

By looking at the trigonometry of Figure 2.7.2, it can be seen that the articulation angle can be expressed in terms of the curve radius of the semi-trailer, r_t, as

$$\gamma = \arcsin\left(\frac{L_t}{r_t}\right) + \arctan\left(\frac{L_a}{r_a}\right) \tag{2.18}$$

where

$$r_a = \sqrt{(r_t^2 - L_a^2)}. \tag{2.19}$$

Since the curve radius of the semi-trailer is not something which currently is measured in the trucks, this parameter needs to be estimated or calculated. When a truck with a semi-trailer is cornering the curve radii of the two components are assumed to be very similar in size, since it is the same turn they are making, but with a difference in time, due to the semi-trailer following behind the truck. Hence, the curve radius of the semi-trailer is modelled as a delayed version of the curve radius of the truck, r. The delay is inversely proportional to the speed of the vehicle, i.e. at high velocities the delay is smaller, and at low velocities the delay is larger.

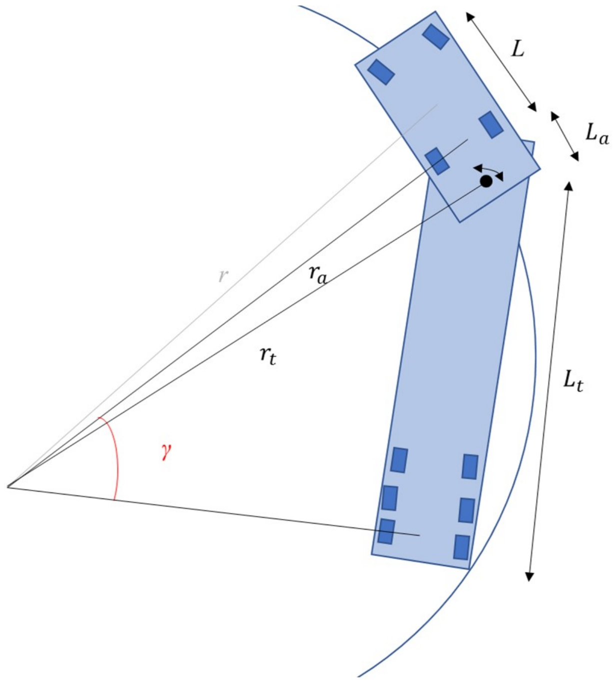

Figure 2.7.2: A truck with a semi-trailer when cornering.

2.8 Model of Truck with Semi-Trailer

As with the model for a rigid truck, this model is based on the road grade, the air resistance and the rolling resistance,

$$F_d = F_{slope} + F_{air} + F_{rr}. \tag{2.20}$$

The rolling resistance of a truck with a semi-trailer depends on both the lateral force acting on the truck and the lateral force acting on the semi-trailer. Therefore, the lateral force presented in Section 2.6 is used in this model too, as the lateral force acting on the truck. The lateral force acting on the semi-trailer is similar to the force acting on the truck, but calculated with the values of the semi-trailer instead of those of the truck. The lateral forces are added to the longitudinal rolling resistance as

$$F_{rr} = C_r g(m + m_t) + m a_y \sin(\alpha) + m_t a_y \sin(\gamma), \tag{2.21}$$

where m_t is the mass of the semi-trailer, a_y the lateral acceleration and γ is the articulation angle. The articulation angle is estimated by combining (2.18) and (2.19) to

$$\gamma = \arcsin\left(\frac{L_t}{r_t}\right) + \arctan\left(\frac{L_a}{\sqrt{(r_t^2 - L_a^2)}}\right). \tag{2.22}$$

The air resistance is kept as

$$F_{air} = \frac{1}{2}\rho_a v^2 C_a A, \tag{2.23}$$

and the road grade is calculated in the same way presented in (2.17),

$$F_{slope} = (m + m_t) g \sin(\theta). \tag{2.24}$$

The known parameters of this model are the wheelbase of the semi-trailer L_t, the distance from the turning point to the rear wheels of the truck L_a, and the frontal area of the vehicle A. The lateral acceleration a_y, the road grade θ and the vehicle velocity v are measured. The vehicle mass m, the mass of the semi-trailer m_t, the rolling resistance coefficient C_r, the air drag coefficient C_a, the curvature radius of the semi-trailer r_t and the air density ρ_a are estimated. The gravitational constant is kept as 9.81.

3

Model Validation

In this chapter the driving resistance models for a truck and a truck with a semi-trailer are validated. When evaluating the models, the traction based model (1.1) and the SAR model (1.2) are used as benchmarks. Three different driving cases are evaluated. The cases are when cornering, braking and during change of slope, i.e the driving situations in which the new model should perform better than the benchmarks.

3.1 Model of Truck

The data used for validating the model of a truck was collected with a Scania G500 6x4 truck, in regular traffic situations in Södertälje. The specifications of the parameters of the truck are presented in Table 3.1.

model	L[m]	L_w[m]	m[kg]
Scania G500 6x4	5	2.2	26000

Table 3.1: Specifications of the truck used for evaluation.

3.1.1 Cornering

When cornering, the impact of the lateral force on the vehicle increase. Consequently, the driving resistance is also expected to increase. In Figure 3.1.1 the truck is entering a roundabout at time instance 10 seconds, where the force from the new model increases. The old models increase as well, but not as much and as quick as requested. By only looking at the old models, a turn would be hard to distinguish. Hence, the new model can estimate the driving resistance better in turns than the old models.

3.1.2 Braking

Due to an unknown braking force, the traction based model is not to be trusted when braking. This results in the traction based model getting peaks and not following the anticipated path when the vehicle is braking. The SAR model and the new model are expected to give almost the same result when braking and to keep a smooth path. Figure 3.1.2 shows a clear example of the behaviour of the traction based model when applying brake. The new model keeps the right course when braking,

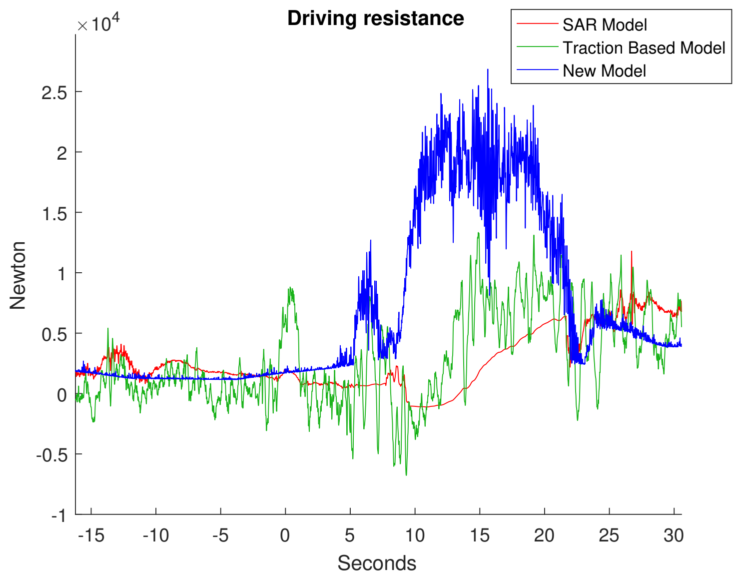

Figure 3.1.1: The driving resistance of a truck driving in a roundabout.

similar to the SAR model. Therefore, it can be concluded that the new model is to be trusted more than the traction based model when the vehicle is braking.

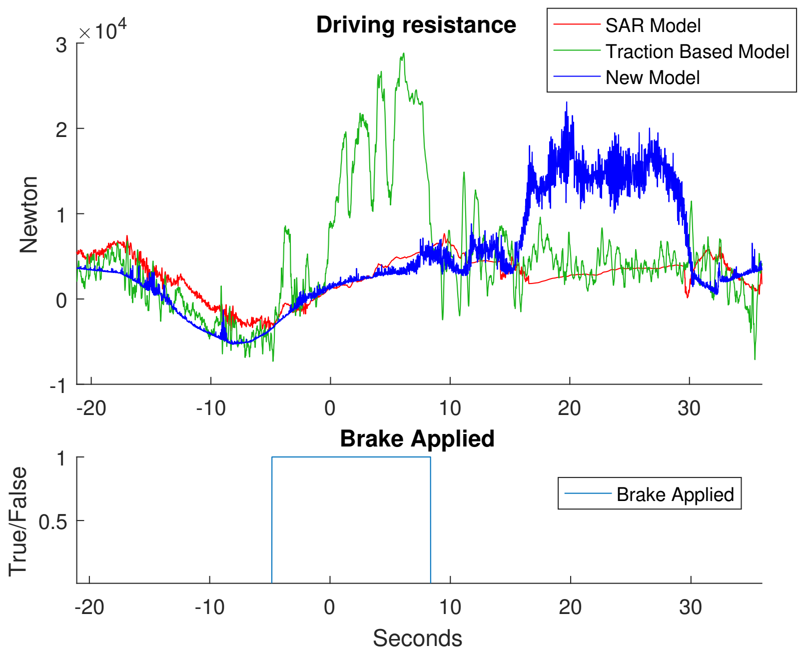

Figure 3.1.2: The driving resistance of a truck driving straight ahead and into a roundabout. Brake is applied during the straight road, as can be seen in the lower plot.

3.1.3 Change of Slope

The slope should increase the driving resistance when it is positive, i.e uphill slope, and decrease the driving resistance when negative, i.e downhill slope. In Figure 3.1.3 the truck is driving downhill. The new model has a smooth behaviour and follows the expected path. The SAR model, too, has a smooth behaviour but has a slower phase than the new model. This is because the SAR model is filtered, which makes it less noisy, and the new model is unfiltered. The traction based model is noisy, with an unclear path, since brake is often applied when going downhill.

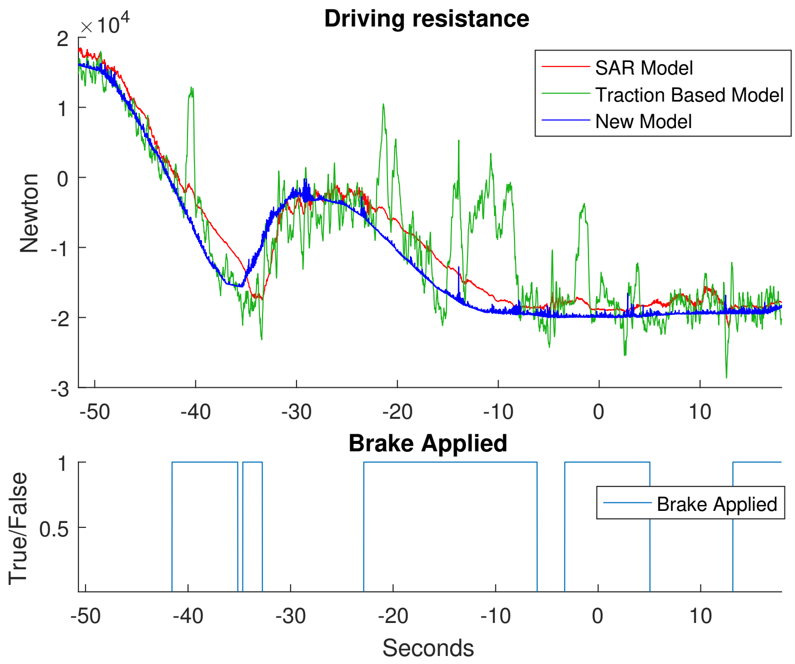

Figure 3.1.3: The driving resistance of a truck driving downhill. The lower plot shows when brake is applied.

3.2 Model of Truck with Semi-Trailer

In order to validate the model of a truck with a semi-trailer a Scania S500 4x2 truck with a semi-trailer was used. The data was collected in regular traffic in Södertälje. The specifications of the truck and the semi-trailer are presented in Table 3.2.

model	L[m]	L_w[m]	L_t[m]	L_a[m]	m[kg]	m_t[kg]
Scania S500 4x2	4	2.2	9	1	16900	22100

Table 3.2: Specifications of the truck and semi-trailer used for evaluation.

3.2.1 Cornering

Since lateral forces are added to the old longitudinal model, and since the lateral forces increase when cornering, the driving resistance is also expected to increase when cornering. Compared to the rigid truck model, the driving resistance of the truck with a semi-trailer should be larger in magnitude. This is mainly due to it having a larger side area and being heavier. As can be seen in Figure 3.2.1, the new model shows a clear behaviour of an increased driving resistance signal when the vehicle is driving in the roundabout. Compared to the driving resistance of a rigid truck in the same driving situation, which can be seen in Figure 3.1.2, the magnitude of the driving resistance of a truck with a semi-trailer seems to be more than twice as large.

3.2.2 Braking

The new model should not be affected by the vehicle braking, i.e. it should keep a smooth behaviour even when brake is applied. In Figure 3.2.1 the vehicle brakes right before entering the roundabout. As expected, the new model does not show any inclination of any unexpected changes and keeps a smooth and stable behaviour.

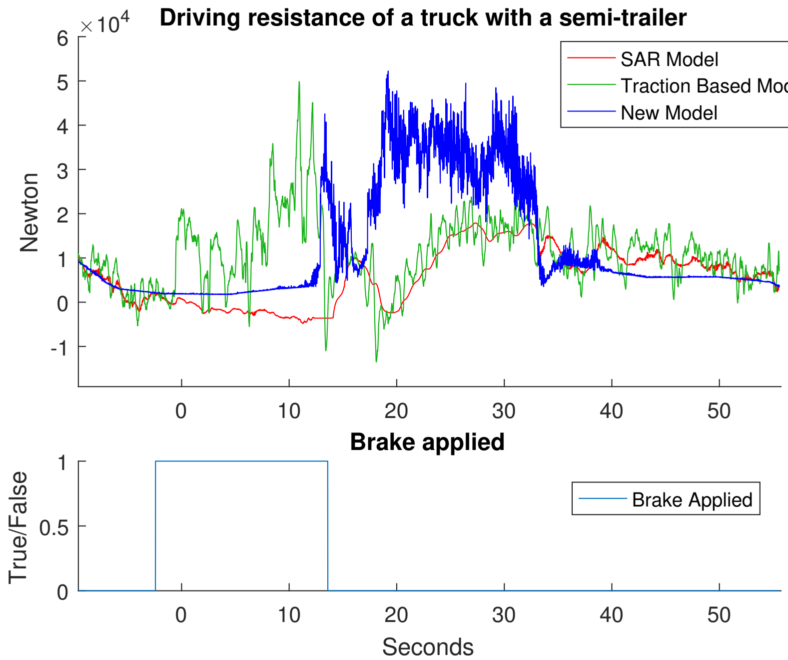

Figure 3.2.1: The driving resistance of a truck with a semi-trailer when driving in a roundabout. The lower plot shows when brake is applied.

3.2.3 Change of Slope

When driving downhill, or uphill, the new model should behave similarly to the old SAR model. In Figure 3.2.2 the vehicle is driving downhill, and sometimes brakes. Since the traction based model does not work when the vehicle is braking, this signal is off at times, but the SAR model and the new model keep a stable and very much alike behaviour.

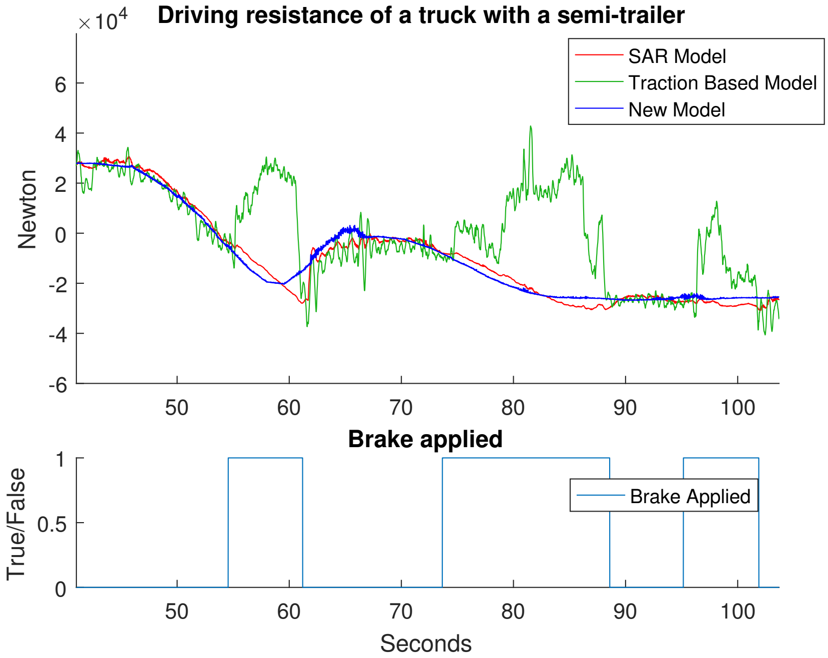

Figure 3.2.2: The driving resistance of a truck with a semi-trailer when driving downhill. The lower plot shows when brake is applied.

4

Filtering and Fusion of Models

Since the new models are improvements of the old SAR model, the old one does not contribute with any additional information. Hence, only the old traction based model and the new models are used hereafter. The new models have proven to be quite accurate at all times. However, since the traction based model is well established, and is not based on estimations, it is preferable to use this model as often as possible. Therefore, the suitable new model for the vehicle driven, a rigid truck or a truck with a semi-trailer, is used as a complement to the traction based model. Depending on the driving situation at hand the model to use is chosen, and so a fusion of the models is created. Since there are two new models there are also two fusions, one for a rigid truck and one for a truck with a semi-trailer.

In order to make further reasoning easier the notation will be changed slightly. From now on, the model for driving resistance based on the traction force, presented in (1.1), will be referred to as $F_{d\_traction}$. The new models of the driving resistance developed in this thesis, presented in (2.12) and (2.20), will be referred to as F_{d_new}. The filtered and smoothed fusion of the models, which will be the complete and final model of the driving resistance, will get the notation F_d. A schematic picture of the fusion is shown in Figure 4.0.1.

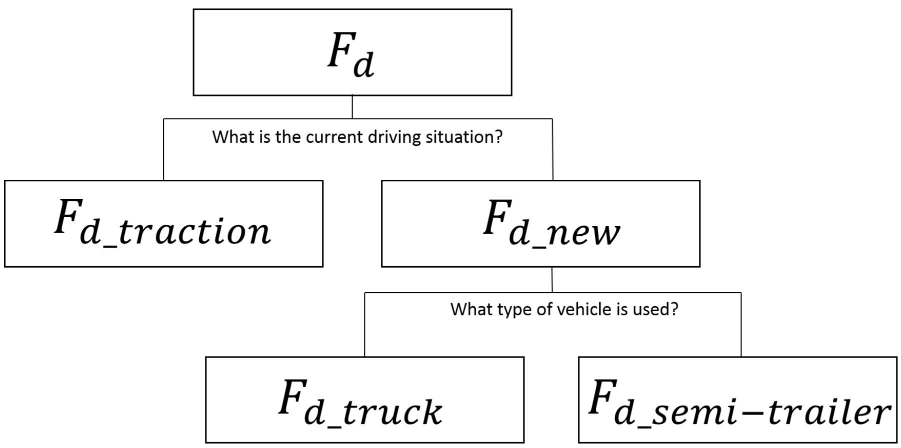

Figure 4.0.1: An illustration of the fusion algorithm.

4.1 Single Exponential Smoothing

Single exponential smoothing is a method for smoothing and forecasting data by using an exponential window function. The method does not require any minimum number of observations to be made beforehand. Hence, the method will produce a smooth and reliable result right from the start. Single exponential smoothing estimates the weighted average, s_t, based on the current observation, x_t, and the previous smoothed average, s_{t-1}, as

$$s_t = \beta x_t + (1-\beta)s_{t-1}. \tag{4.1}$$

Thus, it is a recursive method. Here, $t > 0$ and β is the data smoothing factor. The smoothed result at one time instant t, will be dependent on all previous observations but mainly on the most recent one. This is because the weight of the most recent observation is set to be the largest, and because the weights decrease exponentially with time. The weights are directly correlated to the smoothing factor, β, which is chosen to be a value between zero and one. A large value of β means that little smoothing is applied and that the method is not as dependent on old observations, i.e. that it forgets faster. If the smoothing factor is estimated to be above 0.3, a more complex filter is favourable according to [17]. When using exponential smoothing it is also essential to set an initial value. It is important to note that the smaller the value of β is, the more sensitive the smoothed result will be to the selection of this initial value.

4.2 Choosing Smoothing Factor

Since the driving resistance is estimated in real time while driving, it is important that the signal is fast and without phase shift. If a phase shift would be present, it could be the reason for an uphill or a curvature being represented too late in the driving resistance signal. As a consequence, a too high gear choice could cause a breakdown. Therefore, it is important that the filter will not make the signal too delayed when choosing the smoothing factor. On the other hand, it is desirable to have a signal which is as smooth and continuous as possible in order to reduce unnecessary gear shifts. If the driving resistance signal is very noisy it might indicate that the gear should be shifted, but then in the next time instance it indicates that it should shift back again, and so on. Considering this, it is possible to say that a more steady and smooth signal is much preferred. In Figure 4.2.1 the resulting fusions, with different smoothing factors, are displayed. As can be seen, a smoothing factor of $\beta = 0.07$ gives quite a noisy result whereas a factor of $\beta = 0.01$ gives a very smooth, but a somewhat delayed, result. Since a delay of the signal is very much undesired, a smoothing factor bigger than $\beta = 0.01$ is chosen. $\beta = 0.03$ gives a smoother result than $\beta = 0.07$, but is not as delayed as $\beta = 0.01$ and is thus an appropriate choice.

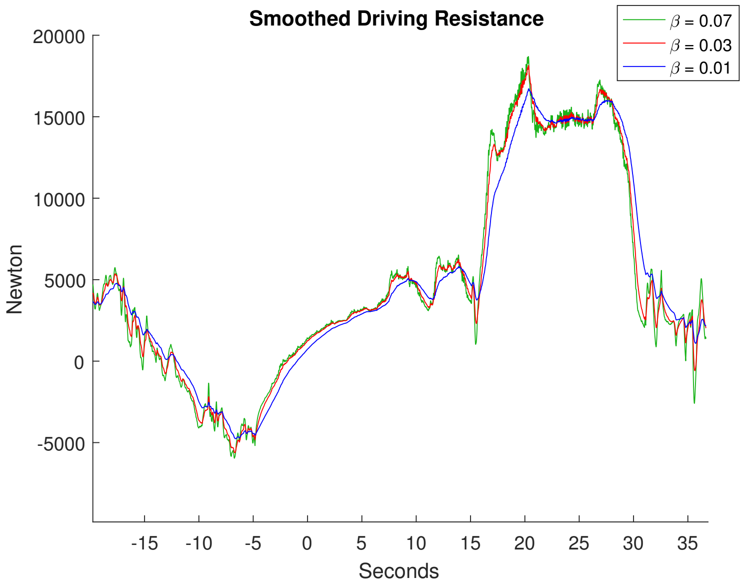

Figure 4.2.1: The smoothed fusion with three different smoothing factors.

The single exponential smoothing method is here used in order to fuse two different models together. Since these two models have a very different degree of noise present, it could be desirable to have two different smoothing factors. The traction based model $F_{d\_traction}$ is the more noisy one out of the two, and thus β is set to be significantly smaller when using this model than when using F_{d_new}. In Figure 4.2.2 the improvement of choosing two different β is shown. As can be seen, the final model has successfully reduced the noise from $F_{d\_traction}$ and the result is a smooth signal which is not delayed.

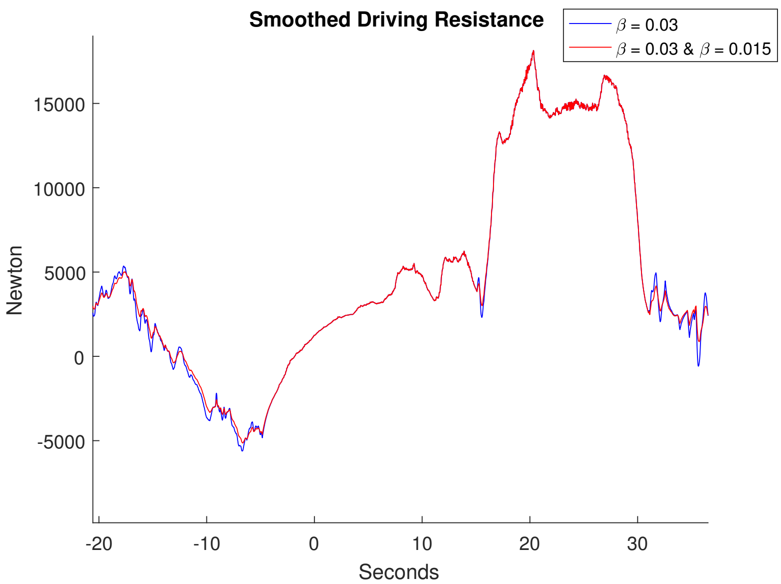

Figure 4.2.2: The smoothed fusion when using the same smoothing factor for both models and when using two different factors.

4.3 Fusion Algorithm

The fusion algorithm is presented in Algorithm 1. By having the most accurate initial value possible, the settling time of the algorithm will be significantly shortened and the single exponential smoothing will perform better. Thus, the initial value of the smoothing algorithm is defined as the first value of F_{d_new}, since F_{d_new} is less noisy than $F_{d\_traction}$. The update of the filter, x, is varied between the two models. The conditions for when each model should be used are defined as *in curvature*, *brake applied* and *low velocity*. These driving cases, when brake is applied, in a curve and during low velocities are when F_{d_new} has a better estimate than $F_{d\_traction}$. Hence, $x = F_{d\_new}$ is used in these cases. Otherwise, $x = F_{d\_traction}$ is used. The last step of the algorithm is the single exponential smoothing presented in (4.1). Since the filter updates come from both models, a fusion of the models has been created. In Algorithm 1, there are two different smoothing factors chosen, one for each model, as discussed in Section 4.2.

Algorithm 1: Fusion by smoothing

```
F_d(0) = F_d_new(0);
for every time instance i do
    if in curvature or brake applied or low velocity then
        x(i) = F_d_new(i);
        β = 0.03;
    else
        x(i) = F_d_traction(i);
        β = 0.015;
    F_d(i) = βx(i) + (1 − β)F_d(i − 1)
```

4.4 Evaluation of Fusion

By first doing an evaluation by simulation, and then, when the results are satisfactory continue on with an evaluation by real life testing is often advantageous. It is generally more complicated and time consuming to fix implementation mistakes in real life testing than in simulation. Thus, by simulating first many of the mistakes can be found and adjusted before implementing the code in a real truck. Since two models have been developed in this thesis, the simulation of fusion algorithm is divided into two parts. First the model of the rigid truck is evaluated and then the model of the truck with a semi-trailer. The same goes for the real life testing. The tests are performed by driving in different driving situations in order to further see if the driving resistance estimation is smooth, and as a result from this that also the gear shifts are performed smoothly. The goal when driving in real life is to get a truck which shifts smoothly between gears.

4.4.1 Simulation of Truck

The simulation of a rigid truck is based on data collected with a Scania G500 6x4. Its specifications are presented in Table 3.1. The smoothed fusion should choose which model to update from depending on the current driving situation. The intention is that it chooses to trust F_{d_new} only in the situations in which the vehicle is braking, cornering or driving with low velocity and otherwise rely on $F_{d\_traction}$. When switching between models, the smoothed fusion should not be left with any peaks or bumps, but should rather be smooth and let the switch go by unnoticed. Figure 4.4.1 shows $F_{d\_traction}$, F_{d_new} and the smoothed fusion model of a truck which is driving into and in a roundabout. Both $F_{d\_traction}$ and F_{d_new} are noisy. However, the smoothed model is smooth enough to use as an input signal to the gear choice without having to be concerned about switching gears up and down unnecessarily. The lower plot of Figure 4.4.1 shows when brake is applied, when driving with low velocity and when cornering is true, respectively. In other words, when at least one of the displayed signals in the figure is true, the smoothed fusion should take its update from F_{d_new}. Therefore, in this particular case the smoothed model should trust F_{d_new} almost the whole session. As can be seen in the upper plot of the figure, this is true. The two models have very similar behaviour most of the time.

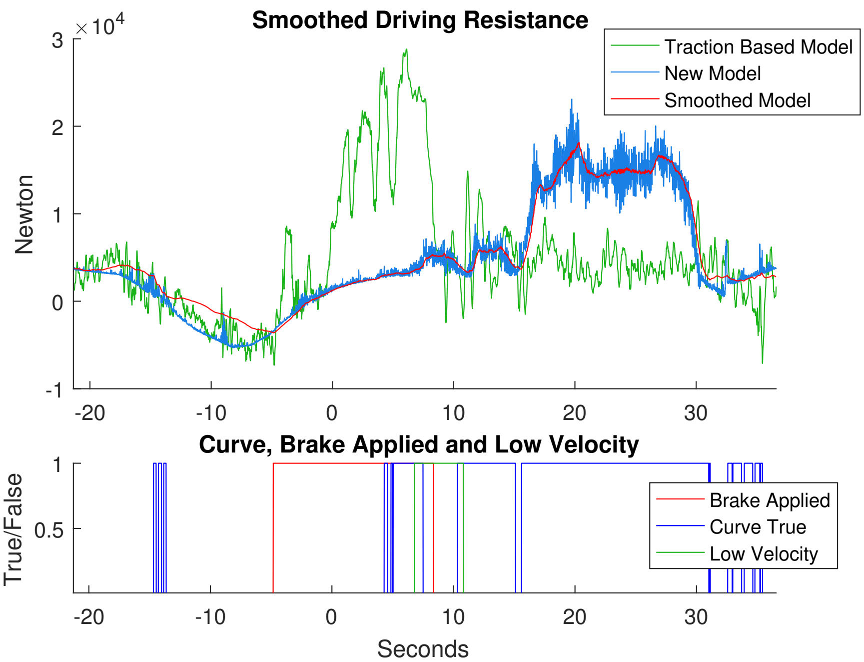

Figure 4.4.1: The fused and smoothed model when a truck is driving in a roundabout. The lower plot shows when the vehicle is braking, cornering or driving with low velocity.

4.4.2 Simulation of Truck with Semi-Trailer

The data used for simulating the smoothed driving resistance of a truck with a semi-trailer was collected with a Scania S500 4x2 truck with a triaxial semi-trailer. The specifications of this truck and the articulated semi-trailer is presented in Table 3.2. When at least one of the three signals in the lower plot of Figure 4.4.2 is true, the smoothed model of the driving resistance should follow the new model's, F_{d_new}'s, behaviour. Otherwise, the smoothed model should trust the traction based model, $F_{d\_traction}$. By looking at the figure, it is possible to say that this is true. The smoothed model is quite noisy at times, but hopefully not so noisy that it will have a significant effect on the gear choice. If this would be the case though, the solution is to have smaller smoothing factors.

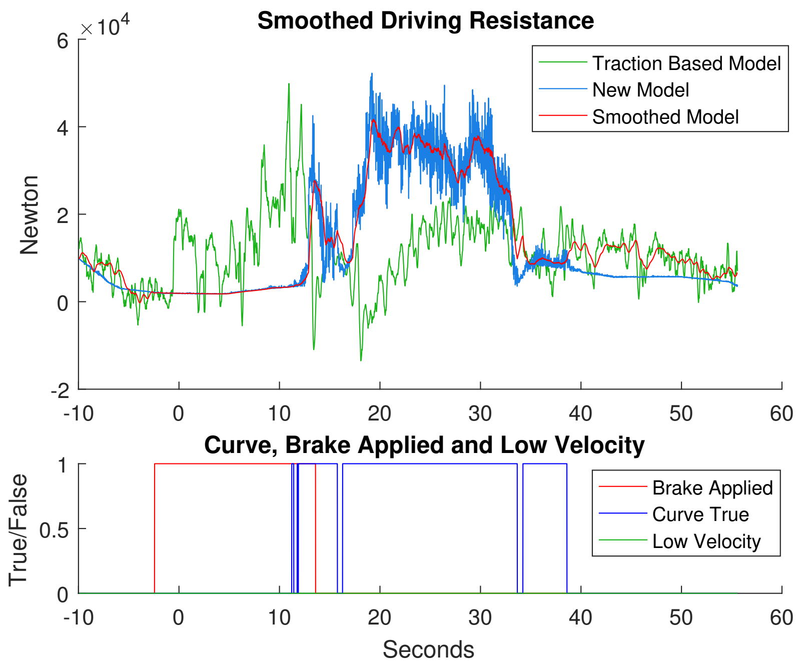

Figure 4.4.2: The fused and smoothed model when a truck with a semi-trailer is driving in a roundabout. The lower plot shows when the vehicle is braking, cornering or driving with low velocity.

4.4.3 Real Life Testing of Truck

When testing the rigid truck model in real life a Scania R650 8x4*4 truck was used. The truck's parameters are presented in Table 4.1. Overall, the truck behaved as expected and the test drive was a success. In Figure 4.4.3 the driving resistance of

model	L[m]	L_w[m]	m[kg]
Scania R650 8x4*4	5.2	2.2	29000

Table 4.1: Specifications of the truck used for real life testing.

the smoothed model in real life and the old real life model is shown together. The driving situation at hand is first a turn leading into a descending slope, then another turn and lastly an uphill slope of 20%. Worth mentioning is that the data has been collected during two different drives, which means that they are not identical in terms of speed and when brake is applied. However, since it is the same route that has been driven it is still possible to make a comparison. As can be seen in the figure, the two models behave quite similarly, even in some cases when the vehicle is cornering.

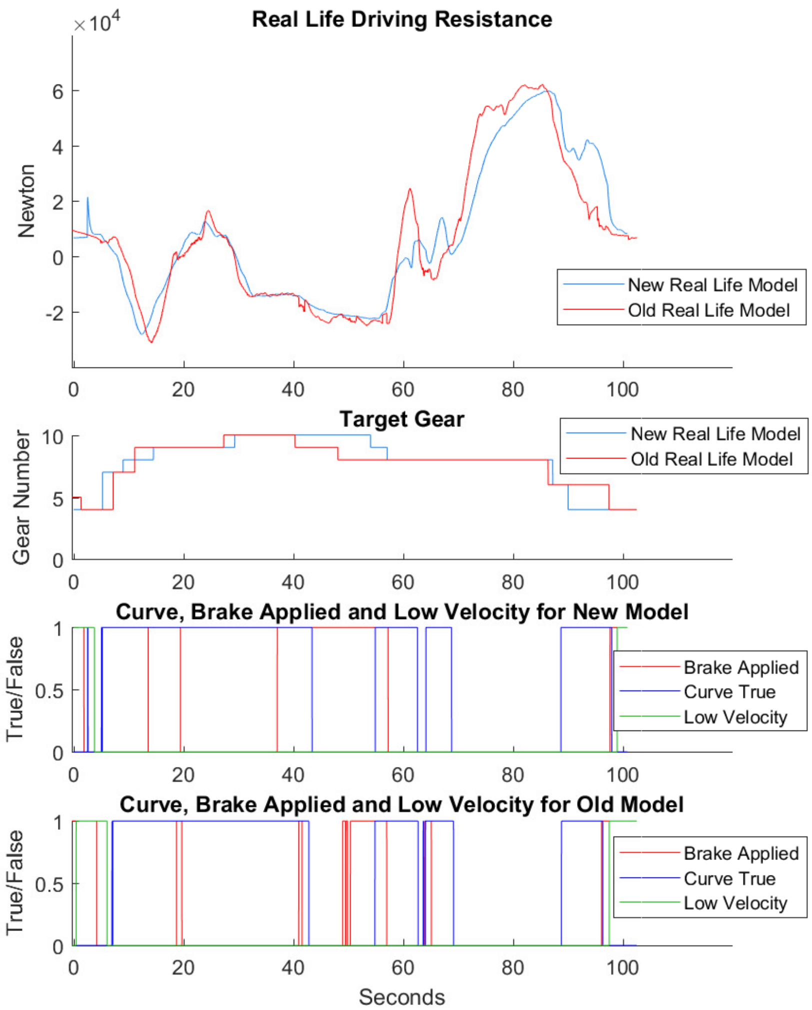

Figure 4.4.3: The driving resistance when using the old model and the new smoothed model in a real truck, together with the target gear of each model. The last two plots shows when the vehicle is braking, turning, and driving with low velocity for each model.

When instead comparing the smoothed model in real life with itself but from another drive, it is possible to establish that the driver and the way of driving is very significant. In Figure 4.4.4 the driving resistance when driving in the same driving situation as in Figure 4.4.3 is shown. During the first drive an engine failure occurred, at time 60 seconds, when driving on the upward slope. During the second drive however, the slope did not cause any trouble and the route went as expected. The reason for this difference when driving the exact same route, with the same model, the same truck and the same driver could only be the way of driving. The first time around, the vehicle accelerated right before the start of the slope, an action which was not repeated during the second drive. However, when looking at the driving resistance for the two cases they are almost identical, except for the second drive being with a lower velocity. Therefore, it is hard to say what causes the gear upshift at the time 46 seconds. The conclusion drawn is that it has nothing to do with the driving resistance but rather with something else which also influences the gear choice, for example the velocity or how much the gas pedal is pushed.

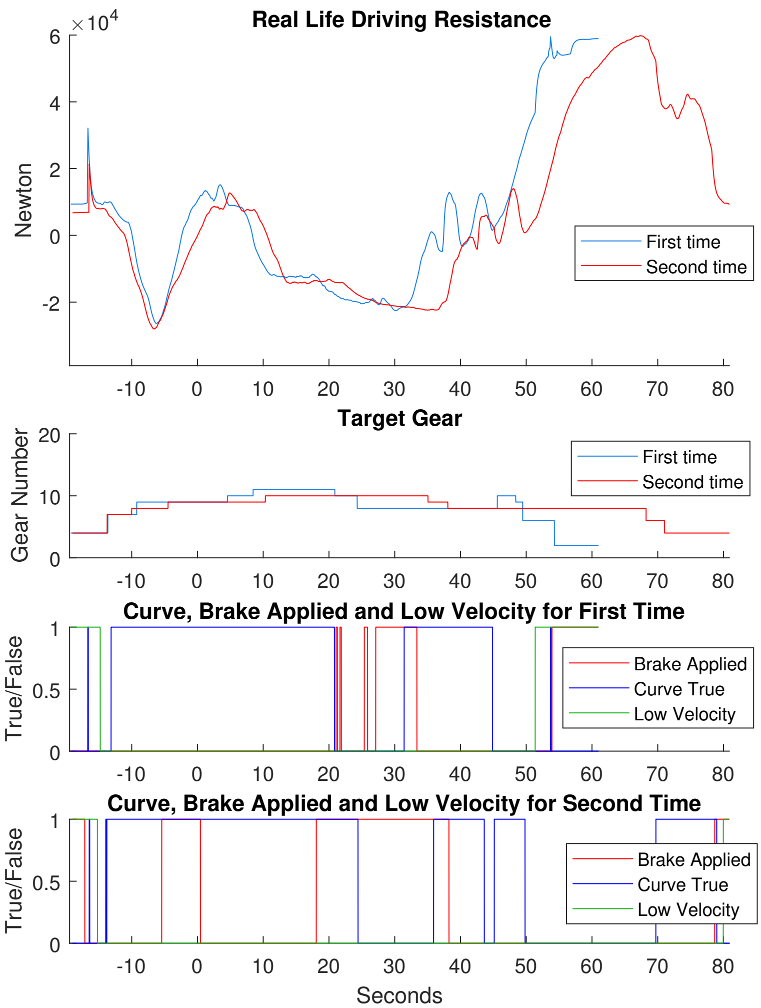

Figure 4.4.4: The driving resistance of the smoothed model in real life, as well as the target gear and information about if the truck is in a curve, braking or driving with low velocity, for two different drives.

4.4.4 Real Life Testing of Truck with Semi-Trailer

The real life testing of a truck with a semi-trailer was made with a Scania S500 4x2 and a triaxial semi-trailer. The parameters of the vehicle are presented in Table 3.2. During the test drive no engine failure or other unexpected movements occurred, which indicate that the model for a truck with a semi-trailer operates as anticipated. The same driving situation as for the rigid truck was tested. The driving situation is extreme, with a 20% uphill slope, in order to make sure that the new smoothed model can handle all kinds of situations. Figure 4.4.5 shows the driving resistance of a truck with a semi-trailer, both with the new smoothed model and with the old model, as well as the target gear for each model. The bottom two graphs show when brake is applied, when the vehicle is cornering and when it is driving with low velocity. The two driving resistance signals are quite similar, especially when there is no brake applied and when the vehicle is not making a turn or driving with low velocity. This is expected, since both the new and the old model use the traction based model during these instances.

As with the rigid truck, the results when driving the same route more than one time with the new smoothed model, will look slightly different from time to time. The reasons for these differences are the way of driving and the driver.

When comparing Figure 4.4.3 with Figure 4.4.5, the differences between the truck and the truck with a semi-trailer can be seen. The driving resistance of a truck with a semi-trailer is larger than of a rigid truck. This is reasonable, since the mass and length of the vehicles are different. The behaviour of the new smoothed model, with the exception of the magnitude of the driving resistance, is similar for both of the vehicles. This is anticipated, since a lateral force has been added in both of the models.

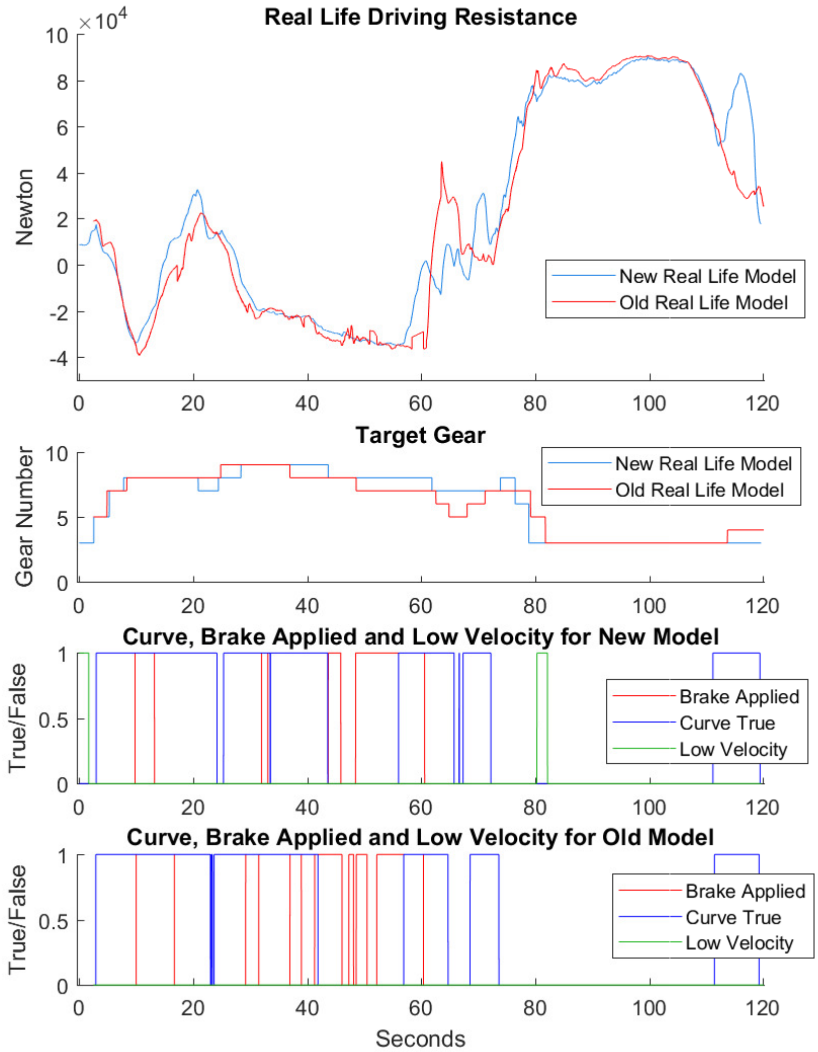

Figure 4.4.5: The driving resistance when using the old model and the new smoothed model in a real truck with a semi-trailer, together with the target gear of each model and information about if the vehicle is in a curve, braking or driving with low velocity.

5
Discussion

It has been shown that the new models for the driving resistance behave similarly to the old model. They all work as expected and without complications. There are no obvious driving situations in which the old model works better than the new ones, or vice versa. Therefore, either of the models could be used. In order to determine which model is the better one, other aspects, rather than just the ability to keep the truck from having an engine failure, have been evaluated. Those are the choice of gear and how the vehicle shifts between gears, the drivability and the emission impact.

5.1 Gear Shift

The gear shifting is expected to be smooth, without any unexpected jumps, and is expected to continuously choose the highest gear fitting. Since the new models are modifications of the old model, the target gear and the driving resistance should be different when cornering, but similar otherwise. In order to evaluate the gear shift, a test drive was done repeatedly on a specific route in Södertälje, which is shown in Figure 5.1.1. The route started with a right turn to the main road, downhill into a roundabout, straight uphill to a second roundabout and ended with a turn back to the starting point. For the test, a rigid truck, see Table 3.1, and a truck with a semi-trailer, see Table 3.2, were used. The route was driven twenty times. There were two drivers who switched vehicle after ten drives, five times with the new driving resistance model and five times with the old. From the results, mean values have been calculated for the new and old models of the truck, as well as the new and old models of the truck with a semi-trailer. Since the route was on public roads with traffic, the behaviours of the other vehicles and people crossing the roads have had an impact on the result. The mean values for the truck respectively the truck with a semi-trailer are compared below, in order to evaluate how the new and old models differ.

In Figure 5.1.2 the mean driving resistances of the old and new models for the rigid truck are plotted. It is possible to see that the driving resistance of the new model is larger compared to the driving resistance of the old model when the vehicle is cornering, at 20 seconds, and in the roundabouts, at 100 seconds and 180 seconds. This causes the choice of gear to be slightly lower for the new model in these driving situations. Quite common for the new model was gear number five or six in the roundabouts, whereas the old model opted for gear number six or seven.

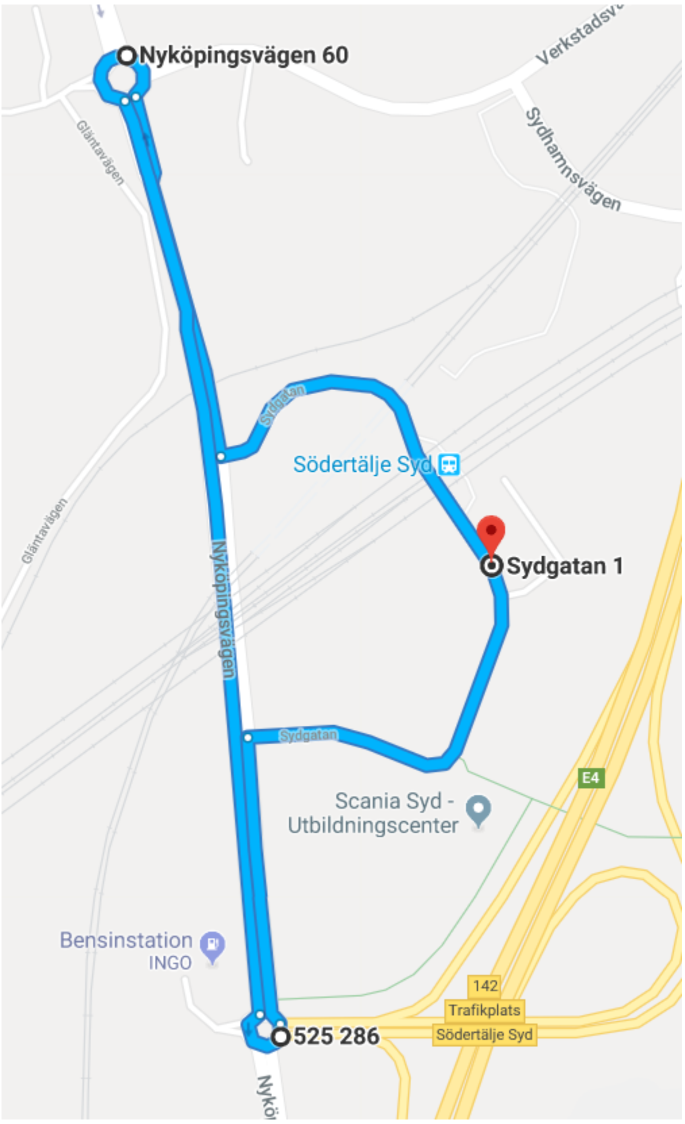

Figure 5.1.1: The route driven for testing. Sydgatan 1 is start and stop. The route was driven counterclockwise.

This is not only possible to establish by looking at the figure, but was also noted while driving. When not cornering, the two mean gear target signals are not always similar to each other, as they are modelled to be. The mean driving resistances, however, are almost always identical when the vehicle is not cornering. This means that the difference in the gear choice is not due to the driving resistance, and it is believed that the difference would disappear if even more test drives were included in the mean value.

A lower gear when cornering makes it possible for the truck to accelerate faster when exiting the roundabout or the curve. This contributes to the responding time of the vehicle being perceived as faster, and the general behaviour of the vehicle as more aggressive. Worth considering though, is if this is necessary. The old model would not be described as slow or unresponsive, at least not while driving on a public road. Off-road vehicles, however, need a more aggressive gear selection due to steeper hills and tougher terrain. Taking this into consideration, a possible conclusion could be that the new gear selection is unnecessarily low for on-road usage, but may be perfect for off-road vehicles.

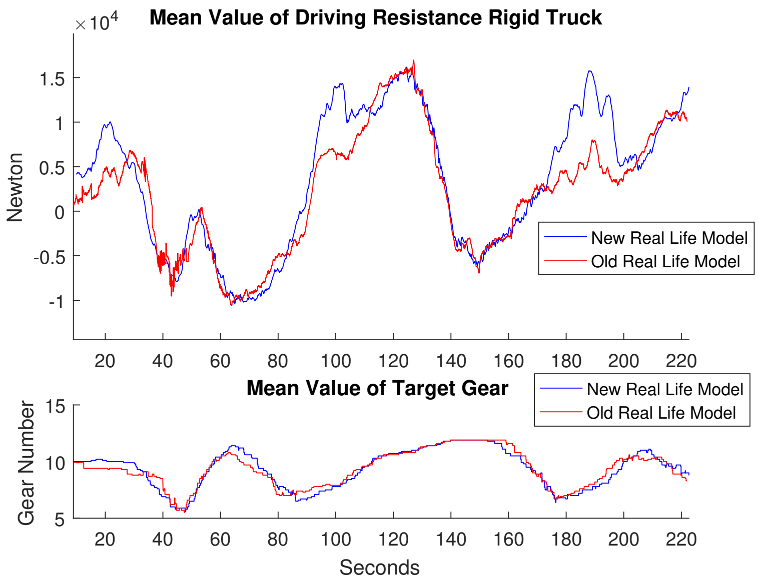

Figure 5.1.2: The mean driving resistance and target gear of the new model for a truck and the old model.

When instead considering a truck with a semi-trailer, similar conclusions are made. Figure 5.1.3 shows the mean values of the driving resistance and the target gear for the truck with a semi-trailer, for both the old and the new model. The new model increases during cornering, clearly shown at 120 seconds and 220 seconds, which gives a larger driving resistance for the new model compared to the old. In the second roundabout, which is small, the new model chooses a lower gear, gear number three, compared to the old model which chooses gear number four. This is expected, since the gear choice is directly related to the increased driving resistance when cornering. However, the engine rotational speed was slightly too high for the new model, 1400 rpm instead of 1000 rpm as for the old model. In the straight forward uphill sequence the traction based model is chosen in both models. Hence, the resulting driving resistance is the same for both of them. During the drive, both brake and retarder were used, especially during the downhill part towards the first roundabout. This gives a strange behaviour for the old model, at 70 seconds. However, the new model was not affected by it and has the expected behaviour.

The drivers have a big impact on the result, both on how the curve is taken and on the magnitude of the driving resistance. One of the drivers gave a consistent result of a little larger driving resistance compared to the other driver. Therefore, the gear shift is different when comparing the different drivers too.

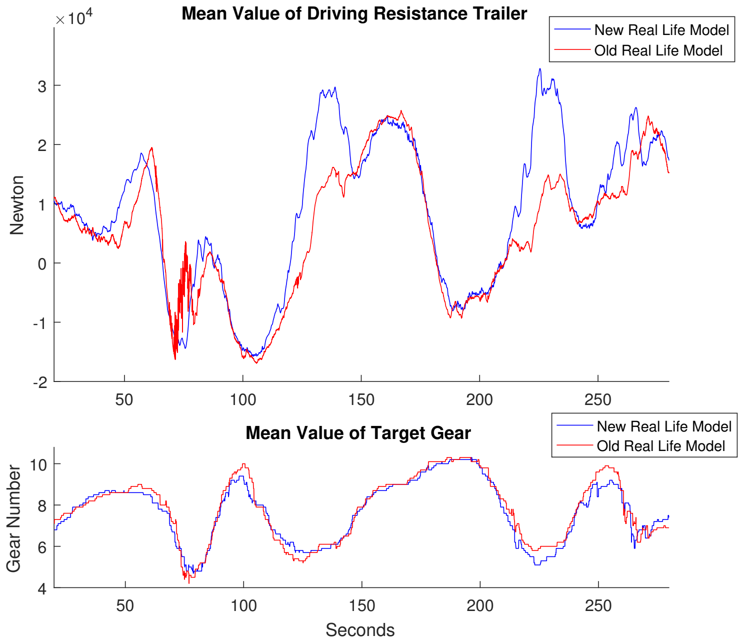

Figure 5.1.3: The mean driving resistance and target gear of the new model for a truck with a semi-trailer and the old model.

5.2 Drivability

Since good drivability is subjective, it is essential to let a number of different drivers test drive trucks with the new gear selection and driving resistance model in order to evaluate this aspect. Each driver's opinion has been considered, and a joint final conclusion has been drawn. The evaluation was made by driving in regular driving situations, in order to see if the truck drivers felt any difference in drivability with the new model. According to the drivers, the new drivability of a truck was good. They felt that the truck could handle all situations it was confronted with and that the truck reacted quickly to the driver's actions. Furthermore, they thought that the drivability with the new model was as good as with the old model, and that it was not possible to recognise the differences between them. These are results that are very satisfactory. The drivers' opinion of the drivability of a truck with a semi-trailer was that it was adequate. However, some of the gear shifts when cornering were thought to be unnecessary because of the high rotational speed of the engine. Hence, a small difference between the new model and the old model could be felt when cornering. However, the difference is not significant and in most situations the models feel equal. In conclusion, the drivability when using either of the new models is similar to the drivability when using the old model.

5.3 Fuel consumption

There are many factors affecting the fuel consumption of a truck. One of them is the gear choice and thus, also the driving resistance. But, since the gear choice is one of many components, changing the gear shift will not have a huge effect on the total fuel consumption of the truck. Hence, changes in the driving resistance model, such as those made in this thesis, will have an even smaller impact. Even if the driving resistance when cornering is modelled to be more than twice as big compared to the old model, this is such a specific driving situations that it will not show in the final results. When considering the whole lifetime of a truck, it is only a fraction of that time that the truck spends cornering. Thus, even if the fuel consumption momentarily is very bad when making a sharp turn, it will not influence the total fuel consumption of the truck with even 1%. In conclusion it can therefore be said that the fuel consumption of the old and the new models are equally good.

5.4 General Discussion

Even though the new models seem to have a correct behaviour, it has not been properly established that they are in fact correct. Since there are a limited number of sensors in a truck, and since the scope of this thesis did not include adding any additional sensors, the values estimated in this thesis have not been compared to the real measured values of the same signal. Instead, the models have been evaluated based on what is thought to be reasonable. Thus, the exact accuracy of the system is not known. If sensors which measure rolling resistance, the side slip angle and the articulation angle were added to a truck it would be possible to say if the correct assumptions have been made and further evaluate the accuracy of the models.

The new models are stated to behave as expected at low velocity. However, the steering angle used when modelling the side slip angle is the one for high velocities, presented in (2.7). Therefore, it could be possible to further improve the behaviour of the new models by setting the steering angle to be equal to the Ackermann angle, shown in (2.6), when the velocity is low. Otherwise, the high speed steering angle should be kept. This could be done by an ordinary if-statement. Nevertheless, the current behaviour of the new models are good and it is uncertain how much of an impact this alteration would give.

When evaluating the new models and comparing them to the old model, different driving situations were considered. However, not once were the trucks reversed. Therefore, it has not been established how the models perform during this action. Even so, there are no obvious reasons for the new models to behave differently from the old one when reversing. As have been proven, the models perform similar on all occasions. Thus, it is believed that they will continue to do so when reversing.

The smoothing factors seem to be correctly chosen and create the behaviour which is wanted for the filter. However, the factors have been found by trial and error while simulating. Therefore, it could be favourable to optimise the factors more in real life to make sure that the most suitable factors are chosen. Also, an evaluation of the difference between the factors for a rigid truck and a truck with a semi-trailer should be done. Even though the smoothing factors do not differ much when changing a decimal, it might have a small impact on the final model.

When evaluating the gear choice, it is stated that the new models for a rigid truck and a truck with a semi-trailer have a slightly more aggressive behaviour than the old model. Therefore, it could be favourable to use them in off-road vehicles. However, the road grade in the new driving resistance models are based on map data, which is only possible to retrieve from public roads. Thus, the errors related to the longitudinal accelerometer will still be a problem. This will mainly cause complications when driving with low velocity, since the cornering force has been modelled separately. Unluckily, off-road vehicles tend to drive with low velocity more frequently than on-road vehicles.

6
Conclusions and Future work

By adding a lateral force to the SAR model, the accuracy when cornering is improved significantly. Previously, the SAR model could not be trusted in this driving situation due to the uncertainties of the measurements from the longitudinal accelerometer. Now, this driving situation is not dependent on the longitudinal accelerometer anymore, but rather on the lateral accelerometer. Further improvement of the SAR model has come from the decision to remodel the road grade in such a way that this signal, too, is independent of the longitudinal accelerometer. Since map data is used as often as it is available in the new models, they can handle driving situations which require low velocity much more accurately than the old model. The strengths and weaknesses of the traction based model and the improved SAR model are displayed in Table 6.1.

By keeping the traction based model as it was and by focusing on improving the SAR model, a new improved model has been developed. This new model is not superior to the old model in behaviour, but has the advantage of always providing a correct estimate. As can be seen in Table 6.1, there are no longer a driving situation which the combination of the traction based and the SAR model can not handle. Therefore, there is never the need to freeze values. This improves the overall accuracy of the driving resistance and the gear selection.

Traction based model, (1.1)		**Improved SAR models, (3.1) and (4.1)**	
+	Curvature	+	Brake
−	Brake	+	Standstill
−	Low velocity	+	Curvature
−	Standstill	+	Low velocity

Table 6.1: The strengths and weaknesses of the traction based model and the new improved SAR models.

The old SAR model has been modified to fit two different types of heavy-duty vehicles. The first modification resulted in a model for a rigid truck, and the second in a model for a truck with a semi-trailer. The behaviours of these two models are very similar. The only difference is the magnitude of the driving resistance, which is larger for the truck with a semi-trailer. This comes as no surprise, since a truck with a semi-trailer is both heavier and longer than a rigid truck. The increased driving resistance causes the gear choice to be slightly lower compared to the gear choice of the rigid truck. This is convenient, since heavier vehicles usually need a lower gear to manage.

6.1 Future Work

A future improvement of the driving resistance model could be to develop a prediction model. A prediction model can predict the future, in this case the gear shift, by looking at the map data of the road ahead. Changes in road grade, road conditions and turns are situations which the prediction can take in consideration for the gear shift. Furthermore, every vehicle on the road affects the other vehicles with its velocity and different driving plan. However, with sensors such as radar, the truck can anticipate their actions and adapt the velocity after the situation. By doing this the gear shift would be improved and unnecessary shifting could be avoided.

Every component of the driving resistance which is estimated might be possible to improve. The vehicle mass and the road grade are considered known in this thesis. Hence, modification of the estimation of them could give an improvement of the final model. Furthermore, even if the aerodynamic drag equation is quite established, components in this equation could be improved. For example, the aerodynamic drag coefficient could be decreased by platooning.

The sensors used in this thesis have been limited to the ones already existing on the truck and the semi-trailer and have been assumed to be correct. Hence, to further improve the accuracy of the driving resistance, the sensors which exists and their error margin should be evaluated. Another scope could be to use a truck with more sensors and evaluate if the estimations and assumptions made in this thesis seem to be correct. For example, comparing the driving resistance estimation with a measured value of the driving resistance may indicate that more alterations are needed to the estimation.

In theory, no large decrease of the fuel consumption is due to happen in consequence of the improvement of the driving resistance. In reality, however, this has not been proved. Hence, a future task could be to clarify this by simulation or measurements in real life.

There exist a lot of different vehicle configurations with a different number of trailers, wheels and axles. However, in this thesis only the driving resistances for a rigid truck and a truck with a semi-trailer have been considered. Therefore, a next step could be to look at other truck configurations such as a rigid truck with a trailer.

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