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Identification of tyre characteristics using active force excitation

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ABSTRACT: Knowledge of the maximum tyre-road friction coefficient can improve active safety systems by defining actuator boundaries and adaptable intervention thresholds. Estimation of the coefficient of friction based on tyre response measurements requires large level of force excitation. Under normal driving conditions, manoeuvres with large tyre utilizations are rare. This study investigates a method where wheel torques with opposite signs are applied to the front and rear axle simultaneously. This procedure allows for an intervention with large tyre excitations without disturbing the motion of the vehicle. The intervention is evaluated in simulations and experiments. Further, a method is proposed which does not require measurement of the vehicle longitudinal velocity. The results show that it is possible to estimate the current friction coefficient with the proposed method, although the assumption made in the proposed method makes the friction estimate sensitive to measurement noise on the wheel speed signal.

1 INTRODUCTION

The demand on improving the road traffic safety is increasing. Sweden have for example formulated vision zero which states that no one should die or suffer serious injuries in traffic. Active safety systems are effective in reducing the number of crashes and the severity of the crashes. The active safety functions of today have very limited information about the present tyres and road conditions. The effectiveness of these systems can be improved if the tyre-road characteristics are estimated online. One very important parameter in this context is the maximum tyre-road friction coefficient. Identifying some tyre characteristics such as the tyre to road friction coefficient up to the limit of adhesion normally requires high levels of tyre excitation, see (Svendenius, 2007). The friction estimation problem has been investigated previously in literature and the most common approach is to wait for the driver to perform a manoeuvre where the tyre-road friction can be estimated, see for example (Svendenius, 2007), (Gustafsson, 1997), (Ahn et al., 2012) and (Rajamani et al., 2012). In normal driving conditions these events are rare and consequently also the opportunities to estimate the tyre-road friction coefficient.

In (Ekmark and Jansson, 2007) & (Matsumoto et al., 2002) methods are proposed where positive wheel torque is added to the front wheels at the same time as negative wheel torque is added to the rear tyres. This allows for a net force neutral intervention where the tyres reach high levels of excitation without disturbing the longitudinal motion of the vehicle. The same approach is used in (Chen and Wang, 2011) in simulations to show that the method can be used to find the road-friction coefficient. However, there is no literature known to the authors where the method has been implemented and tested in a real vehicle. The present study investigates how this method can be implemented in a real vehicle and how the signals required for estimating the tyre-road friction coefficient can be obtained using this method with standard productions sensors. In particular a
method is proposed which removes the need to know the longitudinal velocity. The method and the validity of the assumptions made are evaluated through simulations as well as experiments. A tyre model is fitted to the measurement data using an offline nonlinear least square method to illustrate how the tyre characteristics can be found from the measurement data.

2 METHOD

The goal of the proposed method is to enable estimation of the maximum road-tyre friction coefficient at normal driving with low levels of vehicle accelerations. This is achieved by increasing the longitudinal friction utilization on the front and rear axle, as illustrated in Figure 1. This should ideally be achieved without disturbing the vehicle motion or making the vehicle unstable. The front axle propulsion torque is increased linearly and a controller tries to keep a constant vehicle velocity by requesting two hydraulic brake pressures for the rear friction brakes. This control results in generation of large longitudinal tyre forces acting in opposite directions on the front and the rear axle. The rear brake controller is a straightforward controller of the rotational velocity of the front axle where the error is defined as:

$$e(k) = 0.5(\omega_1(k)R_e1 + \omega_2(k)R_e2) - v_{x,\text{set}}$$  \hspace{1cm} (1)

where $v_{x,\text{set}}$ is the vehicle velocity at the start of the intervention $\omega_1$, $\omega_2$ are the front wheel speeds and $R_e1$, $R_e2$ are the rolling radiiuse of the front wheels. Based on the error $e(k)$, the brake torque on the rear axle is given by:

$$T_{BR} = K_p e(k) + K_i \sum_{k=1}^n e(k) \Delta t + K_d (e(k) - e(k-1))$$  \hspace{1cm} (2)

The longitudinal velocity of the vehicle is required with high accuracy when calculating the slip ratios of the tyres. Both the front and the rear axle have large longitudinal slip ratios during the intervention. It is hence not straightforward to find the longitudinal velocity of the vehicle without measuring it directly. A method is therefore proposed to estimate the rear tyre characteristics and hence to identify the maximum tyre-road friction coefficient of the rear axle, without knowing the longitudinal velocity of the vehicle, at straight ahead driving.

If the vehicle has significantly more weight on the front axle compared to the rear axle e.g. 60% weight on the front axle, the front tyres will have less friction utilization compared to the rear tyres for the same longitudinal force. The front tyres can then be assumed to be well described by linear relations between the slip ratio and the force even when the rear tyres exhibit high friction utilizations levels. The error of this assumption will increase for higher front axle friction utilization and smaller differences between the front and rear vertical load.

Using this assumption, the velocity of the vehicle can be expressed in terms of the front axle slip ratio as shown in Equation 3 & 4. Equation 4 is derived using Equation 3, the fact that the front tyres can be considered to be in the linear range and assuming that the longitudinal tyre stiffness is proportional to the vertical load (Pacejka, 2012).

$$\sigma_{xF} = \frac{v_x - R_F \omega_F}{R_F \omega_F} \Rightarrow v_x = R_F \omega_F (1 + \sigma_{xF})$$  \hspace{1cm} (3)

$$\sigma_{xF} = -\frac{F_{xF}}{C_F} = -\frac{F_{xF} F_{xR}}{F_{xF} C_R}$$  \hspace{1cm} (4)

With the rear tyre slip ratio as defined in Equation 5, and with Equation 3 & 4, the rear slip ratio can be written as in Equation 6. In Equation 6 the terms $\sigma_{xR}$ and $F'_{xF}$ are introduced to make the subsequent equations easier to read.

$$\sigma_{xR} = \frac{v_x - R_R \omega_R}{R_R \omega_R}$$  \hspace{1cm} (5)
Using a simple brush model (Svendenius, 2007), Equation 7, together with the slip ratio definition from Equation 6, gives a modified brush model with two input variables and two tyre parameters $C_R$ and $\mu$, see Equation 8. The term modified brush model is used recurrently in the text for the brush model in Equation 8. The brush model in Equation 7 is referred to as the original brush model. When the two models are compared the slip ratio used for the original brush model is calculated based on the measured longitudinal velocity. It is important to note that a more advanced tyre model could be used for the front axle characteristics in Equation 4 instead of the proposed linear model. However, that would require further assumptions regarding the relation between the front and rear tyre parameters and could cause problems if the front axle tyre characteristics cannot be well described by the implemented tyre model. A linear model is hence more robust since the effect of errors made in the assumptions are more predictable compared to a more advanced model.

$$\sigma_{XR} = \frac{R_F \omega_F - R_R \omega_R}{R_R \omega_R} + \frac{F_{xR} F_{zR} R_F \omega_F}{F_{zF} C_R R_R \omega_R} = \sigma'_{xR} - \frac{F_{xF} F_{zF}}{C_R}$$  (6)

In the modified brush model, Equation 8, the explicit dependency of the longitudinal velocity has been removed. However, the front and rear longitudinal tyre forces and the vertical forces must still be estimated. The sum of the longitudinal tyre forces on the front axle is estimated using the wheel dynamics equation ($i = 1, 2$ (left, right)) and the estimated propulsion torque from the CAN-bus, denoted $T_{PrP}$. Here $e_{roll}$ is the rolling resistance coefficient.

$$I_{wi} \omega_i = \frac{T_{PrP}}{2} - F_{xi} R_{ei} - F_{xi} e_{roll}$$  (9)

$$F_{xF} = \frac{T_{PrP}}{2 r_{e1}} - \frac{F_{x1} e}{r_{e1}} + \frac{I_{w1} \omega_1}{r_{e1}} - \frac{T_{PrP}}{2 r_{e2}} - \frac{F_{x2} e}{r_{e2}} - \frac{I_{w2} \omega_2}{r_{e2}}$$  (10)

The sum of the rear longitudinal tyre forces are estimated using Equation 10 and the longitudinal equation of motion, Equation 11, which can be rewritten as in Equation 12.

$$m a_x = F_{xF} + F_{x3} + F_{x4} - F_{drag}$$  (11)

$$F_{xR} = F_{x3} + F_{x4} = m a_x - F_{xF} + F_{drag}$$  (12)

where $a_x$ is the measured longitudinal acceleration from the Inertial Measurement Unit and

$$F_{drag} = \frac{1}{2} C_d A v_x^2$$  (13)

The longitudinal velocity $v_x$ in Equation 13 is approximated based on the front wheel speeds since they have smaller slip ratios compared to the rear wheels. The vertical forces on the front and the rear axle are estimated based on the static weight distribution and the longitudinal load transfer, Equation 14, where $h$ is the centre of gravity height and $L$ is the wheelbase.

$$F_{zF} = \frac{m l_f}{L} g - \frac{m a_x h}{L} \text{ and } F_{zR} = \frac{m l_f}{L} g + \frac{m a_x h}{L}$$  (14)
2.1 Influence of rolling radius error on slip ratio measurements

Small errors in the rolling radii of the wheels will affect the slip ratio estimation and hence the measurement points to which the tyre model is fitted. Error terms $\Delta R_R, \Delta R_F$ are added to the rolling radii, $R_R, R_F$ in Equation 6 to study the impact on the estimated slip ratios. With the nominal radii denoted as $R_{F0}, R_{R0}$:

$$
\Delta R_R = \Delta R_F = \frac{R_{F0} \omega_F - R_{R0} \omega_R}{R_{R0} \omega_R} + \frac{\Delta R_F \omega_F - \Delta R_R \omega_R}{(R_{R0} + \Delta R_R) \omega_R}
$$

$$
\Rightarrow \text{ but for } \Delta R_R \ll R_{R0} \& \Delta R_F \ll R_{F0}, \frac{1}{R_{R0} + \Delta R_R} \approx \frac{1}{R_{R0}} \tag{15}
$$

$$
\sigma_{\text{XR}} \approx \frac{R_{F0} \omega_F - R_{R0} \omega_R}{R_{R0} \omega_R} - \frac{F_{xF} F_{xR} R_{F0} \omega_F}{F_{xR} C_R R_{R0} \omega_R \omega_R} + \frac{\Delta R_F \omega_F (1 - \frac{F_{xF} F_{xR}}{F_{xR} C_R}) - \Delta R_R \omega_R}{R_{R0} \omega_R}
$$

Equation 15 shows that a small error in the rolling radii of the tyres leads to an error in the rear slip ratio estimation which is proportional to the front and rear wheel speeds. However, assuming that the intervention is stopped before any wheel has a large slip ratio, the front and rear wheel speeds will almost be equal and the dependency on the front axle force improves the situation further. Equation 15 can hence be approximated as in Equation 16. This slip ratio definition is used in when fitting the tyre model in Equation 8 to the data. Using similar arguments a slip ratio for the rear wheels can be obtained for the case when the reference speed is available (through e.g. GPS measurements), Equation 17. Hence the slip ratio offset could be estimated together with the tyre parameters to include small errors in the rolling radii. The introduction of this term presents a risk of incorrectly adapting offsets in the longitudinal force estimation to fit the tyre model. Hence the offset on the longitudinal force estimation should be minimized.

$$
\sigma_{\text{XR}} \equiv \frac{R_{F0} \omega_F - R_{R0} \omega_R}{R_{R0} \omega_R} - \frac{F_{xF} F_{xR} R_{F0} \omega_F}{F_{xR} C_R R_{R0} \omega_R \omega_R} + \sigma_{\text{XR, offset}} \tag{16}
$$

$$
\sigma_{\text{XR}} = \frac{V_x - R_{R0} \omega_R}{R_{R0} \omega_R} - \frac{\Delta R_R}{R_{R0} \omega_R} = \frac{V_x - R_{R0} \omega_R}{R_{R0} \omega_R} + \sigma_{\text{XR, offset}} \tag{17}
$$

2.2 Test setup

Simulations were performed in order to investigate how errors in the assumption made while deriving the modified brush model affects the estimation of the rear tyre characteristics. The real vehicle tests were performed to evaluate the feasibility of using the proposed method in a real vehicle. The test vehicle was equipped with a RT3000 from Oxford Technical Solutions as a reference to measure the longitudinal velocity of the vehicle. The vehicle used in the experiments is front wheel driven with a turbocharged internal combustion engine and an automatic transmission. The automatic transmission has a lockup function to avoid using the torque converter when a gear has been selected. The gear ratio between the engine and wheels was therefore locked during the interventions. IPG Carmaker was used to investigate how the estimation of the rear axle tyre
Table 1. Vehicle parameters for vehicles used in simulation and in experiments.

<table>
<thead>
<tr>
<th>Vehicle Parameters</th>
<th>Notation [unit]</th>
<th>Test vehicle</th>
<th>Simulation vehicle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass</td>
<td>( m ) [kg]</td>
<td>1700</td>
<td>1940</td>
</tr>
<tr>
<td>Rolling resistance coefficient</td>
<td>( e ) [Nm/N]</td>
<td>0.0038</td>
<td>0.0031</td>
</tr>
<tr>
<td>Wheel inertia per wheel</td>
<td>( I_w ) [kgm²]</td>
<td>2</td>
<td>1.7</td>
</tr>
<tr>
<td>Aerodynamic drag coefficient</td>
<td>( C_{D_A} ) [Ns²m⁻²]</td>
<td>0.792</td>
<td>0.6960</td>
</tr>
<tr>
<td>Wheelbase</td>
<td>( L ) [m]</td>
<td>2.640</td>
<td>2.776</td>
</tr>
<tr>
<td>Centre of gravity height</td>
<td>( h ) [m]</td>
<td>0.5</td>
<td>0.5</td>
</tr>
</tbody>
</table>

characteristics was affected by errors in the front tyre stiffness estimation. The vehicle data for the test vehicle and the vehicle data for the vehicle used in simulations can be seen in table 1. The values in table 1 are for an empty vehicle without any passengers. During the experiments the test vehicle had a total mass of 1900 kg with 56.6% of the weight on the front axle. The simulation vehicle had a total mass of 1940 kg during the simulations and 61% of the weight on front axle. The standard on-board production sensors were used to evaluate the feasibility of the proposed method.

3 SIMULATION RESULTS

Simulations were used to investigate how well the rear axle slip ratio can be approximated based on the wheel speed difference between the front and the rear tyre and the front slip stiffness using Equation 3 and 4. Figure 2 shows the approximated and real slip ratio versus force for the right front tyre and the right rear tyre. As illustrated by the figure, the estimated rear axle slip ratio is close to the true value when the front tyre force is within the linear region of the slip-force curve and the correct front slip stiffness is used. When the front tyre enters the non-linear region the approximated rear axle slip ratio deviates from the true value. The front tyre slip ratio start deviating from the linear model at a utilized friction of 0.4, or 40% of the maximum friction since the maximum tyre-road friction coefficient was set to 1. The rear tyre has a friction utilization of 0.62 for the same longitudinal force. However, the transition point between linear and non-linear tyre behaviour varies depending on the tyre, road surface, tyre pressure and other external factors. The validity of the assumption that the front tyres are operating in the linear range and the effect of the rear axle tyre characteristics is therefore dependent on these factors as well as the weight distribution of the vehicle.

The assumption that the slip stiffness is proportional to the vertical force will introduce small errors in the approximation of the front slip stiffness since the slip stiffness is actually somewhat regressive with increasing vertical load. The assumptions made in Equation 4 will hence lead to an overestimation of the front slip stiffness. The effect of having errors in the estimation of the front slip stiffness on the estimated rear axle slip ratio is shown in Figure 3. The longitudinal speed \( v_x \) was approximated from the front longitudinal tyre force, the front slip stiffness and the front wheel speed as in Equation 4. The rear slip ratio was calculated based on the approximated \( v_x \). If the estimation of the front slip stiffness is too large, the rear slip ratio is overestimated and vice versa. If the front slip stiffness estimation is much lower than the true value the estimated rear tyre slip ratio even has the wrong sign. The slip stiffness varies for different tyres and road surfaces and an online estimation of the front slip stiffness is therefore needed. Other uncertainties such as tyre pressure, different tyres on the front and rear axle may also decrease the accuracy of the front axle slip stiffness estimation.

3.1 Effect of noise on the estimation of the rear tyre characteristics

In order to be able to distinguish between high and low friction at low levels of tyre utilization the estimation of the friction coefficient should not follow the current utilization when the tyres are operating in the linear region. The ideal case was studied to analyse sensitivity to measurement
noise of the proposed method. The ideal case implies that the rear tyre can be represented by the brush model, the front tyre by a linear tyre model, the tyre slip stiffness is proportional to the vertical force and the front and rear tyre force are equal. The maximum normalized force on the rear axle was limited to 0.2 and the maximum friction coefficient was set to 1.15. White Gaussian noise was added to the front and the rear tyre wheel speeds and a nonlinear least square fitting algorithm was used in order to estimate the tyre parameters of one rear tyre. The results with and without noise can be seen in Figure 4. The figures show that the estimation of the rear tyre friction based on the modified brush model converges to low values at low friction utilization on high-friction surfaces when the noise level on the measured wheel speed is high.

This noise is challenging to handle since most commonly used model fitting procedures assumes that the input to the model is correct and noise free. Hence, if the rear tyre force is chosen as the measurement and the slip ratio chosen as an input variable, the fitting will be done assuming that the slip ratio measurement is noise-free. This together with the structure of the estimation formulation when the proposed method is used is most likely causing these problems. When the fitting algorithm is applied to the modified brush model, the tyre parameters will also affect the slip ratio which is fed into the brush model. This means that it has other possibilities to fit the noise that origins from the wheel speeds measurements. When the original brush model is used the slip ratio is measured directly and hence it cannot be altered in the estimation algorithm. The modified model can therefore be considered as more sensitive to measurement noise on the wheel speed signals and road disturbances. For example, consider increased tyre slip stiffness estimation in the modified model. The slip ratio measurement points will move towards larger slip ratio values while the stiffness of the tyre model will increase.
3.2 Effect of tyre dynamics

The longitudinal tyre dynamics can be approximated as a first order differential equation as presented in (Clover and Bernard, 1998), Equation 18. The relaxation length is introduced as a parameter which relates derivative of the actual wheel slip ratio with the difference between the measured nominal value, given by the wheel speed and the vehicle speed \( \sigma_{x, \text{nom}} \), and the actual wheel slip ratio, Equation 18. For a vehicle speed of 13.9 m/s and a relaxation length of 0.1 m in the influence of the tyre relaxation is negligible for the torque ramps done during the interventions. An increase of 1200 Nm/s or approximately 3600 N/s corresponds to a slip ratio gradient of approximately 0.01 \( -/s \) in the linear region of the tyre characteristics with an axle slip stiffness of 350000 N/\( -/ \). This corresponds to a difference of 7e–5 between the real slip ratio and the nominal slip ratio or 1.4% error for a slip ratio of 0.005.

\[
\sigma_x + \frac{v_x}{r_{lx}} \sigma_x = \frac{\sigma_{x, \text{nom}}}{r_{lx}} v_x \tag{18}
\]

4 EXPERIMENTAL RESULTS

A number of different experiments were performed. Accelerations in different gears with a linear increase in the requested propulsion torque were done in order to investigate the accuracy of the propulsion torque estimation available on the CAN-BUS. Several interventions with the proposed method, opposite torques on the front and rear axle, was tested on dry asphalt and on wet basalt with different torque gradients. An offline nonlinear least square fitting algorithm was used to estimate the tyre parameter in the brush model using the data from these interventions. The purpose of the tyre parameter estimation is not to propose an online friction estimator design but to investigate the feasibility of the proposed method.

4.1 Propulsion torque estimation accuracy

The torque estimation of the CAN-BUS is not compensated for the inertia of the driveline. The longitudinal acceleration of the vehicle during the manoeuvre was therefore used as a reference to correct the estimated propulsion torque:

\[
T_a x \approx (m_{ax} + F_{\text{drag}} + m g f) R_e + (4 \frac{m_{ax} I_w}{R_e})
\]

where \( f \) is the rolling resistance coefficient, with the aerodynamic drag force defined as in Equation 13 and where \( I_w \) is the wheel inertia of one wheel. The propulsion torque was compensated with two components, one before the gearbox and one after the gearbox. The propulsion torque from the engine at the wheels can be expressed as:

\[
T_w = T_e G R - I_e \dot{\omega}_e G R - I_{\text{AGB}} \dot{\omega}_w
\]

where \( T_e \) is the engine torque, \( G R \) is the total gear ratio from engine to wheels, \( I_e \) is the inertia of the engine and all rotating part before the gearbox, \( \dot{\omega}_e \) is the angular acceleration of the engine, \( I_{\text{AGB}} \) is the inertia of all rotating components after the gearbox, \( \dot{\omega}_w \) is the wheel angular acceleration of the wheel. The angular acceleration of the engine can be approximated by the angular acceleration of the wheels, assuming that there are no compliances in the driveline:

\[
\dot{\omega}_e = G R \dot{\omega}_w, \text{ hence: } T_w = T_e G R - I_e \dot{\omega}_e G R - I_{\text{AGB}} \dot{\omega}_w = T_e G R - I_e \dot{\omega}_w G R^2 - I_{\text{AGB}} \dot{\omega}_w
\]
Figure 5. Inertia compensation for the propulsion torque estimate for two accelerations in different gears on high-friction.

Figure 6. Wheel speeds during the intervention, left plot with slow torque ramp and right plot with a faster torque ramp.

The inertia $I_{AGB}$ is the combined inertia of the rotating components between the gearbox and the differential and the inertia of the rotating parts after the differential. However, due to the fixed gear ratio in the differential this term can be approximated as one inertia component based on the wheel angular acceleration. The inertia $I_e$ & $I_{AD}$ was estimated by fitting the inertia compensated torque to the torque at the wheels approximated from Equation 20 using test data for two different acceleration in 2nd and 3rd gear. The torque estimation for two accelerations, one in 2nd and one in 3rd gear can be seen in Figure 5. The inertia compensated torque fits the wheel torque calculated from Equation 20 well except for some small deviations. These deviations can be due to error in the propulsion torque estimation from the CAN-BUS, bumps in the road which affect the accelerometer or other losses in the driveline which are not modelled. The overall accuracy of the fit however indicates that the main part of the deviation between the reported propulsion torque on the CAN-BUS and the calculated torque from the acceleration is due to the inertia. The intervention time is therefore not limited by the torque estimation accuracy but rather the maximum torque gradient that can be delivered by the engine. The maximum torque gradient that the test vehicle could deliver was around 1200 Nm/s.

4.2 Disturbing the longitudinal motion of the vehicle

For the test cases used in this study the gains in the brake torque controller was not perfectly tuned. They had been tuned for slower torque ramps and hence the integral part of the controller could not compensate for errors in the proportional gain. For the results presented in this paper the vehicle had a small longitudinal acceleration of around 0.7 m/s$^2$ during the intervention. Figure 6 shows the wheel speeds during two interventions, one for a slower torque ramp and one with a faster torque ramp.

4.3 Estimation of rear axle characteristics

The estimations of the rear axle characteristics, for the highest achieved friction utilization during the friction estimation interventions on high and low-friction surfaces are presented in Figure 7. The maximum friction coefficient was approximated from braking manoeuvres with ABS interventions on the different surfaces as a reference. The real tyre-road friction coefficient might therefore deviate
slightly from the line shown in Figure 7. An offline nonlinear least square fitting is performed using the model described in Equation 8. The identified tyre characteristics are compared to a non-linear least square fitting based on the calculated rear slip ratio and Equation 7 and direct measurements of the longitudinal speed. Both models include an additional term to account for offsets in the rolling radiuses, as described by Equation 16 and 17, which is estimated simultaneously as the tyre parameters.

At these higher levels of excitation both the modified and the original brush model converge to reasonable friction level as seen in Figure 7. However, without sufficient excitation the friction estimate based on the original brush model shows too large values since the tyres have not yet entered the non-linear region of the force-slip curve. The friction estimation based on the modified brush model on the other hand shows too small friction values at lower levels of excitation as shown in section 3.1 and Figure 8.

This is most likely due to the structure of the estimation problem formulation as discussed in section 3.1. The wheel speed signals used in these experiments are of quite low-resolution and the filter that is applied to the signals before they are sent on the CAN-BUS is unknown. A better processing of the signals from the wheel speed sensors is therefore needed before this model can be applied to estimate the friction if the slip ratio is considered as the input variable in the fitting algorithm. At higher tyre utilizations the true curvature of the slip-force curve is visible in the data and the estimates adapt to fit this curvature.

One way to resolve the issue with the disturbances on the wheel speed measurements could be to use an inverted brush model with the force as the input variable and the slip ratio as the measurement. This poses similar problems when estimating the maximum friction coefficient since the maximum measured force, including noise, will determine the minimum estimated friction coefficient of the estimator. A total least square method which minimizes the orthogonal distance between the force-slip curve and the measurement points would therefore be preferable. However, total least square fitting of nonlinear models are most often complex in their implementation and require substantial computational memory and power. It might therefore be more straightforward to reduce the noise on the estimated rear axle longitudinal force instead. The current rear axle tyre force estimation will be affected by all disturbances and noise which affect the longitudinal accelerometer signal, e.g. the pitch angle of the car body relative to the road, and has a higher noise level compared to the front tyre force estimation. Another source for estimation of the rear axle tyre forces is therefore preferable. If the vehicle for instance is fitted with electric motors on the rear axle, the electric motors can be used to apply a negative torque to the rear wheels and an electric motor torque estimate could be used to estimate the rear tyre forces.

It was also found that higher relative excitation levels was needed on basalt compared to asphalt to estimate the friction coefficient reliably, around 90% of the maximum achievable tyre force for the tested tyres with the current noise levels and with measured longitudinal velocity. Figure 8, shows the measured slip ratio versus the estimated force for a test run on basalt up to 80% of the maximum achievable tyre force. The data show no or very small indications of being nonlinear. If the tyres are operating in the linear range it is not possible to estimate the friction coefficient by fitting a non-linear tyre model even if the vehicle velocity is measured.

5 CONCLUSIONS

With sufficient excitation it is possible to estimate the maximum rear axle tyre-road friction coefficient using the proposed method on both high-friction and low friction surfaces. However, the assumption introduced to remove the need to directly measure the longitudinal velocity makes the friction estimation more sensitive to noise on the wheel speed signals. Due to the increased noise sensitivity, the proposed method underestimates the maximum friction coefficient at low tyre utilizations on high friction surfaces with the current noise levels. On the tested low friction surface larger friction utilization, relative to the maximum friction coefficient, is required in order to
Figure 7. Nonlinear least square estimation of $C_R \mu$ with measured longitudinal velocity (BM) and with approximated longitudinal velocity (MBM). Two top plots on dry asphalt, bottom two plots on wet basalt. Two left plots with a torque ramp of 600 Nm/s, upper right plot with a torque ramp of 1200 Nm/s lower right plot with a torque ramp of 1000 Nm/s.

Figure 8. Left plot shows the Nonlinear least square estimation of $C_R \mu$ at different friction utilization on dry asphalt. The right plot shows the measurement points for a test run on basalt up to 80% of the maximum friction.

estimate the maximum friction coefficient compared to the tested high friction surfaces. A slower torque ramp may therefore be required on low friction surfaces to avoid excessive wheel slip.

The noise level on the wheel speed signal and the noise level on the estimated rear tyre forces should be reduced in order to separate between low and high friction surfaces at low tyre utilization using the proposed modified brush model. Future work will focus on how the noise levels can be reduced and how the torque should be added to the rear axle in order to maximize the information about the tyre characteristics to the estimator.

REFERENCES


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