Assessment of Brush Model Based Friction Estimator Using Lateral Vehicle Dynamics

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The tire-road friction potential is an essential variable in vehicle dynamics and control. It can be estimated through a brush model based estimator by measuring the tire lateral force, self-aligning moment, and contact length. However, the deviations between the simplified model and physical mechanism limit its performance in real operating conditions. This paper first extends the TreadSim model to incorporate changes in the physical parameters, such as the inflation pressure and tread wear. Validation of the extended model is carried out by means of a comparison with the on-board vehicle measurement. Then the performance and limitations of the estimator are investigated by utilizing the forces and moments from the TreadSim model simulation. In addition, the sensitivity of the estimator to changes in the physical parameters is evaluated.

1. INTRODUCTION

The maximum tire-road friction potential \( \mu_{\text{potential}} \) has an obvious influence on tire forces and moments. Information on the friction potential would enhance the performance of many active vehicle safety systems, including Adaptive Cruise Control (ACC), Anti-lock Braking System (ABS), Electronic Stability Control (ESC), and collision mitigation systems. In addition, it is a necessity for systems that allow autonomous driving.

Various approaches have been proposed in the literature for the estimation of the tire-road friction potential [1][2][3]. Most estimators utilize measurements such as the longitudinal vehicle dynamics, lateral vehicle dynamics, or tire deformation as physical inputs. For the longitudinal and lateral vehicle dynamics, the forces and moments could be measured through a force hub sensor, power steering torque sensor, or tie rod sensor [3][4]. On the other hand, various tire sensors have been developed to measure tire deformations, such as the tire tread deflection and its acceleration [1][5], which can be further exploited to estimate the friction potential.

Most tire-road friction estimators are derived from different types of tire models. The brush model, as a physical tire model, is able to simulate the mechanism of the tire-road forces and moments of a rolling tire [6]. As it is a relatively simple structure, the friction potential values could be estimated through a set of equations in analytical form [4]. However, the simple structure of such an estimator also brings with it an inherent limitation: the real tire stiffness consists of carcass and tread stiffness, whereas the brush model combines them into one parameter. Thus, e.g., an inflation pressure change (carcass stiffness) or tread wear (tread stiffness) may result in a biased friction potential estimate.

The paper studies the performance of the brush model based estimator using lateral vehicle dynamics. The primary objective of this study is to investigate the sensitivity of the brush model based friction estimator to changes in the physical parameters, such as the inflation pressure and tread wear. An extended TreadSim model which incorporates the inflation pressure and tire wear effects is utilized as a benchmark for the simulation of forces and moments. Outdoor vehicle tests were conducted to validate the extension of the TreadSim model.

2. RESEARCH METHODS

2.1 Brush Model Based Friction Estimator

The brush model based estimator using the lateral vehicle dynamics has been widely used to estimate friction potential [1][5-7]. The estimator can be presented in an explicit and analytic form which only requires the lateral force, self-aligning moment, and contact length as inputs [4]. In this form it is independent of the tire tread stiffness, cornering stiffness, and tire slip angle but it assumes that the behavior of the brush model follows the behavior of a real tire. The friction potential \( \mu_{\text{potential}} \) could be estimated as:
\[ \mu_{\text{potential}} = \frac{F_y/F_z}{(\lambda^2 - 1)} \]  
where \(F_y\) is the lateral force and \(F_z\) is the vertical load. The adhesion coefficient \(\lambda\) can be expressed as:

\[ \lambda = 3x_{pa}^2 - 4/3 \rho + 1/3 \rho (x_{pa} + 3) \]  
where the nominal pneumatic trail \(x_{pa}\) as a function of the alignment \(M_c\) is:

\[ x_{pa} = \frac{x}{\alpha} = \frac{M_c}{aF_y} = -\frac{\lambda^3}{\lambda^2 + \lambda + 1} \]  
and the variable \(\rho\) is:

\[ \rho = (27x_{pa} + 3\sqrt{3}(x_{pa} + 3x_{pa} + 14x_{pa} + 27) + 9x_{pa}^3 + 2x_{pa}^3)^{1/3} \]  
For more details about estimator derivation and implementation, we refer the reader to [4].

2.2 TreadSim Model and Parameters

To assess the estimator performance and operating conditions, the ideal inputs are real forces and moments from on-board measurements. Nevertheless, in the actual vehicle testing, it is difficult to have adequate data with a wide range of excitations such as the slip angle and slip ratio [7]. This will be discussed later, in Section 2.5. Therefore, the TreadSim model was selected here as a benchmark to generate steady-state forces and moments for the evaluation of the brush model based estimator.

The TreadSim model adapts an iterative method to simulate the deformation history of tread elements while they are traveling through the contact patch. A schematic diagram of the TreadSim tire model is illustrated in Fig. 1.

![Fig. 1 A schematic diagram of the TreadSim tire model](image)

The input parameters were determined through experiments and used in the TreadSim tire model to generate the resultant forces, moment, and pneumatic trail and tread deformation.

2.3 Extension of the Model for Inflation Pressure

The inflation pressure has a great influence on tire behavior, such as the tire stiffness, cornering stiffness, and friction forces. However, the TreadSim model does not directly include the effect of the inflation pressure. Typically, the input parameters of the model are measured at a standard pressure and there is no direct pressure-related variable input within the model. In order to extend the TreadSim model to cover changes in inflation pressure, its parameters that are relevant to this effect are determined by experiments.

The contact patch length, which is directly affected by the inflation pressure, is measured on pressure-sensitive films under a static load, as shown in Fig. 2. The contact patch dimension is represented by the length \(2a\) and width \(2b\). According to [9] [10], the contact patch length does not change significantly from a static loaded tire to a rolling one. Thus, the static measurements are used directly as an input in this study. A tire sensor-based approach for the rolling tire contact patch measurement can be found in [11].

![Fig. 2 An example of contact patch measurement on pressure-sensitive film (new tire, 2.4 bars)](image)

Measurements were made with both a worn tire and new tire under three different pressures (2.0 bars, 2.4 bars, and 2.8 bars). Both tires were passenger car tires of the same type (205/55 R16), which are also used for the on-board vehicle tests in Section 2.5. The tread thicknesses for the worn tire and new tire are 2 mm and 7 mm, respectively. It is shown clearly in Table 1 that the worn tire has a shorter contact length, wider contact width, and smaller contact area than the new tire under the same pressure.

<table>
<thead>
<tr>
<th>Pressure (bars)</th>
<th>2a (cm)</th>
<th>2b (cm)</th>
<th>area (cm²)</th>
<th>2a (cm)</th>
<th>2b (cm)</th>
<th>area (cm²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.0</td>
<td>11.0</td>
<td>16.5</td>
<td>181.5</td>
<td>12.0</td>
<td>15.8</td>
<td>189.6</td>
</tr>
<tr>
<td>2.4</td>
<td>9.9</td>
<td>16.1</td>
<td>159.3</td>
<td>11.1</td>
<td>15.6</td>
<td>173.1</td>
</tr>
<tr>
<td>2.8</td>
<td>9.0</td>
<td>15.6</td>
<td>140.4</td>
<td>10.4</td>
<td>15.3</td>
<td>159.1</td>
</tr>
</tbody>
</table>

In the TreadSim model, the carcass stiffness refers to the sum of two parts: the sidewall stiffness and the stiffness introduced by the pressurized air. According to the measurement [12], the carcass stiffness is linearly dependent on the inflation pressure. As a result of the lack of a proper test rig for measurements, the empirical linear relationships given in [8] are used in this study as follows:

\[ K_{cL} = 28500P + 71850 \]  
\[ K_{cT} = -1038.5P + 9618.8 \]
where $K_{CL}$ and $K_{CT}$ are the lateral stiffness and torsional stiffness of the carcass, respectively. $P$ is the inflation pressure in bars.

With the aforementioned extensions, the results of the simulation from the extended TreadSim model for the lateral force and self-aligning moment at different inflation pressures are depicted in Fig. 3.

Fig. 3 The influence of inflation pressure on lateral force and aligning moment

2.4 Extension of the Model for Tread Wear

The tread wear as a result of tire-road interactions plays a significant role in tire performance. It is well known that a worn tire is stiffer than a new tire during cornering. In other words, for a worn tire, a smaller slip angle is required to produce the same lateral force as for a new tire. To address the changes in the model for tread wear, the contact patch length and tread stiffness are modified as wear status-dependent variables.

In addition, it has been widely observed that a new tire has a better grip performance than a worn tire. While driving on an identical road surface, the contact interactions with a new tire produce higher friction forces than with a worn tire. The reasons for such differences are complex and include the changes in, e.g., the rubber properties and contact pressures. To simplify the model, only the static friction coefficient in the TreadSim model is modified to meet this observation. The simulation results from the extended model for the lateral force and aligning moment at different wear statuses are shown in Fig. 4.

Fig. 4 The influence of tread wear on lateral force and aligning moment

2.5 On-board Measurements

To verify the extended TreadSim model for the inflation pressure and tread thickness changes, on-board vehicle tests were carried out to characterize the lateral dynamic. Within the test, a Volkswagen Golf Variant V was instrumented with an MTS Spinning Wheel Integrated Force Transducer (SWIFT) sensor which is able to measure the tire forces and moments along the three axes. Moreover, a 2-axis optical velocity sensor (Correvit S350) was used for the velocity and tire slip angle measurements (Fig. 5). Both the SWIFT and Correvit sensors are mounted on the right front wheel. In addition, the relevant CAN bus information from the vehicle was also logged.

Fig. 5 The instrumented vehicle for tests

The skid pad tests were performed at the driver training test track at Vantaa, Finland. With a gradually increasing tire slip angle, the car driving on a circular track is slowly and constantly accelerated up to the lateral acceleration limit. In addition, the car was driven in a clockwise direction.

The recorded forces, moments, and slip angles are further filtered with a low-pass filter and approximated
with polynomial functions. The measured lateral force is normalized by the normal force, as the wheel load of the inner front tire decreased when the acceleration increased. The comparisons of the normalized lateral force against different slip angles for both the new and worn tire under three different pressures are illustrated in Fig. 6. Most of the curves are different from the simulation results from TreadSim. The reasons are twofold: first, a combined slip exists during the measurement as the traction force is nonzero, and second, the slip angle measurement obtained with Correvit is based on the detection of texture changes. However, the low-friction surface was covered with water and results in less accurate and noisy slip angle measurements.

For the inflation pressure effect, as the differences are small, the changes are difficult to observe. However, existing measurements on the tire test trailer of TNO Automotive are provided in [12]. The proposed extension of the TreadSim model for tire pressure changes qualitatively matches the measured tire characteristic for the pure lateral slip.

Regarding the wear effect, the differences which are clearly observed in the measurement are twofold: for the partial sliding zone, the worn tire has a greater cornering stiffness, and for the full sliding zone, the worn tire provides less grip. Similar observations and measurements can be found in [13]. The proposed extension of TreadSim demonstrates a rather good ability to represent the tread wear effect in accordance with measurements.

3. OPERATING CONDITIONS OF THE ESTIMATOR

The simulated forces and moments from the TreadSim model are used as inputs for the brush model based friction estimator. The measured contact lengths are also used as inputs for the estimator. Both the friction coefficient input used in the TreadSim $\mu_{\text{act}}$, which is velocity-dependent, and the friction estimation $\mu_{\text{est}}$ that is estimated by the brush model-based estimator are shown in Fig. 7.

![Fig. 7 Friction estimations $\mu_{\text{est}}$ of the brush model based estimator](image)

For a better comparison, the estimation error $\mu_{\text{error}}$ is defined as:

$$\mu_{\text{error}} = \mu_{\text{act}} - \mu_{\text{est}}$$

As shown in Fig. 7, the estimation error attains its largest values at a small slip angle, then starts to converge rapidly and becomes stabilized by the time the slip angle has reached around 2 degrees. Moreover, for the small slip angle region, the estimator gives an underestimation of the friction potential; however, for the large slip angle zone, it gives a slight overestimation. In general, the brush model based friction estimator produce a plausible performance.

The estimator error is mainly due to the simplification of a tire in the brush model in this study. For small slip angles up to 2 degrees, the estimator gives a larger estimation error. This is mainly due to the assumption of a symmetric and parabolic pressure distribution used in the brush model. However, the actual normal pressure distribution for a rolling tire has an asymmetric shape and it is larger at the leading edge than the trailing edge, which gives a positive rolling resistance moment. According to Equation 3, when the slip angle equals zero, the adhesion coefficient $\lambda$ equals 1 and thus the pneumatic trail attains its largest value, $a/3$, which is not realistic in measurements. The largest values for pneumatic trails vary and depend on the actual contact pressure distribution. To reduce the estimation error, Equation (3) needs to be updated on the basis of the actual contact pressure distribution. In addition, the simplification of the tire carcass stiffness and tread stiffness in the brush model also partly accounts for the estimation error.

The estimator gives an overestimation error for the full sliding zone, where the slip angle is larger than 10 degrees. This is also due to the simplification of the pressure distribution in the brush model. For the full sliding zone, the resultant lateral force in the
measurement can act in the leading half of the contact patch, which results in a negative self-aligning moment. However, as the pneumatic trail equals zero during full sliding, the brush model provides a self-aligning moment with a zero value.

Fig. 8 The estimation error friction potential at different slip angle excitations

4. PARAMETER SENSITIVITY ANALYSIS

In Section 2, the TreadSim model was extended for changes in the inflation pressure and tread wear. Fig. 9 depicts the estimation error of the friction potential at different inflation pressures. The influence of the inflation pressure changes on the estimation error is twofold: for the small slip angle region, a lower inflation pressure gives a better estimation and vice versa, and for the larger slip angle region, the observation has been made that a higher inflation pressure gives a better estimation. The stiff carcass assumption within the brush model might account for such differences, as the high-pressure tire has higher carcass stiffness.

For the effect of tread wear, the estimation errors of the friction potential for a new and worn tire are illustrated in Fig. 10. Compared with the new tire, the worn tire gives a smaller estimation error of the friction potential. Probably, the stiff carcass assumption is again the main reason for such disparities. Hence, to obtain a better estimation of changes in the inflation pressure and tread wear, an advanced brush model with a flexible carcass and measured contact pressure distribution can be utilized.

Fig. 9 Estimations errors of friction potential at different pressures (the $\mu_e < 0.1$ area is represented in yellow)

Fig. 10 Estimation errors of friction potential for new and worn tires (the $\mu_e > 0.1$ area is represented in yellow)

6. CONCLUSIONS

This study investigated the effect of operating conditions on the brush model-based friction estimator. The assessments were carried out on the basis of the forces and moments simulated with the TreadSim model, which has been extended to incorporate inflation pressure changes. The results show that the estimator is able to work when the slip angle is larger than 2 degrees. Thus, the estimator could be disabled for the low slip angle range, where many sources of errors, such as conicity and ply steer, are dominant. In addition, the parameter sensitivity analysis also implies the possible errors introduced by changes in the inflation pressure and tread wear, which are significantly large in the small slip angle zone and small in the larger slip angle zone.

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