PREDICTION OF TIRE GRIP

A new method for measurement of rubber friction
under laboratory conditions

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To my beautiful mother,
The fact that I am missing you every single day, doesn't get old by the time!
This is for you and because of you!

تقدیم به مادر مهربانم
دل من تشنه یک لحظه دیدار تو است

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1 Introduction

The ingenious invention of the wheel had a major impact on human lives. It tremendously improved the lifestyle of ancients because of the low resistance to motion compared to dragging sleighs with high friction to the ground. This is due to a simple physical fact that the relative speed of the wheel contact point to the ground is zero, considering a planar movement of the wheel without slipping on a linear road. It is not so long ago that tires with the current look came into use. The very first built cars had wooden wheels covered with leather [1]. The abrasion resistance of these tires was poor and they lasted for just a limited amount of kilometers. Probably excellent rolling resistance and skid resistance of these tires were not so much an issue on the dirt/mud roads of those times. Figure 1-1 shows the wheel revolution over the years and Figure 1-2 the first automobiles designed by Henry Ford.

Figure 1-1: Wheel evolution in years, 8000 BC, 3500 BC, 2000 BC, 1889, and 1946 from left to right, respectively [2].

Figure 1-2: (left) Ford Quadricycle manufactured by Henry Ford in 1896-1901, (right) Ford model A designed by Henry Ford and manufactured in 1903-1904 [3-5].

Improved road conditions and increasing driving speeds created the conditions for a major breakthrough at that time: a new material called “rubber” was developed and used for solid tires. Because of its viscoelastic nature, rubber manages to adjust its shape to the counter-surface and deform itself in the contact area to create forwards and sideways forces, which allows steering the vehicle in the wanted direction, see Figure 1-3. Rubber is the only material able to perform like this. And since then rubber has remained in use for tires and may stay so for the future to come.

Figure 1-3: Tire wringing by side force, F_y ; for the sake of understanding the sketches are exaggerated [6].

A tire is a construct of a multitude of elements mostly rubber compounds, each contributing to the overall performance profile. This complex construction interfaces with the road as a counter-surface with a special structure and surface-roughness under diverse operating conditions. Therefore, a wide variety of factors influences the interaction between the tire and the counter-surface, commonly called ‘tire grip’, or ‘traction’, or ‘skid resistance’. Tire Grip is a determining factor in transmitting all the forces and moments originated from the vehicle to the road because tires are the only contact of a vehicle with the ground. Together with Rolling Resistance and Abrasion Resistance, it constitutes the so-called “Magic Triangle of Tire Technology”, which aims to indicate that the three are interrelated. Tire grip is the key consideration for driving safety. Sufficient grip or friction between the tire and the road is one of the most critical factors in reducing vehicle-related accidents. Proper tire grip or friction thus provides a good level of handling which is a prerequisite for a vehicle’s steering, braking, accelerating, and cornering. Tires contribute largely to transportation, environment, handling, and safety, see Figure 1-4.

Figure 1-4: Overview of the contribution of tires, vehicle, and road on the environment, handling, safety, and comfort, wherein friction plays a major role; redrawn based on [7].

For material development purposes for the acquisition of superior tire performance, full-scale tire tests are enormously sophisticated, time-consuming, and costly [8]. It is more sustainable and highly desirable to predict tire performance in a laboratory environment before manufacturing a full tire, only with the determination of frictional properties of the rubber materials used in the tire. To consider all the different conditions a tire may encounter during service in one laboratory apparatus seems an impossible task, particularly when it comes to the prediction of the tribological properties of a tire under highly advanced braking conditions such as the presently common Anti-lock Braking System (ABS).

The rubber material is the eventual contact of a full tire with the ground and, still, there is a continuous drive to find methods that allow better simulation of rubber friction in early development stages in a laboratory environment. Many different test configurations have been developed over time with varying success. Among all available tribometers suitable for assessing rubber frictional properties, measurements with a solid rubber wheel are most promising [9-26]. The big advantage is the rolling motion of the rubber specimen as analogous to a tire. Using sliding body measurements does not provide a realistic input for tire modeling [9]. The Laboratory Abrasion Tester (LAT100) is used for the prediction of tire performance making use of a rolling rubber

wheel known as the Grosch wheel, named after the original developer of the equipment: Dr. Karl Alfred Grosch. The equipment is designed to simulate a wide range of tire service conditions to measure abrasion loss and the frictional forces as a function of various tire-road service conditions such as slip, load, and speed on different counter-surfaces. Furthermore, the evaluated friction properties of the tire tread compounds can thus be employed as input data for models and simulations for tire characteristics. The quality and parametrization of the tire models and simulations strongly depend on the correct input of rubber friction properties [9, 10, 12, 13, 25].

The LAT100 is a laboratory apparatus. The first purpose for which it was developed was to measure rubber abrasion; therefore, its designation is “Laboratory Abrasion Tester”. Gradually it was realized that the apparatus could also be used as a tribometer to characterize friction properties. Consequently, there is a steadily increasing interest to develop this concept further, which was the main purpose of the research presented in this thesis.

1.1. Aim of this thesis

In previous studies by Grosch and Heinz [19], it has been demonstrated that the wet grip, abrasion, and rolling resistance can be characterized by the LAT100, provided the right conditions for testing are employed. The question has been raised whether testing of various tire compounds under **dry** conditions could also be employed to predict tire frictional properties in longitudinal as well as lateral conditions. This was taken as the original objective of the present work, to include different tire tread formulations, as well as different counter surfaces. The results of LAT100 tests were to be correlated with actual tire tests on standardized roads, under ABS-conditions, and different test modalities for the tires on the real roads.

Later on in the project, a new test configuration was designed and further developed by introducing a dynamic interface to the LAT100. This was set as an additional objective to understand the machine as a tribometer because it already provided tire operating conditions with a wheel-shaped specimen. This similarity provides a high potential for mimicking tire grip in a laboratory environment. Therefore, to predict tire grip with the evaluation of rubber friction using LAT100, the thesis is structured as explained in the next section. Therewith, two main characteristic friction components: adhesion between the rubber and the counter-surface and deformation related to the viscoelastic nature of the rubber, can be attributed to the tire or wheel specimen rolling friction.

The development outlined in the present thesis was based on real tire testing in comparison with LAT100 experiments. In the course of the project, three types of real tire tests were executed with different tire sizes and tread compositions. The tire tests were to evaluate dry grip in three different test modalities: ABS braking distance, the lateral and longitudinal test modalities so-called α -sweep, and κ -sweep, respectively. The LAT100 measurements were to be performed in two test modes, locked wheel as a sliding friction tester and sweep test with a rolling wheel. The actual tire data and compound formulations were considered proprietary by Apollo Tyres. However, the point was to compare and rank the tire data and laboratory results to be similar. Thus, the ratings provided by normalized tire data are still comparable with the absolute values and the results of correlations remain the same. Therefore, the chapters were designed as stand-alone texts based on the wide range of tire data, types, and test modalities.

1.2. Thesis outline

The outline of the present thesis is built on six main pillars, in which a strong cross-disciplinary approach for the research is being employed. *Chapter 2* is a survey of the literature on friction and tire grip concepts and a short introduction to the Persson's contact theory. Also, the available laboratory tribometers and tire testing methods are introduced and compared. *Chapter 3* presents mainly preliminary studies on the LAT100 to discover the apparatus possibilities and limitations to be exploited for dry tire friction evaluations. In this chapter, the frictional forces of four different tread compounds are determined by collecting individual measurement points at defined test conditions. The results are correlated with tire ABS braking ratings. In *Chapter 4*, the LAT100 is further employed as a sliding friction tester with blocked wheels for the prediction of dry ABS braking distance by deriving a mathematical model using Design of Experiments for the input factors. The mutual interactions of factors on tire friction are quantified, where the effects of factors alone are not adequate. In *Chapter 5*, the LAT100 is re-designed and a new dynamic interface is developed by which it is possible to perform and mimic tire test modalities in a laboratory environment for a small rubber wheel. Comprehensive tire testing is performed to provide sufficient data for the method validation. Strong correlations are obtained between LAT100 results and the lateral tire test modality called α -sweep, but not for the longitudinal or so-called κ -sweep. In *Chapters 6 and 7*, counter-surface and contact area are analyzed using power spectral density and Persson's friction theory to obtain a better explanation for the acquired correlations. And finally, *Chapter 8* is dedicated to understanding the dynamics and physics of a rolling rubber wheel on a counter-surface disc of the LAT100

test set-up utilizing the renowned physical “brush model” for tires. Therewith, the behavior of the rolling wheel in lateral and longitudinal movements is investigated in comparison to a real tire. A better comprehension of the intricate differences between the α and κ sweep test configurations is obtained in association with rubber friction components.

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2 Literature study

2.1. Tribology definition

Tribology is the “λογος” (logos) or science of “τρίβω” (tribo) or slide/rub. An exact translation defines tribology as the study of sliding or rubbing. The modern and broadest meaning is the study of friction, lubrication, and wear [1].

Tribology of elastomers can be defined as "the science and technology for investigating the regularities of the emergence, change and development of various tribological phenomena in rubber and rubber-like materials and their tribological applications". Tribological phenomena concern the combination of interactions between surfaces in relative motion and the environment, including all of the following interactions: mechanical, physical, chemical, thermo-chemical, mechano-chemical, and tribo-chemical [2].

2.2. Friction definition and mechanism

Friction can be simply described as: “We need to overcome friction to move one material against another, a common phenomenon in our everyday world. Nails hold because of friction. We couldn’t walk or even crawl without friction [3]”.

Probably Leonardo da Vinci (1452-1519) was the first who developed the basic concepts of friction and Amontons (1699) was inspired by his drawings and sketches. “Amontons’ law” showed the proportionality between the frictional force and the normal force [4]. A mathematical form of the friction law was described by Charles Augustin Coulomb who conducted a set of experiments to analyze the magnitude of the Coefficient Of Friction (COF) during sliding as follows:

$$F = \mu F_N \quad (2.1)$$

where F is the friction force (N), COF is often symbolized by the dimensionless Greek letter μ , and F_N the normal force (N). The classic laws of friction are summarized as follows [4-7]:

- The static coefficient is greater than or equal to the kinetic coefficient;
- There is a proportionality between the friction force and the load;
- The COF is not dependent on the apparent contact area nor the sliding speed.

Figure 2-1: a) Friction resistance and forces, b, c) static and kinetic friction drawn according to classic friction laws [4-7].

Figure 2-1 represents a simplified view of friction laws. By applying force, the friction resistance in the interface between the two surfaces reaches a threshold and then the object begins to move. Static friction occurs between two surfaces when they are not moving relative to each other which prevents the object from sliding. The kinetic COF which is also termed as dynamic or sliding friction occurs when the object has a relative movement on the surface. The strength of the classic laws is their simplicity and versatility which served as a guide for investigators and reflected remarkable insight into the mechanisms of friction for many years. However, the Amontons-Coulomb approximation is just a rule-of-thumb of describing a complicated physical interface. The basic law is an empirical construct and is not entirely true for all systems, especially not for elastomers and rubber-like materials; when the surfaces are conjoined, Coulomb friction becomes a poor approximation. The static COF often denoted as μ_s can be even smaller than the kinetic coefficient of friction μ_k [8]. The relation between μ_s and μ_k requires understanding of fundamental physics behind the friction phenomenon in the interface which can be described by chemical bonding between two surfaces or interlocking due to the asperities and irregularities of the surfaces. The friction of rubber materials depends on load, the real area of contact, speed, and temperature [9-12]. New models present more detail about friction especially of rubber materials for instance developed by Grosch, Savkoor, Kane, and Persson [13-16]. In Section 2.3.3, the Persson friction model will be elaborated in more depth.

The basic mechanism of friction in dry relative motion can be attributed to two main contributions. The first is the adhesion force, which occurs in the region of the real contact area. The second term may be described as a deformation component. By assuming no interaction between the two factors, the total friction force (F) may be written as [5]:

$$F = F_{adhesion} + F_{deformation} \quad (2.2)$$

Dividing each component of this equation by the load N , the corresponding equation is written in terms of friction coefficients as follows:

$$\mu = \mu_A + \mu_D \quad (2.3)$$

where the suffixes A and D denote the adhesion and deformation terms, respectively [5, 6]. Moore [5] described that by choosing proper experimental conditions, the contribution of either the adhesion or the deformation coefficient is negligible. For instance, the dominant mechanism on an optically smooth surface is purely adhesion. Alternatively, the measured friction force can be attributed solely to the deformation component on lubricated rough surfaces. He also mentioned that in the normal case of

dry sliding between rough surfaces, the coefficient of adhesional friction is generally at least twice as high as the deformation contribution [5]. However, the contribution of the attributed friction mechanism is still debatable.

2.3. Rubber friction

In the contact area between an elastomer and a rough surface in relative motion, the adhesion component of the friction force is referred to as molecular interaction between the rubber and the substrate. The deformation term is due to a delayed recovery of the elastomer after indentation by a particular asperity, which is generally termed the *hysteresis* component of friction [5].

Figure 2-2: Principal components of elastomeric friction on a dry surface [5].

Figure 2-2 shows how the total friction force develops in sliding over a single asperity, separated into an adhesion and a deformation part [5]. Therefore:

$$F_{\text{deformation}} = F_{\text{hysteresis}} \quad (2.4)$$

$$F = F_{\text{adhesion}} + F_{\text{hysteresis}} \quad (2.5)$$

$$\mu = \mu_A + \mu_H \quad (2.6)$$

where μ_H designates the term due to hysteresis. Moore [5] also described the adhesion as a surface effect, whereas hysteresis is a bulk phenomenon that depends on the viscoelastic properties of the elastomer.

A disruptive stick-slip process at a molecular level is fundamentally responsible for adhesion and several theories exist to explain the phenomenon. However, both adhesion and hysteresis can be attributed to the viscoelastic properties of rubber at the macroscopic level [5].

2.3.1. Load dependency

In earlier work, Schallamach [11] presented some experimental evidence that the load dependency of rubber friction can be explained by the proportionality of the frictional force to the true area of contact between rubber and track. He deduced the real area of contact and contact pressure by considering elastic deformation of the rubber surface on spherical asperities using a simple model. For soft rubber on smooth surfaces, the real or actual contact area A (m²) increases with the nominal or apparent pressure P (Pa) as follows:

$$A \propto P^{2/3} \quad (2.7)$$

This outcome is in agreement with a theoretical expression derived by Hertz from the classical elasticity theory for spherical contacts, which leads to the following relation for COF:

$$\mu \propto P^{-1/3} \quad (2.8)$$

Schallamach carried out extensive experiments with several hard tire tread compounds on surfaces with various asperity shapes and coarseness and showed that the exponent in Eq. (2.8) changes to the value -1/9; therefore the load dependency may be neglected in the case of rough surfaces [11, 13]. He explained that the true area of contact and frictional force under low load is smaller since the rubber is in contact with the “tallest asperities” and the “smaller asperities” will gradually come into contact with the counter-surface as the load increases [11].

Fujikawa et al. [17] measured an actual contact ratio α , which is defined as the real contact area to an apparent contact area on various surfaces and contact pressures as shown in Figure 2-3. For instance, the actual contact ratio of a passenger tire with a typical apparent contact pressure of 300 kPa to 500 kPa for three kinds of test pavements, is about 0.1; this level shows that only 10 percent of the nominal contact area is the true contact area.

In more recent work, Lang and Klüppel [18] described the relation between the coefficient of friction and normal load as a function of temperature and velocity on a coarse granite surface. They compared their results with the extended friction theory

of Klüppel and Heinrich [19]. The results showed that with increasing nominal pressure the COF decreases as expected, but the dependency of COF vs. speed is different at various temperatures from 2 °C to 100 °C, this will be elaborated in the next section by explaining the friction master curve. They presented a decrease in the ratio of the real area of contact by nominal area vs. speed at various nominal pressures as shown in Figure 2-4.

Figure 2-3: Actual contact ratio vs. apparent contact pressure for three kinds of test pavement [17].

Figure 2-4: The real area of contact/nominal area (A_c/A_0) vs. velocity v (m/s) at various nominal pressure (bar) and 70 °C temperature, taken from ref. [18].

Fortunato et al. [20] presented the origins of non-linear pressure dependency of μ . It is due to frictional heating which softens the rubber and increases the area of contact. In Figure 2-5, they showed the area of real contact between an elastic solid with a Poisson's ratio ν of 0.5 and Young's modulus of E on an asphalt road surface. For pressures up to the order of E the relation between the ratio of the real area of contact to the nominal area (A/A_0) increases linearly with the term p/E , and accordingly the same is expected for the coefficient of friction. In Figure 2-6, based on numerical calculations they obtained the coefficient of friction as a function of low frequency reduced modulus defined $E_{r0} = E_0/(1 - \nu^2)$, where subscript of zero in E denotes low frequency. Results are shown for three different asphalts with various skewness surface values (SK).

To conclude, the COF of elastomers depends on the real contact area owing to the viscoelastic properties of rubber which act in various ways based on the types of asperities of the counter-surface or the surface roughness. For smooth surfaces, Table 2-1: shows a good summary of a very general idea about the load dependency of soft rubber.

Figure 2-5: The non-linear pressure dependency of the real area of contact/nominal area (A/A_0) of an elastic solid with Young’s modulus E and Poisson’s ratio $\nu = 0.5$, and a rigid asphalt road surface, taken from ref. [20].

Figure 2-6: The viscoelastic contribution of friction coefficient as a function of nominal pressure divided by low frequency reduced modulus $E_{r0} = E_0/(1 - \nu^2)$ at the sliding speed $v_0 = 2.336 \text{ m/s}$ for three different asphalts with various skewness surface values (SK) [20].

Table 2-1: Comparison of load dependency of solid material with a rubber-like material, based on [1].

Plastically deformed by asperities		Elastically deformed by asperities
$A \propto N$	← Effect of load on the real contact area →	$A \propto N^{2/3}$
$\mu = \text{const.}$	← Coefficient of friction →	$\mu \propto N^{-1/3}$

2.3.2. Temperature and speed dependency

Rubber friction is strongly dependent on speed and temperature, which is dominated by viscoelastic properties. An equivalence exists between the effects of temperature and speed, which was first explained by Williams, Landel, and Ferry (WLF) [21] based on classical work of Boltzmann; The famous WLF time-temperature superposition demonstrated this equivalence for viscoelastic properties of practically all pure polymers by applying a universal function. Pioneer work of Grosch showed that this transformation equation applies to rubber friction for both gum polymer (unreinforced) and filled rubber compounds [10, 22]. The COF was measured at various temperatures over a range of velocities; by considering that the highest speed is low enough to neglect the temperature rise during the measurement. By plotting the COF versus $a_T v$ in a logarithmic scale, where v is velocity and a_T is the shift factor as a function of temperature given by the WLF equation [22, 23], a “master curve” is generated. The master curve is a single curve of COF from many segments as shown for an Acrylate-Butadiene Rubber (ABR) gum compound as an example in Figure 2-7. The type of rubber compounds and track surfaces affect the shape of the master curve and its position on the $\log a_T v$ axis as it can influence the rubber dynamic and mechanical properties.

Figure 2-7: (left) Experimental coefficient of friction on the y-axes as a function of $\log(\text{speed})$ on the x-axes at different temperatures for an Acrylate-Butadiene Rubber gum compound on a clean dry silicon carbide 180 track surface; (right) resulting master curve, referred to room temperature, taken from the Goodyear medal paper of dr. Grosch [22].

2.3.3. Persson friction theory

The theoretical friction theories calculate and explain the real area of contact on micro-scale, how the rubber follows the counter-surface asperities, and to what extent the surface “cavities” are filled out by rubber. Persson’s theory predicts the real area of contact and the coefficient of kinetic friction by attributing this to the rubber viscoelastic modulus $E(\omega)$ with ω the excitation frequency and power spectra of a randomly hard substrate $C(q)$; at a speed low enough – typically smaller than 1 mm/s – that frictional heating can be ignored.

Persson defines a magnification parameter $\zeta = L/\lambda$ or q/q_0 where λ is the wavelength between the asperities in the counter-surface, L the length of contact, $q = 2\pi/\lambda$ is the magnitude of the wavevector or spatial frequency of the asperities with q_0 as the smallest relevant wavevector (see chapter 6). He describes the nominal area of the surface as L^2 , claimed to be of order of λ^2 where $h/\lambda \ll 1$, see Figure 2-8 with h the amplitude of the asperity. Theoretically, he shows for an elastic body and no plastic deformation if $\zeta \rightarrow \infty$ then $A(\lambda) \rightarrow 0$ where $A(\lambda)$ is considered to be the area of real contact on the length scale λ . In such a situation the rubber excitation frequency is well

in the glassy range. Qualitatively, Persson discusses that for an elastic rubber if h/λ is $\approx \sigma_0/E$, then σ_0 will be just large enough to consider that the rubber fills out all the cavities; where the mean perpendicular pressure is σ_0 i.e. the normal load divided by the nominal contact area A_0 . As an example, he estimates for a filler reinforced rubber with $E=10$ MPa and σ_0 of 0.2 MPa as average pressure on a passenger car tire, a partial fill-out of the counter-surface cavities by the rubber [15]. The theory defines the (projected) contact area between the rubber and the surface with introducing the $P(\zeta) = A(\lambda)/A_0$ or $P(q) = A(q)/A_0$ which is the relative area at magnification ζ to the nominal one. $P(q)$ is the error function (*erf*) of $1/2G(q)$ as follows:

$$P(q) = \text{erf} \left(\frac{1}{2G(q)} \right) = \frac{2}{\pi} \int_0^\infty dx \frac{\sin x}{x} e^{-x^2 G(q)} \quad (2.9)$$

where

$$G(q) = \frac{1}{8} \int_{q_0}^q dq q^3(q) \int_0^{2\pi} d\varphi \left| \frac{E(qV \cos \varphi)}{(1 - \nu^2)\sigma_0} \right| \quad (2.10)$$

$E(\omega) = E(qV \cos \varphi)$ is the elastic modulus as a function of the pulsating frequency induced by sliding velocity V on asperities in polar coordinates with Poisson ratio of ν (for rubber material 0.5) [15]. By denoting σ_ζ for the average pressure in the length scale of λ in the contact area and $P(\sigma, \zeta)$ for the pressure probability distribution, it can be written [15]:

$$\sigma_0 A_0 = \sigma_\zeta P(\zeta) A_0 \quad (2.11)$$

$$\sigma_\zeta = \int_0^\infty d\sigma \sigma P(\sigma, \zeta) / \int_0^\infty d\sigma P(\sigma, \zeta) \quad (2.12)$$

The function $P(\sigma, \zeta)$ becomes a broader distribution with considering higher magnification ζ which corresponds to shorter length scales. The actual area of contact was underestimated in the original theory [24]. A correction factor $S(q)$ was applied as follows [25]:

$$S(q) = \gamma + (1 - \gamma)P(q)^2 \quad (2.13)$$

by introducing an empirical parameter γ in the range of 0.4-0.5. The theory calculates the viscoelastic μ as follows:

$$\mu \approx \frac{1}{2} \int_{q_0}^{q_1} dq q^3(q) C(q) S(q) P(q) \times \int_0^{2\pi} d\varphi \cos \varphi \left| \frac{E(qV \cos \varphi)}{(1 - \nu^2)\sigma_0} \right| \quad (2.14)$$

$C(q)$ is the spectral density which represents the surface roughness, see Chapter 6.

Figure 2-8: A schematic view of the contact between the rubber and a hard substrate at a different magnification of ζ according to Persson's theory [15].

2.4. Tire grip or traction

Tires are built up of a multitude of different elements such as the tread, sidewall, bead, shoulder, and various plies, which mostly comprise of rubber compounds; each contributing to the overall tire performance profile. Tires support and transmit all types of loads or forces such as vertical, longitudinal braking, driving, cornering forces, and camber thrust. All forces are necessary for the directional control of the vehicle. The best compromise in tire construction regarding carcass stiffness, tread pattern and compound is to engineer the tire in such a way to meet the requirement of the “vehicle suspension”.

Tire grip is a concept that describes the interaction between the tire and the road. A tire with better grip provides e.g. a shorter stopping distance which is the distance that the vehicle needs to come to a full stop. A proper tire grip provides a good level of handling. To describe the tire grip of various tire constructions on the road with a variety of surface textures, the tire-road interface has great importance. Some of the important features of tire behavior at the interface are:

- Normal force distribution
- Actual slip velocity at various points in the contact area
- Shape of the contact area
- Deformation of the tread blocks at a range of speeds as a function of tire geometry.

Tire grip or traction is generated by the frictional forces which are created by the relative movement of these tread blocks on the ground. This relative movement is known as the slippage which occurs only during cornering, braking, and accelerating when the tread block has a displacement; otherwise, the relative speed of the tire in the contact line would be zero. To achieve a better understanding of the tire grip, it is essential to have a closer look into the tire forces acting during the braking (accelerating) and cornering states and their corresponding slippage [26].

2.4.1. Longitudinal friction force and slip ratio

Figure 2-9 shows a simplified schematic view of tread blocks during braking in the contact area. During every revolution of the tire on the road, each tread block enters the contact area, touches the ground, and leaves the trailing edge until the next revolution. The longitudinal displacement or slippage is shown in Figure 2-9: the alteration in the orientation of the tread blocks in the leading edge which varies from the trailing one. In the front part the tread blocks adhere to the surface, micro slippage is dominant; towards the trailing edge macro slippage plays a role. By increasing the tire slippage, more portions of the contact patch involve the blocks in the macro displacement. This phenomenon dissipates energy stemming from tread deformation, shear stress in the stationary region, and frequency excitation in the non-stationary region of the rubber material in contact with the road. Because of the limited motion of micro-slippage in the front region of the tire, more static COF μ_s and adhesion contributions are expected to be involved. In each tire revolution and a certain slippage level, every single tread block: 1) experiences a tread deformation and shear stress from the front stationary region; and 2) undergoes a noticeable sliding under a range of frequency excitations which originate from the surface roughness of the ground in the non-stationary region. This friction in a rolling tire is different from Figure 2-1. This means that in ordinary driving, even though the tire is in motion, the major part of the contact patch relative to the ground is stationary. Therefore, both μ_s and μ_k are involved.

Figure 2-9: Tire tread blocks in the contact area in the longitudinal direction.

2.4.2. Longitudinal friction curve

The difference between the longitudinal driving speed V_x (m/s) and the equivalent circumferential velocity of the wheel ($R_e\Omega$) is the longitudinal slip, where Ω (rad/s) is the rotational velocity of the wheel [27, 28]. According to the SAE¹, the longitudinal slip ratio is defined as:

$$\kappa = - \frac{V_x - R_e\Omega}{V_x} \quad (2.15)$$

where R_e (m) is the effective tire radius which can be measured during the experiment. It is defined as the radius of the tire when rolling with no external torque applied to the spin axis. Since the tire flattens in the contact patch, this value lies between the tire's un-deformed radius and static loaded radius [29].

It can be deduced from equation (2.15) that during braking, the slip ratio is negative e.g. in "locked" braking tires, the slip ratio equals (- 1.0) [28]. Conversely, in an accelerating condition, the slip ratio is always positive. Figure 2-10 shows the negative braking slip ratio in the locked wheel situation.

¹ Society of Automotive Engineers

Figure 2-10: Longitudinal force in arbitrary units (a.u.) vs. braking slip ratio, redrawn from [28].

2.4.3. Lateral friction force and slip angle

The lateral (side) force is the force that a tire transmits to the ground during cornering in parallel to the rolling axis of the tire. Figure 2-11 shows the overall lateral force contributed by the four tires at the center of gravity (CG) of the vehicle. At any time of cornering, the path forms an arc. The radius of that arc might be changing, but at any instant, the path describes a specific arc.

Figure 2-11: The overall lateral force of the four tires at the center of gravity (CG) of the vehicle [30].

Every point on the tread notices a regularly repeating vertical force and the tire momentarily bears its share of the vehicle's weight. As soon as the driver turns the steering wheel, the tread pulls the rest of the tire and generates a lateral force that goes through the wheel and the suspension to turn and change the vehicle's path. The tire

tread deforms as it passes the contact patch area. It is the tire's resistance to this deflection that creates the lateral force that turns the vehicle [30].

A tire generates a lateral force with a slip angle as shown in Figure 2-12. The angle between the tire's plane of rotation and the actual velocity direction of the vehicle in cornering is termed slip angle. In other words, the slip angle occurs when the steering wheel is turned from straight-ahead; it is the angle between the direction in which the tire is pointed and the direction in which the vehicle is actually heading ("a" or α in Figure 2-12). It is noteworthy to quote that "the elastic nature of a tire makes the slip angle possible" [30].

Figure 2-12: Slip angle occurrence in steering [30, 31].

2.4.4. Lateral friction curve

Figure 2-13 shows the general relationship between the lateral force and the slip angle. This relationship is a specific characteristic of a tire design that includes carcass and various rubbers used in different parts of the tire structure, especially the rubber compounds in the tire tread. The shape of this curve is unique for each type of tire.

Figure 2-13: Lateral force in arbitrary units (a.u.) vs. slip angle, redrawn from [28].

As shown in Figure 2-13, the curve has two main regions in three distinct shapes:

1. A linear (elastic) region at small slip angles: a rise in slip angle leads to a proportional increase in the lateral force. The slope of this section of the curve is the "cornering stiffness" of the tire which has a linear shape. Higher stiffness in the tread and sidewalls results in a steeper slope. A typical range for a radial tire is 1 kN/degree - 4 kN/degree, with a lower range for passenger car tires and a higher range for specialty racing tires [28].
2. A non-linear region in two different shapes:
 - Transitional region: at larger slip angles, portions of the tread elements in the tire contact patch slide and the rate of increase in the lateral force is less. When the curve reaches the maximum, more tread elements of the contact patch begin to slide and the tire generates less lateral force.
 - Frictional region: after the peak of the curve, the lateral force can be reduced within a few degrees of increasing slip angle. At high slip angles, most of the tread elements in the contact patch are sliding and produce a lot of heat and wear.

The schematics in Figure 2-14 are obtained from visual observations of contact patch shapes at various slip angles. The tire is rolling in the direction of the top of the page and the turning direction is to the left. First, it shows that at higher angles the slip area increases, and simultaneously the adhesive area decreases. Second, it presents the leading edge of the contact patch curves towards the turn. The leading edge of the

contact patch points in the steering direction while the rear portion of the contact patch lags on the old heading direction.

Figure 2-14: Visual observations of contact patch shapes at various slip angles [30].

2.5. Laboratory scale tribometers

Friction and wear are interrelated subjects, simply because friction is involved and plays its part in wear mechanisms. It is possible to study and measure friction and wear in the same experiment. Abrasion or wear tests are quite common in the rubber industry and has been developed and standardized while the friction tests are seldomly standardized [32].

2.5.1. Methods of measuring friction

To measure the frictional force in a friction test, the essential requirements are two contacting surfaces with a relative motion to each other. Figure 2-15 shows various arrangements for friction tests. Based on these basic arrangements, many researchers have tried to develop an apparatus to measure the coefficient of friction [19, 31-40]. The friction behavior has also been calculated using modeling and simulation techniques [41-44].

Figure 2-15: Basic arrangements for friction tests: (a) linear track; (b) rotating shaft; (c) towed sled; (d) pin and rotating disc; (e) inclined plane; N : normal force, V : direction of motion, W : weight of the test piece [32].

One of the phenomena that often occur in rubber friction testing is 'slip-stick', in which the relative velocity and the coefficient of friction between the two surfaces both oscillate around a mean value. The amplitude and frequency of the slip-stick vibrations depend on the rigidity and damping of the testing system as well as on the properties of the surfaces. To minimize slip-stick it is necessary to construct the test apparatus, particularly the drive and force measuring elements, as stiff as possible. The other factors such as lubricants, wear debris, aging of the surfaces, and humidity also affect friction. Therefore, the friction test procedure must be carefully selected in a way that resembles the service conditions.

The Rapra apparatus is a device that measures the coefficient of friction in conjunction with a suitable tensile tester. The advantages of the apparatus are accurate measurements of small forces, a good range of velocities, and a wide range of temperatures. The test piece geometry can be readily changed; tests on products or parts of products are feasible [32].

The linear friction tester (LFT) is a friction measuring device that is designed based on the longitudinal friction principle, for instance Figure 2-15(c). In the longitudinal

friction principle at a 100% slip ratio ($\kappa = 1$), there is no need to use a whole tire as a test object a rubber sample or tire tread block can represent the whole tire friction [26].

Figure 2-16 shows some examples of portable devices such as the “Skid resistance pendulum” and the “Abrollgleiter”. These devices determine the friction coefficient based on the dissipated energy ΔE_d in the applied friction process. Energy ΔE_d can be translated into a certain value that represents the friction coefficient. Both portable devices presented in Figure 2-16, convert potential energy into kinematic energy. For these systems, there is no electrical measurement acquisition, i.e. there is no possibility of monitoring the occurring friction process over time. It requires high precision for adjustment which might lead to failed measurements.

Figure 2-16: Examples of portable devices; skid resistance pendulum (left), Abrollgleiter (middle and right) [31].

There are various demands for friction tests. To measure rubber friction on roads and floor surfaces, it is desirable to measure the coefficient of friction in-situ. As a result, a portable device is necessary. The friction of road surfaces is often measured with a slide tester developed by the Road Research Laboratory which is widely used on various surfaces, for example, floors and artificial sport surfaces [32]. Several reviews of methods and equipment for friction measurements of rubbers have been reported, but the correlation of the results from different lab methods with tire test data is often relatively poor. It is perhaps not surprising considering the variety of geometries and test conditions in use and the complexity of the interaction between surfaces [32].

2.5.2. Friction standard methods

ISO 15113 standard [45] for the frictional properties of rubber was published in 1999 and is probably the most comprehensive one for friction standards. It was developed from a British standard, BS 903 Part A 61 [32]. No ASTM method explicitly describes the determination of rubber friction for a laboratory specimen.

The standard methods do not describe a specific apparatus but explain the importance of tight control of the parameters and indicate remarkable guidance both in the text and annexes on factors to be considered in obtaining friction measurements. Three procedures for determining dynamic friction are given: the initial friction, friction after repeated movement between the surfaces, and friction in the presence of lubricants or contaminants. Some procedures are indicated for preparing the sliding surfaces.

In most test procedures, the objective is to provide the best correlation with service conditions together with good reproducibility between laboratories. Regarding the aforementioned statement that friction and wear are interrelated and test configurations are similar, an overview of different abrasers is summarized in the next section which might be opted for the friction measurements.

2.5.3. Types of abrasion tests

There are several standard methods for measuring abrasion for specific applications:

- ASTM D5963: Rubber property-abrasion resistance (rotary drum abrader) [46];
- ASTM D1630: abrasion resistance-footwear abrader [47];
- DIN ISO 4649:2010 Rubber, vulcanized, or thermoplastic-determination of abrasion resistance using a rotating cylindrical drum device.

ISO 23794 [48] shows a range of abrasion test apparatus which are based on various wear mechanisms. In a real situation, more than one mechanism is usually involved but one may predominate. It is possible to categorize them as follows:

- Abrasive wear: is caused by sharp asperities cutting the rubber. It requires hard, sharp cutting edges and high friction.
- Fatigue wear: is caused by particles of rubber being detached as a result of high dynamic stress during a specific time. It occurs with smooth or rough but blunt surfaces and does not need high friction.
- Adhesive wear: is the transfer of rubber to another surface as a result of adhesive forces between the two surfaces. It is less common but can occur on smooth surfaces.
- Roll formation: is sometimes considered as a separate mechanism. It happens when there is a progressive tearing of layers of rubber and they form a roll. It occurs in high friction and at relatively poor tear strength. Roll formation

results in a characteristic abrasion pattern of ridges and grooves at right angles to the direction of movement (often called Schallamach waves).

- Corrosive wear: is due to a direct chemical attack on the surface.
- Erosive wear: is sometimes used for the action of particles in a liquid stream.

Another distinction between various tests is related to the test specimen geometry and abradant. Some common combinations are shown in Figure 2-17. Abradants can be classified into the following types:

- Abrasive wheels;
- Papers and clothes;
- Metal knives;
- Smooth surfaces;
- Loose abradants.

The proper test method should be selected based on the service conditions of the real application. Every category has its pros and cons, but reproducibility and availability in a convenient form are a necessity.

The operating and test conditions can be summarized as follows:

- Temperature: controlling the temperature of the contact surfaces during the test is very difficult, although the tests are carried out at ambient temperature; it has a huge impact on the correlation between laboratory and service conditions.
- Degree and rate of slip: the relative movement or slip is crucial in determining the wear rate. The higher slip also gives a higher heat generation.
- Contact pressure: under some conditions, if the abrasion mechanism changes, a large rise in temperature occurs dependent on the friction between the surfaces.
- Continuous contact: when the test piece is continuously in contact with the abradant and there is no chance for the generated heat at the contact surfaces to be dissipated.
- Intermittent contact: the contact occurs in a regular interval not continuously.
- Lubricants and contamination: the interface between surfaces is important because any change in the nature of the contact surface creates a big difference in the final results. Two arguments arise regarding the lubricants and contamination; first, deliberately applying another material or media between surfaces to simulate service condition, for instance, lubricants such as water or introduction of particulate material to a surface to simulate tire running on a

wet or dusty road; second, removing wear debris by continuously brushing the test piece or by the use of air jets. It should be ensured that the air supply is not contaminated with oil or water from the compressor [48].

A summary of the types of abrasion apparatus is listed in Table 2-2 according to ISO 23794:2010.

Figure 2-17: Test piece (1), abrasant (2) in several forms [48].

Clogging or smearing of the abrasant surface is a common problem that leads to invalid test results. It is normally caused by the high temperature at the contact surfaces and, although the problem can sometimes be reduced by introducing a powder between the surfaces, it should be treated as an indication that the test conditions are not suitable. If high temperatures are experienced in service, a test method should be chosen in which a new abrasant surface is continually generated [48].

Table 2-2: Summary of the types of abrasion apparatus [48].

Type	Test place	Abradant	Motion	Slip	Speed (m/s)	Contact
Akron	Wheel	Abrasive wheel	Test piece driven	Variable	0.2-0.35	Intermittent
DuPont	Molded or cut from sheet	Abrasive paper	Abrasive paper rotates	100%	0.19-0.3	Continuous
Frick Taber	Flat disc	Wheel+powder	Test piece rotates	100%	0.25	Intermittent
LAT100	Wheel	Abrasive disc	Disc rotates	Variable	0.0005 to 27	Intermittent
Lambourn	Wheel	Abrasive wheel	Test piece/abradant driven	Variable	0.15-3.6	Intermittent
Martindale	Disc	Various	Test piece driven in patterns	100%	Varies	Continuous
NBS	Square cut from sheet	Abrasive paper	Rotating drum	100%	0.3-0.4	Continuous
Pico	Disc	Tungsten knives	Test piece rotates	100%	1.7-5.3	Continuous
DIN	Disc	Abrasive paper	Rotating drum	100%	0.31	Continuous
Mill	Disc	Abrasive particles	Tumbled in drum	NA	NA	Continuous
Schiefer	Disc	Various	Test piece/abradant driven	100%	varies	Continuous
Taber	Flat disc	Various wheel	Disc rotates	100%	0.045	Intermittent

2.6. Tire testing for grip

Indoor and outdoor test facilities are available to assess the performance of a full-scale tire. The type of tests can be categorized based on the friction principles, i.e. longitudinal or lateral as follows [49]:

- κ -sweep (Kappa): longitudinal slippage sweep which is the variation of κ , while maintaining the slip angle α equal to zero. The degree of κ is continuously varied by applying a braking force to the wheel.
- α -sweep (Alpha): lateral slippage sweep in which the α is varied over one complete cycle of slip angle.
- Combined slip: variation of the longitudinal slip κ for non-zero values of the slip angle α [50].

It is usually possible to perform the measurements at various normal loads and camber angles. The camber angle is the angle between the plane of rolling of the wheel and the vertical axis of the vehicle when viewed from the front or rear. The lateral and

vertical forces and self-aligning torques are measured and by employing tire models e.g. the Magic Formula (MF) of Pacejka [43] the data are presented in the typical friction curves as illustrated in Figure 2-10 and Figure 2-13.

2.6.1. Outdoor tire tests

The outdoor tests being employed for measuring tire grip are vehicle-based or trailer tests carried out on real test tracks. A vehicle test is conducted on a commercial car equipped with Anti-lock Braking System (ABS) to evaluate the stopping distance under specified test conditions. The distance that the car reaches from a specified high speed to nearly a full stop is the stopping or braking distance. A test procedure in ASTM F1649 exists for the wet braking traction performance of passenger car tires.

The trailer test installation is built on a truck or hanger. A tire can be mounted on a special wheel suspension equipped to a guidance system to which a measuring hub is attached, see Figure 2-18 [50, 51]. The output is a friction curve based on the applied friction principles. There are dedicated fields and test tracks for carrying out outdoor tire testing, one example is given in Figure 2-19.

Many variables influence the ranking of the tread compounds for the grip properties in outdoor testing. Under certain test conditions like temperature, car type, and tire size, the achieved ranking should be repeatable. However, sometimes the non-reproducibility of grip characterization of tread compounds with another ABS is a limitation for the tests [26].

Figure 2-18: TASS tire test trailer [51].

Figure 2-19: Tire test track at Ladoux, Clermont-Ferrand, Michelin tire company, France [52].

2.6.2. Indoor tire tests

In indoor tire testing, the tire is running on a road simulator or a rolling rig which consists of a drum covered with an artificial surface or replica of a real road. Flat track tire testing machines determine tire performances in an indoor laboratory on a full tire on corundum sandpaper. The design of such systems presents a high degree of complexity to control the flatness of the counter-surface as well as its dynamic performance to maintain the motion of the counter-surface flat and straight interfacing with the tire [31, 53], see Figure 2-20. The measurements have a higher precision in

comparison with tire trailer testing because of the controlled laboratory conditions and environment but are less realistic due to the type of test track [54]. At an actual range of loads and speeds of the tire, the flat track tests are not in good agreement with the trailer testing due to the sandpaper substrate which normally becomes saturated with abraded rubber particles that fill out the asperities of the test track. The possibilities of applying a simulated road in the indoor devices have been extensively discussed to obtain proper friction properties in agreement with real roads [55].

Figure 2-20: Flat-trac® tire test [56].

2.7. Concluding remarks

Friction is an intricate phenomenon in nature due to the physical interaction of two surfaces in the interface in a stationary or non-stationary situation. Although the basic friction laws of Amontons and Coulomb are relatively simple, they generally do not apply for rubber materials due to their viscoelastic nature. Yet rubber is the only material in nature that can deliver the required tire grip on the road. Owing to its rubbery elastic nature, the tire can distort and drag the vehicle to the desired direction which allows traction and steering to be integrated into one vehicle: car, bicycle, motorbike, etc.

This complicated system interfaces with the road substrate which also has a wide variety, complex structure, and roughness which makes an elaborate task to predict tire behavior in such a conjoined system. Practically, real tire testing is a common way of evaluating their performance. To predict tire performances in a laboratory

environment, a test set-up is required to mimic the tire service condition as close as possible. This was the basis to launch the task which was formulated as the “leitmotif” of the present thesis.

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Measuring rubber friction using a Laboratory Abrasion Tester (LAT100) to predict car tire dry ABS braking

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Abstract

Evaluating tire grip on the road, an extremely complicated tribological system, is enormously energy and time consuming but essential for safety. To predict grip on the road, tires with four different tread compounds were tested on ABS braking distance on a dry test track. Corresponding solid rubber wheels were characterized on the Laboratory Abrasion Tester (LAT100) on four different electro-corundum discs of various grain sizes. With increasing speed the side force coefficient (SFC) decreases. A lower disc grain size induces a higher SFC. A correlation coefficient of 0.93 between the LAT100 data and road results for the four tread compounds has a high potential for predicting the car tire ABS braking distance.

Keywords: LAT100, rubber friction, tire traction, dry grip

² A self-contained chapter; the content of the paper has been transformed completely into this chapter.

3 Prediction of ABS braking distance, A preliminary study

3.1. Introduction

Tires transmit all types of vehicle loads or forces such as vertical, longitudinal braking, driving, cornering, and camber thrust to the road. These forces are necessary for directional control of the vehicle. Tire traction is the result of the generated frictional forces in various driving states such as cornering, braking, and accelerating. It is a concept that describes the grip between the tire and the road. A tire with better grip provides a shorter braking distance wherein the vehicle comes to a full stop. Proper tire grip provides a good level of handling which is a prerequisite for proper vehicle steering, braking, accelerating, and cornering [1].

Tires are built up of a multitude of different elements, mostly rubber compounds, each contributing to the overall performance profile. To improve and develop new tires, extensive testing is required which is time-consuming and costly. Particularly for new rubber tire tread compounds, it is very attractive to be able to predict their performance in a laboratory environment before manufacturing a full tire. To achieve a good

impression of the tire grip, it is essential to have a close look into the forces acting in the two main friction modes: lateral during cornering and longitudinal while braking or accelerating, both with their own specific characteristics.

During cornering, when turning the steering wheel, the tread pulls the rest of the tire and generates forces that pass through the wheel and the suspension system to turn the vehicle. This force generated by the tread is termed lateral or side force; both terms can be used interchangeably. It is actually the tire's resistance to deflection that creates the side force and turns the vehicle [1]. Side force is the force that tires transmit to the ground during cornering in a perpendicular direction of the symmetry plane of the tire. The angle between the tire plane of symmetry and the actual velocity direction of the vehicle in cornering is termed slip angle.

Figure 3-1 shows the general relationship between the side force and the slip angle which is a specific characteristic of a tire design that includes carcass, rubbers used in the tire structure, and the tire tread. This curve is unique for each type of tire which typically has two main regions. In the elastic or linear region at small slip angles, an increase in slip angle leads to a proportional increase in side force. The slope of this linear region of the curve is termed the "cornering stiffness" of the tire. The non-linear region at higher slip angles consists of two different parts. The transitional region: portions of the tire contact patch slide and there is less increase in side force with an increase in the slip angle. This is mainly because of the elastic nature of the rubber. The frictional region: after the peak of the curve, most of the contact patch slides resulting in heat generation and wear.

Figure 3-1: Side (lateral) force vs. slip angle; taken with permission from [1].

During braking, the forces deform the sidewall of a tire in a way that the contact patch moves a noticeable distance in the moving direction generating a longitudinal force [1]. The difference between the longitudinal driving speed and the equivalent circumferential velocity of the wheel is the longitudinal slip [2, 3]. According to the Society of Automotive Engineers (SAE), the longitudinal slip ratio κ is defined as:

$$\kappa = - \frac{V_x - R_e \Omega}{V_x} \quad (3.1)$$

where V_x (m/s) is the longitudinal driving speed and $R_e \Omega$ is the equivalent circumferential velocity of the wheel, where Ω (rad/s) is the rotational velocity of the wheel and R_e (m) is the effective tire radius. This is defined as the radius of the tire when rolling with no external torque applied to the spin axis. Since the tire flattens in the contact patch, this value lies somewhere between the tire's un-deformed and the static loaded radii [4].

During hard braking, the tire rotates less than in the condition without slip. Consequently, the slip ratio is always negative for braking. Therefore, according to Eq. (3.1), in "locked" braking tires, the slip ratio equals a negative -1.0 [3]. Whereas all these phenomena also occur during accelerating conditions, but adversely, the slip ratio is then always positive. Figure 3-2 shows the negative braking slip ratio in a locked wheel.

Figure 3-2: Longitudinal force vs. braking slip ratio.

In order to evaluate tire grip, it is indispensable to have an insight into friction phenomena. Since friction is an intricate phenomenon, tire grip prediction is difficult

and remains a contentious subject of research [4-7]. The texture or roughness of the counter surface is one of the dominant factors of rubber friction. The relationship between the area of real contact determined by the applied load and the surface morphology of the counter surface is a question of intensive research [8, 9]. Various textures contribute differently to the friction components, being adhesion and hysteresis: Eq. (3.2). The basic mechanism of friction in dry relative motion attributes to these two main factors, see Figure 3-3. The first is the adhesion force, which occurs in the real contact area. The second term may be described as a deformation or hysteresis component, related to the viscoelastic damping properties of the tire compounds. The results of high frictional forces are wear or abrasion. This adds a contribution to the total friction force, but is commonly taken along in the hysteretic component. By assuming no interaction between the two components, the total friction force (F) may be written as [10, 11]:

$$F = F_{adhesion} + F_{hysteresis} \tag{3.2}$$

The shape and size of the grains in the counter surface influence directly the friction components. Rubber friction on rough or smooth surfaces involves different mechanisms. Particularly the fine-scale texture below about 0.5 mm is of fundamental importance for dry roads and interacts directly with the tire rubber on a molecular scale [12]. Heinrich, Klüppel, and Persson [8, 13, 14] presented different models for rubber friction on self-affine fractal surfaces that relate the frictional force primarily to hysteretic energy losses of the rubber. On the other side, Grosch [15] indicated that the adhesion still plays a major role in hysteresis friction, so the assumption of no interaction in Eq. (3.2) is still subject of debate.

Figure 3-3: Key components of rubber friction or tire traction on a dry surface.

Grip and wear evaluations of tread compounds suffer from the fact that not only absolute values but also relative ratings strongly depend on the experimental conditions under which they are obtained in road tests [16]. Therefore, a meaningful laboratory evaluation is really needed. Various types of tribometers and testing machines are available which are able to measure tribological properties. However, to measure tire tribological properties a tribometer is required that can simulate tire-road conditions for rubber samples in a laboratory environment.

The so-called Laboratory Abrasion Tester (LAT100) was originally developed by Grosch [16] to measure the side force and the abrasion loss of a sample in the form of a solid rubber wheel as a function of various slip angles, loads, speeds and abrasive surfaces. Afterward, it was further developed to measure wet traction and rolling resistance. It has been demonstrated before that the results of measuring **wet grip** with the LAT100 give a good correlation with ABS braking on the road [16, 17]. Further, good correlations between tire data and laboratory test results have been obtained for abrasion and rolling resistance [16, 18-20]. The machine is able to simulate a wide range of test conditions in a laboratory environment. with a wheel shape sample. This has been mentioned as a high similarity to a vehicle tire which gives a good correlation with real tire data [21]. Therefore, it has the potential to be used as an advanced machine for measuring and predicting tire performance.

In the present manuscript, the rubber friction under **dry conditions** is evaluated in detail by using the LAT100 and compared to real tire test results. For this purpose, various tread compounds were selected and mounted on a standard tire body. The braking distance of each tire was measured in real tire tests and the resulting ratings have been calculated. By using the LAT100 in the laboratory environment, the side forces of the same tread compounds were measured at different test conditions. Finally, the results of the laboratory measurements are correlated with the real tire data ratings and the effects of various speeds and LAT100 disc roughness are characterized.

3.2. Experimental

3.2.1. Tire data

Four proprietary passenger tread compounds with various grip properties were selected for the investigation. Full tires were manufactured with the same construction process using the same body components together with the four different tread compound formulations. The ABS braking distances were measured. Table 3-1 shows

the relative tire ratings which were provided by Apollo Tyres Global R&D based on ABS dry braking tests on the dry test track of the existing proving ground in Europe; the test procedure was comparable to ASTM F1649 for wet braking traction performance of passenger car tires. The higher the rating the shorter the braking distance which indicates better dry traction. The tread named “Tread 1” in Table 3-1 is considered as the reference for the rating calculations.

Table 3-1: Tire ABS braking distance ratings.

Tire sample No.	Rating (%)
Tread 1	100.0
Tread 2	102.6
Tread 3	105.3
Tread 4	102.0

3.2.2. LAT100 test set-up

The machine consists of a driven counter-surface disc on which a solid rubber test wheel is pressed under a normal load $F_z (N)$ at a slip angle $\alpha (^\circ)$: Figure 3-4. The circumferential velocity of the sample wheel V_c is induced by the disc traveling velocity, $V_t = R * \omega$ where R is the distance of the disc center to the center of the wheel contact patch and ω (rad/s) is the rotational velocity of the disc. The traveling speed, slip angle, and load can be varied over a wide range. The average slip velocity V_s in the contact area is created by combining disc traveling velocity and slip angle, as shown in LAT100 trigonometry: Figure 3-5, and given by:

$$V_s = V_t \sin \alpha \tag{3.3}$$

The resultant forces: F_c counter-centrifugal (radial) force, F_s side force, and F_f tangential friction force, are generated by the slip velocity and the acting normal force as shown in Figure 3-5. All force components acting on the wheel during the tests are recorded. To determine dry traction, the side forces are evaluated over a wide range of test conditions on various surface roughness of the discs under a given normal load and calculated slip velocity V_s . Since the aforementioned slip velocity is generated by varying the slip angle, the side force can be evaluated as an important value that enables the rating of the various compounds.

Figure 3-4: Schematic view of the measurement unit of the LAT100, redrawn from [18].

Figure 3-5: Scheme of velocities of the sample wheel and the resultant forces, redrawn from [17, 20].

3.2.3. LAT100 discs

The employed discs were provided by VMI Holland BV- Tire Industry Equipment. They are composed of electro-corundum white Al_2O_3 powder bound in a ceramic binder with a weight ratio of 85/15%. They are classified according to ISO 525:2013(E) as given in Table 3-2, of which the size-fraction of the corundum-particles is the most

important variable in the present context. Each disc code represents the disc grain size; a higher code number indicates a smaller disc particle size as indicated in Table 3-2. The corundum particles in the ceramic binder are pressed together and the disc surfaces flattened by a grinding process using fine steel powder. The letter “B” for disc 180B indicates “Blunt”, which means that the disc 180 was smoothened one step further in the finishing/flattening process using a grinding disc containing diamond powder. This smoothening process does not influence the grain size, however, it smoothenes and flattens the disc surface including the corundum particles. This disc was exploited in an earlier study for **wet grip measurements** with the LAT100 machine and employed first in the present study for comparison [17]. The surface root mean square (rms) S_q of the disc roughness was measured on an area of $1414\text{ }\mu\text{m} \times 1080\text{ }\mu\text{m}$ as given in Table 3-2 as well.

Table 3-2: LAT100 disc characterizations.

Disc designation	60	180	320	180B
Disc code	EKW 60 SHARP	EKW 180 SHARP	EKW 320	EKW 180 BLUNT
Average particle size (μm)	225	60	32	
Corundum sieve analysis (μm)	0%>425	0%>125	Max. of 52*	Same as disc 180
	Max. 30%>300	Max. 15%>90		
	Min. 40%>250	Min. 40%>63	Min. of 19*	
	Min. 65%>212	Min. 65%>53		
S _q ** (μm)	175	65	48	44

* Photosedimentometer analysis (μm)

**The height ranges below than $10\text{ }\mu\text{m}$ were excluded from the calculations

Figure 3-6 shows microscopic images of the disc surfaces obtained with confocal laser scanning microscopy VK 970 Keyence; the height numbers are given in μm . It is clear that with increasing disc code: 60 – 180 – 320, the grain asperities decrease in size. Further, the blunting process of the disc 180B has resulted in a much smoother, flattened surface, in which the individual corundum particles can still be discerned to be of about the same size as in disc 180, but with the asperities reduced to about half the height and a lower rms comparable to disc 320.

Figure 3-6: Roughness surface analyses with confocal laser scanning microscopy of the LAT100 discs employed; height numbers and scales in μm .

3.2.4. Preparation of rubber samples for LAT100

From the same four tire tread compounds, LAT100 rubber sample wheels with an external diameter of 84 mm and thickness of 19 mm were vulcanized in a Wickert laboratory press WLP 1600 according to their optimum curing conditions $t_{c,95}$, as determined with a Moving Die Rheometer (MDR) at 160 °C according to ISO 6502. Due to the thickness of the wheels, 13 mins. extra time was applied to be fully vulcanized throughout; therefore, the curing conditions were set at 160 °C and $t_{c,95}+13$ mins under a pressure of 100 bar. Finally, the samples were stored for 24 hours at room temperature, 22 °C $\pm$ 1 °C. Therefore, the initial temperatures of the samples were equal when starting the LAT100 tests.

3.2.5. LAT100 tests

First screening was performed at constant normal load, various traveling speeds over a wide range of slip angles on disc 180B as shown in Table 3-3 row 1. The wide range of slip angles was selected in a way to cover the potential area of ABS braking for the experiments. This range of slip angles was in a form of single experiment points. After the first screening more detailed experiments were executed on various disc roughness based on the optimized variables, as shown in Table 3-3 row 2.

For realistic and repeatable side force evaluations, it is necessary to remove the shiny outer vulcanization skin from the LAT100-samples surface before measuring the side force. For this purpose, a sample run-in was carried out with 75 N normal load, 25 km/h speed, 13° slip angle, over 500 m distance, and repeated for the right and the left sides of the LAT100-samples on disc 60; according to ISO-23233. Additionally, a disc run-in is also required to be performed before using a brand-new disc. To prepare a correct friction curve, it is crucial to collect all the single experimental points at a same level of surface roughness. Any deviation from the expected surface roughness based on the disc code interferes with the final results. The disc run-in is a pre-conditioning and by means of this, sharp asperities are leveled and the sharpness of the asperities over the disc becomes uniform. After every single test for each slip angle, disc cleaning was performed using an air pressure nozzle which removed the rubber debris and particles from the disc surface. All the tests were performed at 75 N normal load which is equal to the average load in a tire contact patch on the road [16, 18].

Table 3-3: Variables employed in dry grip tests.

Speed (km/h)	Corresponding distance (m)	Slip angle range (°)	Disc code
0.2, 2.1, 20	2, 17, 120	3 to 35	180B
2.1	2, 17, 120	3 to 35	60, 180, 320, 180B

For all test procedures, the experiments were carried out for the same 4 tread compounds. For every procedure 5 LAT100-samples were prepared, of which LAT100-samples 1 and 5 correspond to compound 1. This compound is considered as the reference to cross-check the experiments in each measuring conditions. To avoid creating uneven wear or a conical shape on the surface of the LAT100-samples during the tests, the side force measurements were performed for both sides of the LAT100-samples. The calculated Side Force Coefficient (SFC) for each data point is an average

SFC over the right and the left sides of the LAT100-sample. To compare the friction properties, the SFC was calculated according to:

$$SFC = \frac{F_s(N)}{F_z(N)} \quad (3.4)$$

To correlate the LAT100 results with real road data, the SFCs of the four tread compounds measured by LAT100 were rated as follows:

$$SFC \text{ rating } \% = \frac{SFC \text{ Sample}}{SFC \text{ Reference}} \times 100 \quad (3.5)$$

3.3. Results and discussion

The first screening was performed at three different traveling speeds on the disc 180B based on the earlier study on wet grip using the LAT100 [17]. Figure 3-7 shows the SFC versus slip angle curves for the five LAT100-samples (the four tread compounds) at various traveling speeds. All the SFC values are normalized by dividing through the largest SFC value of the reference compound and expressed in percentage. Therefore, the orders of magnitude of the SFCs are comparable. The expected peak position of the curves varies with different speeds in this experiment. With increasing speed the peak position shifts to the left, towards lower slip angle, representing the LAT100-samples to start sliding at lower slip angles. At the speed of 0.2 km/h, none of the compounds demonstrates all frictional regions according to Figure 3-1. The moderate speed, 2.1 km/h, shows all aforementioned friction curve regions. At high speed, only compound 1 shows all the described regions.

Figure 3-7: SFC vs. slip angle at various traveling speeds for the five LAT100-samples on the 180B disc, samples 1 and 5 are the same compound.

Above 20° slip angle at 20 km/h, the severe test condition leads to an elevated sample temperature and abrasion into rubber particles [22]. It has been shown that changes in the rubber particle size affect tribological properties which are closely related to slip-stick phenomena [23]. The steadily increasing curves at speed 20 km/h for samples 2, 3, and 4 are the result of this high accumulated temperature on the rubber surface. This higher temperature increases the friction coefficients and consequently raises the SFCs. The drop in the SFCs beyond 20° for the reference compound 1 most probably depends on the type of abrasion pattern. This compound abrades with another form of abrasion called “roll formation”. This phenomenon occurs due to continuous tearing of the rubber and accumulation of the separated shred in the form of a circumferential roll on the surface of the rubber wheel. It occurs for a relatively high coefficient of friction in the rubber contact with the counter surface and leads to internal failure of the outer surface of the rubber sample [10]. This roll formation decreases the side and friction forces.

Figure 3-8 shows the SFC versus slip angle for each tread compound at various speeds. The higher disc traveling speeds show mostly a larger SFC for all the compounds; which is seemingly opposite to the drivers feeling at high speeds. The trend does not occur for compound 1 at the severe test conditions, which is mainly because of elevated temperature and abrasion in the contact area as indicated before.

Figure 3-8: SFC vs. slip angle for each compound at various speeds on the 180B disc.

Figure 3-9: SFC vs. slip velocity for compound 1 on the 180B disc; insert zoomed in at the lower slip velocity range.

In order to solve the previous ambiguity, the curves are replotted as normalized SFC based on the average slip velocity in the contact area according to Eq. (3.3) instead of traveling speed. As an example, the SFC versus the slip velocity for compound 1 is plotted in Figure 3-9. Now the lower slip velocity shows higher friction values. At 0.2 km/h the slip velocity varies in the range of 0.003 to 0.11 km/h, corresponding to a slip angle of 3° to 35°, which is also enlarged in Figure 3-9. The lower the speed the smaller the slippage which leads to higher friction. It reflects that with increasing vehicle speed the friction decreases and the tire traction level could drop to even an insufficient level, as the LAT100 also demonstrates.

To correlate the LAT100 results with the tire braking distance data, the averages of the SFC ratings over different ranges of slip angles were calculated: for the full slip angle range, for the low slip angle range between 3° - 23°, and for the high slip angle range between 25° - 35°, as indicated in Table 3-4. The reason for this division was based on the shape of the friction curves which is termed in this study SFC curve. In most curves in Figure 3-7 and Figure 3-8 two regions are visible comparable to Figure 3-1 and Figure 3-2, with a maximum of around 20° - 25° slip angle. Therefore, the ranges of slip angles in Table 3-4 were chosen accordingly.

Based on the calculated average values and the tire data, the correlation coefficients R^2 and linear slopes were evaluated by linear regression using Matlab R2015b software and are also summarized in Table 3-4. All correlation coefficients are on a low level which means that there is no significant correlation between the performed LAT100 tests and the tire data on the disc 180B. It indicates that the blunt disc 180B does not properly reflect the road test conditions in the current experiments. The surface of the 180B disc with blunted corundum particles is apparently not providing enough grip in the rubber contact patch to sufficiently reflect hysteretic tire friction on the real road.

The question may be raised whether there are temperature differences between the four compounds in the aforementioned grip tests. Dependent on the speeds, the difference between the final surface temperature of the LAT100-samples at the lowest and highest slip angles, the temperature increments are as little as 2 °C for the low speed of 0.2 km/h, 24 °C for the speed of 2.1 km/h, and maximally 100 °C for the speed of 20 km/h. Despite this, the differences among the respective compounds are comparable; for speed 0.2 km/h it was negligible, for speed 2.1 km/h maximally 1 °C, and for speed 20 km/h 12 °C for the largest slip angle. In order to minimize the effect of temperature differences among the samples to make them comparable to one another, this will be an important argument for the next part study.

Table 3-4: Correlation coefficients and linear regression slopes for the first screening with 180B disc.

	Slip angle range	0.2 km/h	2.1 km/h	20 km/h
R^2	Full	0.30	0.66	0.51
	25°-35°	0.61	0.39	0.61
	3°-23°	0.18	0.50	0.09
Slope	Full	-0.82	-0.53	-2.88
	25°-35°	0.50	0.16	-6.56
	3°-23°	-1.41	-0.84	-1.25

Subsequently, to determine which disc predicts the road conditions, a set of experiments including the three additional surface roughness discs at a constant moderate speed of 2.1 km/h were performed: Table 3-3, row 2. The constant speed of 2.1 km/h was adopted from the previous measurements, wherefore the SFC curves covered all regions indicated in Figure 3-1 and the differences between the surface temperatures among all compounds were comparable.

The results for all compounds are shown in Figure 3-10; the larger the disc grain size the lower the SFC. A smaller grain size induces a larger real contact area because of the higher number of asperities within the apparent unit area. This is due to the fact that the rubber can follow and penetrate into the indentations of the counter-surface, as determined by the hardness/modulus of the compound. Although a single surface irregularity induces lower hysteretic friction, the cumulative frequential excitation of the rubber over all irregularities of the counter surface is higher: Figure 3-3. The higher the cumulative hysteresis, the higher the SFC which leads to more energy dissipation of the rubber. This explains the higher hysteretic friction for disc 320 in the range of surface irregularities according to Table 3-2. On the other side, the disc 60 induces the lowest SFCs, because it has the biggest particle size and highest rms as a surface parameter in Figure 3-6.

In spite of the same particle size of discs 180 and 180B, the blunting has reduced the rms to approximately the same level as disc 320, as indicated by the height order in the laser microscopic images: Figure 3-6. The disc 320 shows SFC values mostly in the same order as the 180B disc. Because of the flattening process of the 180B, it may result in a lower hysteretic component and a higher contribution of an adhesion component. The contributions of both friction components in dependence of speed and surface micro- and macro-structure remain a point of debate. Persson in his review article of 2005 [9] implied that there is almost no adhesion between a rubber ball and a substrate with a rms larger than 1 μm . The order of magnitude of all employed discs rms is much higher for all. Grosch and others state that the adhesion component still plays a role and has an important contribution at low speeds and dry conditions [15, 24-27] which was extensively discussed based on the area of real contact for dry rubber friction [27]. Wagner even suggested an adhesive part for wet surfaces [25]. The results of the present study indicate that the lower disc rms and particle size are both involved in higher friction. Anyhow, the argument here is not to identify which friction component is responsible for these phenomena. For this purpose, more experiments would be needed in order to identify the friction components [14, 25, 28, 29]. The current results are showing that the lateral friction force decreases with increasing the measured rms, and the LAT100 clearly distinguishes this.

Figure 3-10: SFC vs. slip angle curves, the effect of the disc surface roughness for the four compounds, at a speed of 2.1 km/h.

Similar to the previous series of tests, the correlation coefficients and the linear slopes between the tire data and the average SFC ratings over the different slip angle ranges for the various discs are summarized in Table 3-5. Disc 180 in the slip range of 25°- 35° shows the best correlation between the LAT100 results and the tire data with a positive regression slope. The negative slopes in Table 3-5 are to be taken as outliers without physical meanings. Figure 3-11 (A) demonstrates the normalized SFC versus slip angle curves for the five LAT100-samples on disc 180. Figure 3-11 (B) shows the correlation between the tire data and the average SFC ratings over the 25°- 35° slip angle range with the corresponding standard deviations.

Figure 3-11: (up) SFC vs. slip angle curves for the five LAT100-samples on disc 180, (bottom) correlation between tire data and average SFCs over 25°- 35° slip angle at a speed of 2.1 km/h.

Table 3-5: Correlation coefficients and linear regression slopes between the tire ratings and the average SFC ratings over different slip angle ranges for various discs, at a speed of 2.1 km/h.

	Slip angle ranges	Disc 60	Disc 180	Disc 320	Disc 180B
R ²	Full	0.26	0.87	0.35	0.08
	25°-35°	0.81	0.93	0.41	0.02
	3°-23°	0.49	0.49	0.02	0.29
Slope	Full	0.17	0.60	0.56	-0.20
	25°-35°	-0.49	0.93	1.40	-0.20
	3°-23°	0.54	0.42	0.04	-0.20

It does indicate that the disc surface roughness has a significant influence on the collected SFCs with the LAT100 as discussed. Disc 180 simulates the tire condition on the road the best and accordingly gives a promising correlation between the lab results and the current ABS braking distance data, provided the LAT100 data are taken in the slip angle range between $25^\circ - 35^\circ$. The original, non-blunted asperities from the corundum-particles in the present disc 180 do provide a substantial contribution of hysteretic friction contact relative to the blunted variant. By comparison of Figure 3-1 and Figure 3-2 with Figure 3-11(A) it is clear that the transitional region observed within this slip angle range does correspond with the transitional region of the longitudinal tire friction force, ref. Eq. (3.1), there where ABS-braking is indeed active.

The applied normal load of 75 N on the LAT100-samples corresponds to around 0.34 MPa contact pressure in the LAT100-sample apparent contact area, which is approximately 2.2 cm^2 for a rubber sample with a medium hardness of 65 °Sh A. This contact pressure is in a similar range like for an average size passenger car tire. The maximum coefficient of friction for a car tire is expected for a slip rate between 5% and 15% [30]. For example, based on Eq. (3.1) the slip velocity of a car tire traveling at 90 km/h at 10% slip ratio is 2.5 m/s. The correlation presented in Figure 3-11(B) was made for a traveling speed of 2.1 km/h and in the range of $25^\circ - 35^\circ$ slip angle. Therefore, it is possible to estimate the average slip velocity in the contact area of the LAT100-samples according to Eq. (3.3) which is calculated to be 0.29 m/s for the 5 experimental points for each LAT100-sample in the range of $25^\circ - 35^\circ$ slip angle in Figure 3-11(A). This value of 0.29 m/s is around 8 times smaller than the tire slip velocity which is comparable to the ratio of tire circumference to the LAT100 wheel circumference. This is still an interesting point to be studied further by modeling and investigating the difference between the friction mechanisms: the longitudinal friction for ABS braking and the lateral type friction for the LAT100 wheel.

The question is why the disc 180 offered the best correlation. By comparing the surface roughness parameter before and after the test procedures: Table 3-6, it is observed that except for disc 60, there are no large changes of rms. This shows that the discs except for 60, remain stable during the test procedure. A possible reason for the strong correlation for disc 180 is similarity of the frequential excitations of the tread compounds in their respective counter-surface contact areas: between the LAT100-samples and the disc, the tires on the road. As soon as a LAT100-sample starts slipping on disc 180, the macro- and micro-roughness of the counter surface causes a frequential excitation of the rubber. This has to be in a similar range as the penetration of tire rubber into the indentations of the road surface. This excitation frequency is dependent on the distance between surface irregularities and the slip velocity of the rubber over

the counter-surface [30]. Provided that the frequency is calculated according to the surface macro-irregularities with a distance d (m):

$$\text{Frequency (s}^{-1}\text{)} = \frac{\text{slip velocity (m/s)}}{d \text{ (m)}} \quad (3.6)$$

for the average slip velocity of 0.29 m/s of disc 180 with a distance of 200 μm for hysteresis macro-irregularities (Figure 3-6), the value of this frequential excitation is $1.5 \cdot 10^3 \text{ s}^{-1}$. The molecular adhesion frequency range between rubber and a road varies from 10^6 s^{-1} to 10^9 s^{-1} and for hysteresis due to road roughness between 10^2 s^{-1} to 10^6 s^{-1} [30]. The value of $1.5 \cdot 10^3 \text{ s}^{-1}$ is therefore in the range of hysteresis friction or road indentation. Apparently, adhesion is not playing an important role here. The disc is composed of a range of particle sizes which are influencing the surface irregularities distance though. This is an open point for more investigations to estimate the actual area of contact and the range of frequential excitations. Understanding the grip mechanism of disc 180 will therefore be the subject of more research.

Table 3-6: surface roughness parameter before and after the test.

Disc designation	60	180	320	180B
S_q^* (μm) before	175	65	48	44
S_q^* (μm) after	125	50	46	33

*The height ranges below than 10 μm were excluded from the calculations

3.4. Conclusions

To predict the car tire ABS braking distance on dry tracks, friction properties of solid rubber wheels were measured using the Laboratory Abrasion Tester (LAT100). Tires with four different tread compounds were rated by measuring ABS braking distances on the road. The same compounds were characterized on the LAT100 at various traveling speeds and surface roughness of the counter-discs. At a constant slip velocity the Side Force Coefficient (SFC) decreased with increasing speed. The LAT100 clearly distinguishes the effect of disc surface roughness on the SFC. The lower the disc grain size and rms roughness, the higher the SFC because of the larger real contact area of the rubbers with the electro-corundum discs. That induces a higher cumulative frequential excitation of the rubber over the irregularities of the counter surface. The results of LAT100 were further correlated with the tire data. Disc 180 shows the best correlation

and the most promising results among the various discs. It reflects the road surface conditions in the best way between the available tire data and LAT100 test conditions with a correlation coefficient of 0.93 for the four tread compounds. It has a high potential for predicting the car tire ABS braking distance.

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Parameter optimization for a laboratory friction tester to predict tire ABS braking distance using design of experiments

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Abstract

Predicting tire friction in a laboratory environment is a complicated issue. Varying one factor at the time for experimenting to obtain insight into tire friction does not necessarily provide a proper judgment. It is needed to take all interactions between the operating factors into account to determine the optimized test conditions to predict tire friction. The research methodology of the Design of Experiments (DoE) is particularly suited for such a system influenced by multi-factors; a technique that constructs a predictive mathematical model to describe the relationship between responses and critical factors. With the aid of DoE, the influence of various operating parameters of the Laboratory Abrasion Tester (LAT100) as a tribometer on tire friction is investigated: load, speed, and slip angle. Two counter-surfaces are employed: a sandpaper- and a corundum-based substrate. A predictive mathematical model is extracted from the DoE and suggests various optimal test conditions to predict the tire friction. It provides new insights into the impact of factor interactions on the coefficient of friction of tire tread compounds. The variation of the coefficient of friction vs. speed on the corundum-based substrate is in good agreement with the ABS braking distance data of actual tires on the road.

Keywords: Tire friction, ABS, braking distance, rubber, LAT100, Design of Experiments, tribometer

³ A self-contained chapter: the content of the paper has been transformed completely into this chapter.

4 Prediction of ABS braking distance using design of experiments

4.1. Introduction

Evaluating tire grip on the road as an extremely complicated tribological system, consumes time and energy but is essential for safety. Proper tire grip provides a good level of handling which is a prerequisite for vehicle steering during cornering, braking, and accelerating. The tire grip is the result of the generated frictional forces which are created by the tire slippage in the contact patch. Commonly, with an Anti-lock Braking System (ABS), the braking force is optimized based on a specific region of tire slippage, typically 5 to 15 percent [1, 2], to provide adequate frictional forces and avoid tire skidding. Numerous factors influence tire friction: the tire construction, tread pattern and composition; the characteristics, condition, and structure of the counter-surface; and last the weather and temperature conditions [3]. Overall, the tire tread compound contributes the largest portion of around 75% of grip on a dry surface [4]. Arrigoni S. et al. reported that the rubber coefficient of friction has the largest influence on ABS performance: a 5% increase leads to roughly 2 m reduction of ABS braking distance [2].

Rubber friction is strongly dependent on operating conditions such as speed, load, and their interactions, due to its viscoelastic properties. Therefore, tire grip prediction is difficult and remains a contentious subject of research. For tire friction evaluations, so far abundant attention has been paid to the prediction of tire friction especially for car ABS braking or stopping distance [5-13]. Between laboratory tribometers, both linear and rotary measurements, with ABS braking data good agreements have been acquired. A Linear Friction Tester (LFT) provides a strong correlation with outdoor ABS data on wet and snow surfaces, but not for dry surfaces [9, 14]. The portable LFTs provide the possibility to perform in-situ measurements on the actual testing fields or road surfaces. Some good agreements have been reported between Laboratory Abrasion Tester (LAT)100 results and ABS braking data on different dry or wet surfaces [15-17]. Klüppel and Heinrich [7] predicted ABS dry braking on asphalt using the physics of rubber friction theory for 4 tread compounds.

Varying One Factor At the Time (OFAT) is not adequate for friction experimenting. Taking the interactions between the operating factors into account allows assessing the optimum test conditions together with distinguishing the causes of various unexpected outcomes. Design of Experiments (DoE) is a research methodology that can construct a predictive mathematical model to describe the relationship between responses and critical factors and their mutual interactions. If the model is adequate, it can be applied to explore the optimal test conditions for desired responses and also to predict the outcome for new observations. The other advantage of DoE is randomization which balances the influence of uncontrollable conditions, reduces the risk of systematic errors, and increases the chance of obtaining unbiased and reproducible results.

Based on our previous research [17, 18], the LAT100 – despite its name – can also be used as a tribometer able to simulate a wide range of test conditions in a laboratory environment with a so-called Grosch wheel mimicking a real tire. This apparatus allows varying the experimental operating parameters in a broad range. The objectives of the presently introduced methodology are to investigate the influence of various operating parameters of the LAT100 as a sliding friction tester. To study the interactions between these factors for achieving the optimum test conditions of LAT100 in correlation with the tire ABS braking distance on a dry surface.

4.2. Design of experiments methodology

The approach and used terminology of DoE as employed in the present context is as follows [19-21]:

- **Factors:** can be denoted as input variables that affect the process or the final outputs. Factors can be continuous (velocity, load, ...) or categorical (a type of surfaces or additives, ...).
- **Responses:** the product or process characteristics of interest to the experimenter and often referred to as output variables, e.g. coefficient of friction, correlation coefficient, the goodness of a fit. Adequate response criteria are Specific, Measurable, Attainable, Realistic and Time-limited (SMART).
- **Factor levels:** the particular values or types of factors. For a continuous factor such as velocity, the levels could be 1 km/h and 2 km/h, and a categorical factor, such as type of surface could be type1: sandpaper, type2: asphalt, etc.
- **Design:** a pre-defined set of experimental runs e.g. two-level full (fractional) factorial or multi-level central composite design.
- **Model:** a predictive mathematical equation describing the response as a function of the factors;
- **Main effect:** there is a main effect when different levels of one factor affect the response differently.
- **Interaction effect:** when the effect of one factor depends on the level of the other factor. An interaction plot can be used to visualize possible interactions.
- **Residuals (or residual error):** the difference or the error between the observed value and the predicted value by the model.

The reader is referred to 4.7.1 for details and elaborations on steps taken for the DoE and interpretations of the results.

In the present study, experimental designs are defined for load, speed, and slip angle as influential factors. The initial steps of DoE such as the screening process were eliminated based on the previous publications and prior knowledge of the LAT100. Two modeling designs, DoE 1 and 2, are created based on the three factors on two different experimental tracks of sandpaper P120 and LAT100 disc 180; with large level ranges to include the extreme test conditions. In the next step, conformation runs were carried out to compare the results with the predictive mathematical model. Optimization was carried out by applying a multi-level general factorial DoE 3, which was performed on the optimal surface. The effect of load and speed was further investigated to explore the underlying reasons for the correlation and to obtain more insight into the friction phenomenon.

4.3. Experimental

4.3.1. Tire data

Five proprietary summer passenger tread compounds were selected for the investigation. Summer tires size 225/50R17 from the various tread compounds were built. The full tires were manufactured with the same construction process using the same body components together with the five different tread compound formulations. The tire inflation pressure was 0.230 MPa. The test procedure of ABS braking was comparable to ASTM F1649 for wet braking traction performance of passenger car tires. The standard specifies that the test vehicle can be any commercially available car that is equipped with ABS. For the current experiments, the same commercial vehicle and ABS version on the same counter-surface were employed. The braking test can be conducted on a straight line and the stopping distance can be measured, which is defined as the distance that the vehicle comes to stop from an initial speed. The typical ratio of the slip velocity to the forward velocity of the tire, i.e. the slip ratio, for a passenger car tire during ABS braking lies in between 5% -15% [1, 17]; this is the range that the ABS mechanism is optimized to maintain the highest grip or friction between the tire and the road. Table 4-1 shows the relative tire ratings which were provided by Apollo Tyres Global R&D based on ABS braking tests on the dry test track of the existing proving ground in Europe; the higher the rating the shorter the braking distance. The tread named "Tread A" in Table 4-1 is considered as the reference for the rating calculations. The road and air temperatures were 35 °C and 26 °C, respectively.

Table 4-1: Tire ABS braking distance ratings; CV is the Coefficient of Variation: the ratio of the standard deviation to the mean value.

Tire codes	Rating (%)	CV (%)
Tire A	100.0	0.6
Tire B	104.3	1.1
Tire C	104.5	1.0
Tire D	98.3	0.8
Tire E	100.1	0.7

4.3.2. Laboratory sample preparations

From the same tread compounds, laboratory rubber test samples known as Grosch wheels were prepared. The test samples were solid rubber wheels with a thickness of 19 mm and external and internal diameters of 84 mm and 35 mm, respectively. The test

samples were vulcanized in a Wickert laboratory press WLP 1600 for 11 minutes at 170 °C under a pressure of 160 bar, optimized on basis of t_{c90} curemeter data. The curing times of both test laboratory wheels and tire treads were comparable. Finally, the samples were stored for 24 hours at room temperature, 22 °C $\pm$ 1 °C. Therefore, the initial temperatures of all samples were equal when starting the experiments. After curing, the samples were pre-conditioned with the LAT100 to remove the vulcanization skin according to the test conditions which were explained in our previous study [17].

4.3.3. Laboratory test set-up

The LAT100 manufactured by VMI Holland B.V., the Netherlands, is used as the laboratory tribometer. The details of the test set-up were fully described in our previous manuscript [17]. The measurement unit consists of a rotary counter-surface disc with a diameter of 350 mm and a Grosch wheel which is a solid rubber test wheel fixed to a measuring hub. Using a metal pin the Grosch wheel is locked to the hub so that it cannot rotate with the counter-surface disc for the current experiments. Therefore, the sample performs as a locked wheel like a sliding body on the counter-surface disc. This differs from our earlier study wherein the rubber wheel was free to rotate in contact with the disc [17]. The wheel is pressed under a vertical load of F_N (N) normal to the driven disc under a specific angle α . The wheel slides over the counter-surface as a function of the rotation speed of the disc, however, the sliding speed of the sample in the contact area is influenced by the disc radius which rises as the slip angle increases. The resultant frictional forces are generated by means of three input machine parameters: the disc velocity, the angle known as 'slip angle', and the acting normal force. A schematic view of the measurement unit is depicted in Figure 4-1. The area of the sliding body in contact with the counter-surface disc is hatched.

Figure 4-1: Test set-up and rubber wheel under normal load F_N (N), angle α ($^\circ$), friction force F_f (N), and velocity V_t (km/h).

4.3.4. Set-up of the design of experiments

4.3.4.1. Factors

Table 4-2 shows the input and output parameters of the LAT100 apparatus and the possible ranges. The selected factors in the current experimental designs were narrowed to normal load, speed, and (slip) angle. The importance of normal load and speed is well studied in the rubber tribology field [13, 18, 22-24]. The slip angle variation not only alters the leading edge of the rubber wheel in contact with the counter-surface but affects the collected friction and lateral forces in the experimental set-up as explained due to disc curvature in section 4.3.3. Based on our previous studies, the distance for each velocity was optimized to the lowest possible amount to obtain a steady and stable friction coefficient and simultaneously avoid excessive abrasion in

the contact area [25]. The measurements were carried out at room temperature which is close to the temperature where the tire data were performed. The final DoE models for two experimental tracks were also further compared: sandpaper P120 and a LAT100 disc 180. The former surface is the test track that is widely being employed for tire indoor flat-track tests which being reported as unrealistic surface compared to the road [6, 26]. The data from Flat-track rig is used for vehicle simulation. The latter is the optimized counter-surface based on our previous studies in which the grip properties of the experimental rubber wheels are best in agreement with actual tire data on the test track road surface [17, 18]. Disc 180 is composed of electro-corundum white Al_2O_3 powder bound in a ceramic binder with a weight ratio of 85/15 classified with code 180 according to ISO525:2013 (E). Sandpaper P120 is the grit 120 of fine grade of Silicon Carbide which is classified according to ISO/FEPA Grit Designation. Microscopic images of the experimental tracks were obtained with confocal laser scanning microscopy VK 970 Keyence over an area of 5 cm^2 with a pixel width of $10 \text{ }\mu\text{m}$, see Figure 4-2. The surface roughness is characterized by the 2D-PSD (Power Spectral Density) and compared with a typical concrete road surface, see Figure 4-3.

Table 4-2: Input and output parameters of LAT100 for dry grip.

Input parameters	The possible range	Output parameters
Normal load	10 N to 140 N	Friction force (5 to 100 N)
Speed	0.002 km/h to 100 km/h	Side force (-120 to 120 N)
Slip angle	-45° to 45°	The temperature of the rubber wheel
Distance	1 m to 20 km	
Disc temperature	-20 °C to +80 °C	
Disc type	Various

Figure 4-2: Laser scanning microscopic images of the experimental tracks: LAT100 disc 180 and sandpaper P120; scale is $100 \text{ }\mu\text{m}$.

Figure 4-3: 2D-PSD of the experimental tracks compared to concrete.

4.3.4.2. Factor levels

In the first step, quite a large range of loads, speeds, and slip angles was taken into account. Based on that, two designs of experiments were employed: DoE 1 and 2. Based on the outcome of the two first designs, DoE 3 was created, see later. The summary of all factors, levels, and responses for all designs are shown in Table 4-3.

Table 4-3: The summary of factors, levels, and responses.

Factors	Levels						Responses
	DoE 1 & DoE 2			DoE 3			
	lower	center	upper	lower	center(s)	upper	
A Load (N)	35	55	75	35	55	75	CC ⁱ COF ⁱⁱ
B Speed (km/h)	0.1	5.05	10	0.005	0.01, 0.1, 1	5	
C Slip angle (°)	0	15	30	15			

i Correlation Coefficient, ii Coefficient Of Friction

4.3.4.3. Responses

The responses are considered the Coefficients of Friction (COF) and Correlation Coefficients (CC) between the LAT100 experimental results and ABS braking distance data of the tires. CC is the linear regression value R^2 between the measured coefficients of friction of the 5 tread compounds in the laboratory and the tire data presented in

Table 4-1. To avoid confusion between R^2 of the response value and R -sq of the predictive mathematical models, the notation of CC is being used for the response.

4.3.4.4. Designs and calculations

Multi-level designs were selected for the present study. The properties of a full factorial design are an excellent approach for the experimenter to be aware of confounding⁴ effects. DoE 1 and 2 were performed with Response Surface Modeling (RSM, see 4.7.1) using a Face-centered Central Composite (FCC) design for the 3 factors. DoE 3 was created to optimize the output of the first two designs by exploiting a multi-level full factorial design with 5 levels for speed and 3 levels for the load. The (cube) plots of the designs are illustrated in Figure 4-4. One example of 20 test runs of test conditions according to FCC design is shown in Table 4-4, as carried out for each compound. The 20 runs were performed for all 5 compounds, a total of 100 experiments with additional randomization of the wheel specimens A-E. This reduces the effect of compound type in the sequence of test runs. It should be noted that in Table 4-4 due to randomization the design order and run order are different, as was employed for all of the design experiments in the present study.

A novel and effective way of cleaning the counter-surface was developed within the project, in which the trace of previous rubber compounds was removed from the experimental tracks. Using a rubber eraser with a unique formulation, the trace and effect of the previous compound were fully erased before each test run, see Figure 4-5.

The coefficients of friction were calculated by dividing the friction force by the applied normal load for 100 experiments for DoE 1 and 2. Then, each set of friction coefficients for the 5 compounds at every test condition of Table 4-4 were correlated linearly to the ABS braking distance data for the 5 tires. The response CC, the regression coefficients of the 20 runs of experiments for the 5 compounds, were applied for the interpretations of the experimental designs and the generation of the predictive models with statistical analysis. An overview of the whole experimental designs is given in Table 4-5.

⁴ Confounding: a situation in which a measure of association or relationship between exposure and outcome is distorted by the presence of another variable [<https://online.stat.psu.edu/stat507/node/34/>].

Figure 4-4: The plots of A) Face-centered Central Composite (FCC) and B) General full factorial, multi-levels for 2 factors; load (N), and speed (km/h).

Figure 4-5: Counter-surface disc cleaning process.

Table 4-4: Test runs for DoE 1 and 2 for one compound.

Test runs (randomized)		Factors		
Run order	Design order	Load (N)	Speed (km/h)	Angle (°)
1	13	55	5.05	0
2	18	55	5.05	15
3	16	55	5.05	15
4	14	55	5.05	30
5	8	75	10	30
6	15	55	5.05	15
7	7	35	10	30
8	3	35	10	0
9	5	35	0.1	30
10	1	35	0.1	0
11	9	35	5.05	15
12	19	55	5.05	15
13	2	75	0.1	0
14	12	55	10	15
15	20	55	5.05	15
16	10	75	5.05	15
17	17	55	5.05	15
18	11	55	0.1	15
19	6	75	0.1	30
20	4	75	10	0

Table 4-5: Overview of the DOEs.

	DOE 1	DOE 2	DOE 3
Load range (N)	35-75	35-75	35-75 (3 levels)
Speed range (km/h)	0.1-10	0.1-10	0.005-5 (5 levels)
Slip angle (°)	0-30	0-30	15
Disc type	P120	Disc 180	Disc 180
Design	FCC	FCC	Multi-level factorial
Number of runs per compound	20	20	15
Total number of runs	100	100	75

4.4. Results and discussion

The objectives of the designed experiments are to identify the optimized test conditions which predict the tire ABS braking distance with the LAT100 in sliding mode. More importantly to gain insight into the impact of each parameter on friction

and the underlying reasons for the possible correlations. Each DoE can imply whether an optimized response exists. With the analysis of DoE 1 and 2, the influence of the two counter-surfaces i.e. sandpaper P120 and disc 180 can be compared. A final optimization is performed with DoE 3.

4.4.1. Analysis of DoE 1: Sandpaper P120 counter-surface

DoE 1 was carried out on the counter-surface of sandpaper P120. For the designed test conditions in Table 4-4, the friction coefficients of the 5 compounds are presented in Table 4-6. The last column is the response CC: the correlation coefficient value between the friction coefficients of the 5 laboratory compounds at each designed test condition and the data of the 5 tires in Table 4-1. The target value of the response is a maximum of 1.0 and the desired range above 0.9. At first glance, it might seem that the test conditions with the highest value preferably above 0.9 of CC, is the region of interest for the optimal test conditions: i.e. test runs 18 and 19. However, further interpretations and analyses are required to prove whether this is an arbitrary coincidence of correlation or not: Table 4-7, Table 4-8, Figure 4-6, and Figure 4-7. The confidence and error levels are considered 95% and $\alpha = 5\%$, respectively.

Figure 4-6: Pareto chart of the Standardized Effects of a full quadratic model including all possible terms in the vertical axis; DoE 1 on sandpaper P120.

Table 4-6: The coefficients of friction of the laboratory results for each test run of DoE 1 and the calculated responses CC with sandpaper P120 counter-surface.

Run Order	Laboratory Coefficient of Frictions (COF)					Response
	Comp A	Comp B	Comp C	Comp D	Comp E	
1	1.025	1.221	1.309	1.156	1.242	0.36
2	1.059	1.326	1.259	1.029	1.191	0.83
3	0.994	1.248	1.170	1.025	1.152	0.66
4	0.891	1.061	1.047	0.879	0.994	0.82
5	0.739	0.877	0.846	0.804	0.812	0.53
6	1.007	1.229	1.129	1.006	1.116	0.69
7	1.026	1.119	1.045	0.922	1.029	0.69
8	1.190	1.317	1.278	1.066	1.224	0.83
9	1.093	1.251	1.293	0.969	1.180	0.86
10	1.290	1.412	1.473	1.194	1.441	0.65
11	1.031	1.368	1.329	1.064	1.298	0.65
12	0.942	1.176	1.173	0.980	1.116	0.70
13	1.200	1.360	1.375	1.070	1.283	0.85
14	0.938	1.173	1.004	0.956	1.007	0.50
15	1.008	1.261	1.142	0.974	1.105	0.77
16	0.909	1.164	1.015	0.928	1.027	0.56
17	1.019	1.309	1.177	0.970	1.175	0.69
18	1.139	1.412	1.333	1.042	1.219	0.91
19	0.925	1.142	1.149	0.885	0.970	0.98
20	0.938	1.204	1.053	0.945	1.082	0.54

Figure 4-7: Normal plot of the Standardized Effects; DoE 1 on sandpaper P120.

Table 4-7: ANOVA of DoE 1, response surface regression of the full quadratic model on sandpaper P120.

Source		DF ⁵	P-Value	P-Value < 0.05
Full quadratic model		9	0.110	-
Linear terms		3	0.042	Significant
Load	A	1	0.601	-
Speed	B	1	0.014	Significant
Slip angle	C	1	0.130	-
Square terms		3	0.697	-
Load*Load	A ²	1	0.828	-
Speed*Speed	B ²	1	0.275	-
Slip angle*Slip angle	C ²	1	0.741	-
2-Way Interaction terms		3	0.140	-
Load*Speed	A*B	1	0.052	-
Load*Slip angle	A*C	1	0.867	-
Speed*Slip angle	B*C	1	0.185	-

Table 4-8: Model summary of DoE 1.

model	S	R-sq	R-sq(adj)	R-sq(pred)
Linear	0.132	39%	28%	0%
Linear + interactions	0.140	44%	18%	0%
Full quadratic	0.123	67%	37%	0%

From the Pareto chart of the standardized effects of a full quadratic model on DoE 1, it is deduced that the speed bar crosses the reference line and statistically is the only significant parameter with a 95% confidence interval. The normal plot represents that increasing speed reduces the chance of predicting the tire data with the sliding body in the LAT100 test set-up. Table 4-7 presents the model terms and the P-value of the ANOVA interpretation of DoE 1, response surface regression of the full quadratic model. The P-value smaller than 0.05 is marked as significant. The summary of the different fitted models including various terms is extracted in Table 4-8. The goodness-of-fit in a 'model summary table' is determined with S, R-sq, R-sq(adj), and R-sq(pred) statistics therewith examine how well the model fits the data.

⁵ It is comprised of the number of terms in the model; for instance, DF in Table 4-7 consists of 3 linear, 3 squared, and 3 2-way interactions terms.

- S represents the variation of the data values from the fitted response surface.
- $R\text{-sq}$ measures how well the model fits the data.
- $R\text{-sq}(\text{adj})$ is used for model comparisons that have different numbers of predictors.
- $R\text{-sq}(\text{pred})$ shows how well the model predicts the response for new observations.

The reader is referred to 4.7.1 for more explanations regarding the statistics. The full quadratic model has the lowest S and highest $R\text{-sq}$ among others, however, the value 67% for $R\text{-sq}$ is a rather low value for a model. 0% of $R\text{-sq}(\text{pred})$ shows that the model cannot predict the response for new observations and is substantially less than $R\text{-sq}$ which indicates that the model is over-fit. Figure 4-8 is the visual representation of catastrophic response and is very unlikely to fit a model for DoE 1 on sandpaper P120. The current interpretation suggests that the test runs 18 and 19 are an arbitrary coincidence of correlation with the tire data.

Figure 4-8: Representation of chaotic response which is very unlikely to fit a model for DoE 1 on sandpaper P120.

4.4.2. Analysis of DoE 2: disc 180 counter-surface

DoE 2 is carried out and analyzed similarly as DoE 1, created with the same experimental design as DoE 1, performed on a LAT100 disc 180. Therefore, a comparison between the final models of the DoEs suggests comparability or contrast

between the counter-surfaces. Table 4-9 presents the friction coefficients of the 5 compounds and the response CC at each test run. Surprisingly, most of the experimental runs indicate that the response CC is above 0.9 and only 4 test conditions show a considerably lower value. However, interpretation steps of DoE are required to attain a trustable picture of influential factors on the correlation.

The Pareto chart of the Standardized Effects of a full quadratic model on DoE 2 depicts the significant parameters like speed, slip angle and a quadratic term of slip angle, see Figure 4-9. The normal plot in Figure 4-10 describes the quadratic term as a curvature with a maximum for the slip angle. The P-values of the ANOVA interpretation are shown in Table 4-10, the P-value larger than 0.05 is specified as the significant parameters which are in agreement with the Pareto chart and normal plot.

Figure 4-9: Pareto Chart of the Standardized Effects of the full quadratic model, DoE 2 on disc 180.

Table 4-9: The laboratory results of friction coefficients at each test run of DoE2 and the calculated response CC with LAT100 disc 180 counter-surface.

Run Order	Laboratory Coefficients of Friction (COF)					Response
	Comp A	Comp B	Comp C	Comp D	Comp E	CC
1	1.198	1.180	1.389	1.072	1.162	0.56
2	1.128	1.200	1.303	0.996	1.095	0.87
3	1.093	1.269	1.291	0.979	1.095	0.99
4	0.943	1.084	1.183	0.882	0.949	0.93
5	0.825	0.930	0.891	0.842	0.865	0.73
6	0.999	1.282	1.213	1.005	1.057	0.90
7	0.902	1.104	1.188	0.863	1.007	0.89
8	1.043	1.313	1.504	1.124	1.158	0.75
9	1.033	1.250	1.305	0.981	1.044	0.98
10	1.234	1.428	1.499	1.128	1.280	0.96
11	1.176	1.402	1.507	1.146	1.202	0.93
12	1.053	1.277	1.343	1.037	1.071	0.94
13	1.106	1.344	1.368	1.047	1.116	0.99
14	0.972	1.171	1.185	0.973	1.038	0.94
15	1.023	1.272	1.277	0.994	1.080	0.97
16	0.966	1.167	1.120	0.940	1.000	0.95
17	1.017	1.212	1.186	0.972	1.060	0.97
18	1.076	1.343	1.394	1.020	1.151	0.97
19	0.922	1.121	1.141	0.887	0.968	0.98
20	0.936	1.071	1.106	1.019	1.010	0.56

Figure 4-10: Normal plot of the Standardized Effects, DoE 2 on disc 180.

Table 4-10: ANOVA of DoE 1, response surface regression of the full quadratic model on sandpaper P120.

Source		DF	P-Value	P-Value < 0.05
Full quadratic model		9	0.018	Significant
Linear terms		3	0.007	Significant
Load	A	1	0.295	-
Speed	B	1	0.004	Significant
Slip angle	C	1	0.028	Significant
Square terms		3	0.052	-
Load*Load	A ²	1	0.526	-
Speed*Speed	B ²	1	0.347	-
Slip angle*Slip angle	C ²	1	0.010	Significant
2-Way Interaction terms		3	0.295	-
Load*Speed	A*B	1	0.139	-
Load*Slip angle	A*C	1	0.988	-
Speed*Slip angle	B*C	1	0.226	-

In the model summary, Table 4-11, the best-fitted model is an improved quadratic model in which two terms of AC and AA were excluded from the model. This model shows the lowest possible S and highest R-sq among others, with an improved R-sq(adj) and enhanced R-sq(pred). By reducing two terms of the full quadratic model, the R-sq(adj) is improved which suggests the model reduction was helpful to acquire better adjustment for the model terms. The model reduction made the gap between R-sq and R-sq(pred) smaller which enhances the chance of prediction for new observations, see 4.7.2. In total, the model summary table of DoE 2 carried out on LAT100 disc 180 offers better values compared with DoE 1 performed on sandpaper P120, see Table 4-8 and Table 4-11. The conclusion that can be drawn is the superiority of disc 180 on sandpaper P120 in the current study. However, the optimal test conditions for prediction of tire ABS data needs further analysis and investigation.

Table 4-11: Model summary of DoE 2 included with different model terms.

model	Terms omitted	DF	S	R-sq	R-sq(adj)	R-sq(pred)
Linear		3	0.105	46.8%	36.9%	14.1%
Linear + interactions		6	0.107	55.8%	35.4%	0.0%
Linear + quadratics		6	0.088	69.9%	56.0%	4.9%
Full quadratic		9	0.084	78.9%	59.9%	0.0%
Improved full quadratic	A*C, A*A	7	0.078	78.0%	65.1%	23.9%

The resulting predictive model of DoE 2 based on Table 4-11 last row – the terms load*slip angle and Load*Load were omitted – reads as follows:

$$\begin{aligned}
 CC = & 0.867 + 0.00097 \text{ Load} - 0.0273 \text{ Speed} + 0.02178 \text{ Slip angle} \\
 & + 0.00255 \text{ Speed} * \text{Speed} - 0.000661 \text{ Slip angle} * \text{Slip angle} \\
 & - 0.000483 \text{ Load} * \text{Speed} + 0.000517 \text{ Speed} * \text{Slip angle}
 \end{aligned} \quad (4.1)$$

The final step of the DoE interpretation in 4.7.1 was the analysis of the residual plots to determine whether the model meets its analysis assumptions. Figure 4-11 illustrates the residual plots of the model in Eq. (4.1). The probability plot of the residuals is distributed almost normally and approximately follows a straight line. A few points lying away from the line imply the measurement outliers. The residuals versus fits plot fall randomly on both sides of 0, with no recognizable pattern in the points which verifies that the residuals are randomly distributed and have a constant variance. A point far away from zero is an outlier and far away from the other points in the x-direction represents an influential point in the model. The last plot indicates that the residuals are independent of one another and no particular trends or pattern can be recognized in time order. Overall, it can be concluded that the model meets its assumptions. Hence, an optimal test condition can be extracted from this model.

Figure 4-11: Residual plots of DoE 2: residual normal probability, residual vs. fitted value, and residual vs. observation order; CC is the response.

The main effect of the factors on the responses according to the model is visualized in Figure 4-12. Slip angle shows curvature with a maximum, as was expected due to the quadratic term in the model. Speed impacts more on the response than load. The mutual interactions between the factors in the terms which were included in the model are illustrated in Figure 4-13. Parallel lines in an interaction plot indicate no interaction. The greater the difference in slope between the lines, the higher the degree of interaction. The Load*speed plot demonstrates a synergetic interaction indicating that load impacts more at larger speeds resulting in reducing the response. Slip angle and speed show not much interaction, the maximum point in the slip angle*speed graph varies in parallel graphs at a different speed. The interactions also replotted in the middle column of the matrix based on speed variation in the model. Figure 4-14 gives

3D representations of the response vs. the factors which show the main effects together with their interactions. It also visualizes where the optimal test conditions could be.

Figure 4-12: Main effects plot for the response CC, DoE 2 on disc 180.

Figure 4-13: Full matrix of the interaction plots for the terms in the model DoE 2 on disc 180. The vertical axis is the fitted mean of the response CC, the horizontal axis is Load (N), speed (km/h), and slip angle ($^{\circ}$). The holding values for each interaction plot of response vs. two factors in the matrix are given in the legends.

Figure 4-14: Response surface plots of the improved quadratic model of DoE 2; the hold values for the third factor in each plot of response vs. two factors are 5.05 km/h, 55 N, and 15 ° respectively.

4.4.3. Confirmation runs

After the optimization, a few confirmation tests should be run near the optimum settings. It is recommended to check the model at intermediate settings to assure that it does not deviate from the “true response”. Table 4-12 discloses that the responses measured and calculated from the confirmation runs fall in the desired range of the response predicted from the model as shown in Figure 4-12. Now the optimum test conditions are statistically specified and one example of the correlation is plotted in Figure 4-15. The repeatability of the measurements was also checked with ANOVA analysis, see 4.7.3.

Table 4-12: 5 Confirmation runs, values of response in comparison with the value from the predictive model.

Run Order	Load (N)	Speed (km/h)	Angle (°)	CC (from test runs)	CC (from model)
1	75	1	15	0.99	0.90 < CC < 1.00
2	55	5	25	0.92	
3	75	5	25	0.95	
4	65	2.5	15	0.98	
5	65	2.5	15	0.97

Figure 4-15: One example of the correlation between the ABS braking distance data and LAT100 friction coefficients; the example belongs to the test condition of the last row of Table 4-12.

These analyses provided substantial information to observe the variation of the response in a larger frame to select the optimal test conditions, than finding a coincidental correlation at a single test condition. The optimal test conditions of load and speed are in the similar ranges of a car tire. The mean contact pressure of a car tire of 0.3 MPa approximately is equivalent to the average pressure for the Grosch wheel. Considering around 1.5 cm² for the stationary contact area of the Grosch wheel (i.e. the wheel standstill) under load from 35 N to 75 N, the average pressure varies between 0.23 MPa and 0.5 MPa. This is the rule of thumb for a typical tread compound with average hardness. The slip speed of the tire is within 5% -15% slip ratio; indicating that at speed 90 km/h by applying ABS, supposing a 10% slip, the slip velocity is 9 km/h. Therefore, during the test, the tire experiences a distribution of slip velocities; in this example from 9 km/h until it comes to a full stop. Hence, considering a distribution of

slip speeds and contact pressures for the range of tread compounds with various hardnesses, the optimal test conditions which are deduced from DOE 2 are logical. In the present context with the locked Grosch wheel, the slip angle factor α ($^\circ$) is a machine parameter and mainly affects the sliding velocity magnitudes. Referring to section 4.3.3., the Grosch wheel slides over the counter-surface as a function of the rotation speed of the LAT100 disc, however, the sliding speed of the sample in the contact area is influenced by the disc radius which rises as the slip angle increases. This creates a distribution of sliding velocities around the traveling velocity of the LAT100 disc. Last but not least, by applying angle to the Grosch wheel, an effect of shape is introduced to the contact area which is also shown as an important influence [27] even at a similar nominal contact area [28]. The effect of all factors is more elaborated in the next section. Also, on the road during ABS, it is being dealt with as a distribution of factors. It is of high interest and value to envisage why such a vast range of test conditions correlate with the actual tire data that allow one to exploit the test set-up as a predictor for ABS braking distance. Would this correlation be valid for another set of variants? The next part is tailored to dive deeper into the underlying reasons for these questions.

4.4.4. Effect of speed and load

For each factor at the upper and lower levels of the designs, different phenomena are involved in the experimental set-up, which influence the friction measurements. The first factor load: by increasing the load the sample spreads more on the counter-surface since the test samples are solid rubber wheels as shown in Figure 4-1. Subsequently, the area of contact becomes larger and the pressure is not constant over the contact surface. The local pressure on the tips of the asperities of the counter-surface varies over the contact and is influenced by the form of the macro-pressure profile of the wheel sample as a whole. The second factor speed: since the rubber test wheel is blocked on the measuring hub of the device, the speed is the average sliding velocity in the contact area which is determined by the disc traveling speed. The speed range between the lower and upper levels of the experimental design is quite wide. The upper limit of 10 km/h is 100 times larger than the lower 0.1 km/h, which generates abrasion debris and a smeared layer of rubber on the disc, and plays an important role in the friction measurements. The relation between the coefficient of friction and the speed of the test specimen is non-linear; as Grosch showed the friction coefficient increases until it reaches a maximum at a specific speed and then decreases [29]: Figure 4-16. The influence of the friction peak should be taken into account to observe if it falls within the defined levels of factors in the designed experiments.

Figure 4-16: The master curve of friction coefficient vs. speed of ABR gum compound on a dry silicon carbide surface, taken with permission from ref [29].

Last, slip angle: the large range of slip angles introduces an additional impact on the shape of the contact area. The shape of the contact area changes if the orientation of the sample varies from the traveling speed direction of the LAT100 disc. Therefore, the leading edge can differ from a straight line to a pointy part of a lozenge. As a result, the edge impact should be taken into account. Although what an actual tire experiences in the contact between the tread blocks and the real road are combinations of various block orientations based on different tire patterns, the variation of slip angle in the present study overshadows the influence of speed and load and their interaction on the coefficient of friction.

Therefore, to look into the answers to the questions stated in the last paragraph of section 4.4.3, DoE 3 was created. A multi-level general factorial based on the results of DoE 1 and 2; at a constant slip angle of 15° , in 5 levels of speed, 3 levels for load, while disc 180 was chosen as the optimal counter surface: see Table 4-3 and Table 4-5. In Table 4-13, the measured friction coefficients for each compound, and the response CC are presented and sorted based on the design order. It should be noted that the experiment's run order was randomized. The CC values mainly fall in the desired range larger than 0.9 as expected from the predictive model of DoE 2 in Table 4-11. A deviation from the expected range of responses for the third run (design order) of the tests stands out as an outlier according to the residual plots presented in 4.7.4.

Table 4-13: The measured friction coefficients for each compound and the responses CC, sorted based on the design order, the run order was randomized.

Run Order	Factors			Laboratory coefficients of friction					Response
	Design Order	Load (N)	Speed (km/h)	COF A	COF B	COF C	COF D	COF E	CC
1	1	35	0.005	1.131	1.280	1.339	1.099	1.185	0.93
6	2	35	0.01	1.170	1.416	1.379	1.081	1.185	0.99
15	3	35	0.1	1.334	1.402	1.302	1.151	1.306	0.48
13	4	35	1	1.408	1.546	1.622	1.265	1.457	0.88
12	5	35	5	1.249	1.570	1.635	1.270	1.331	0.93
2	6	55	0.005	1.080	1.269	1.279	1.073	1.121	0.96
10	7	55	0.01	1.129	1.307	1.269	1.058	1.139	0.98
11	8	55	0.1	1.253	1.331	1.348	1.131	1.267	0.84
4	9	55	1	1.340	1.475	1.534	1.255	1.320	0.97
7	10	55	5	1.153	1.403	1.469	1.149	1.189	0.94
9	11	75	0.005	1.039	1.258	1.225	0.990	1.037	0.98
14	12	75	0.01	1.079	1.285	1.264	1.042	1.058	0.96
8	13	75	0.1	1.216	1.332	1.323	1.109	1.222	0.93
3	14	75	1	1.261	1.406	1.416	1.185	1.275	0.99
5	15	75	5	1.112	1.244	1.270	1.081	1.106	0.98

The coefficients of friction of each compound are also deemed as new responses separately to further observe the interaction of load and speed. For each compound, a predictive model was generated and validated according to the DoE interpretation steps as explained in sections 4.4.1 and 4.4.2. The model reduction summary of the compounds A-E to find the best-fitted ones are brought in 4.7.5. Based on the models for all compounds, the friction coefficients show a linear relationship with the load. But the speed factor B in the models contains the quadratic term BB and curvature is visible in the 3D graphs, see Figure 4-17.

Figure 4-17: 3D representation of the coefficients of friction COF vs. Load (N) and log speed (km/h) for the 5 compounds A-E: COF A, COF B, COF C, COF D, and COF E, respectively.

Figure 4-17 indicates that the compound dependence on speed varies by altering the normal load; with increasing load the peak becomes more pronounced. The peak at

the lower load is expected to appear at a higher speed. The variations of experimental friction coefficients vs. log speed for compounds A-E are plotted in the separate 2D graphs at different loads, see Figure 4-18. At 35 N, the plots are inclined towards a plateau at larger speed than 1 km/h for compounds B, C, and D, but not for compounds A and E. The trends of friction coefficient variation over the log speed axis for the 5 compound only shift vertically and are in parallel to one another without notable intersection between the curves. At 35 N and speed of 0.1 km/h, for compound C the sudden drop is the outlier of the experiment which was already pointed out in Table 4-13. This trend is more detectable in the intermediate range of speeds, higher than 0.01 km/h and lower than 5 km/h. This is in agreement with the results of DoE 2 and what the model proposed as the optimal range of test conditions to obtain the desired response in Figure 4-12. The parallelisms explain why a range of test conditions offers the best correlation with tire ABS braking distance data.

Figure 4-18: Variation of experimental friction coefficient vs. speed (km/h) in a log scale at loads 35 N, 55 N, and 75 N.

4.5. Concluding remarks

To predict tire friction, varying one factor at the time for experimenting does not necessarily provide a proper judgment. The research methodology of Design of Experiments (DoE) is implemented in the present study in which a predictive mathematical model is extracted to describe the relationship between responses and critical factors. Various operating parameters of the LAT100 as a tribometer on tire friction are investigated: load, speed, and slip angle. Measurements were performed on two counter-surfaces of a sandpaper- and a corundum-based substrate. The interaction of load and speed influences the coefficient of friction of tire tread compounds. The variation of the coefficient of friction vs. speed on the corundum-based substrate is in agreement with the ABS braking distance data of actual tires on the road, but not for the sandpaper substrate. The present study highly emphasized the importance of the design of experiments in a friction study. It gives guidance by a predictive mathematical model in selecting optimal test conditions regarding desired responses for present and also new observations, rather than just choosing arbitrary experimental settings.

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4.7. Appendices

4.7.1. DoE steps and interpretations

The typical steps of a DoE to identify all the possible factors that may influence the process to obtain an optimized response are as illustrated in Figure 4-19.

Figure 4-19: Typical steps in a DoE.

The DoE is carried out and analyzed using Minitab® 18.1 software. The steps and interpretations are extracted and summarized as a guideline for the reader [19-21] as follows:

Step 1: Pre-experiments formulate the problem and guide to decide which factors will be taken into account in the experimentation, and choose an adequate strategy for those not included.

Steps 2, 3: Create a screening design for numerous factors e.g. 6 to 15. A screening model is fitted which indicates the main effects to explore the most important factors. This phase is more an exploratory study and has a qualitative nature.

Steps 4, 5: Create a modeling design based on the results of the previous steps that include the most important factors e.g. 2 to 5. In this phase, the relationship between

responses and factors is described in sufficient detail. Fit a linear model that includes main effects and 2-way interactions and analyze the results and look for evidence of curvature in the relationship between the factors and the responses.

Steps 6, 7: If no curvature is detected in step 5, use that model to identify factor settings that optimize the response. If curvature is detected, add points for curvature to the design. Fit a quadratic model that includes square terms to model the curvature and analyze the results using Analysis of Variance (ANOVA) and Response Surface Modeling (RSM). ANOVA is a statistical test used to identify the effects of independent variables (factors) on the outcomes of an experiment (responses). RSM is a technique that models responses via polynomial equations and creates a map of responses in 3D surface graphs or 2D contour-plots. By using the final model, factor settings can be identified that optimize the response.

An example of the mathematical predictive model tested in ANOVA for 2 factors including interaction and quadratic terms reads as follows:

$$Y = \beta_0 + \beta_1 X_1 + \beta_2 X_2 + \beta_{12} X_1 X_2 + \beta_{11} X_1^2 + \beta_{22} X_2^2 \quad (4.2)$$

where,

Y : the predicted response;

X : input factor;

XX : 2-way interaction term;

X^2 : a quadratic term;

and β are the model coefficients; the subscriptions denote different factors.

Step 8: If an optimum is predicted using RSM, the results should be checked by confirmation runs. A few confirmation experiments near the anticipated optimum are recommended.

For **Interpretations**, four main steps are taken:

Step 1 examines the terms that contribute the most to the variability in the response using a measure called Pareto bar chart which can be obtained by ANOVA and diagnostics of the residual errors. This measure can be employed to rank the importance of the effect of each term in the predictive mathematical model. The Pareto chart is very useful especially for non-statisticians because it works with standardized effects; a dimensionless statistical scale that manifests the relative size of the effects for comparison. The dimensionless statistical scale is the t-value and is defined as the magnitude of the effect divided by the standard error. The following equations are employed for the calculations:

$$t \text{ value} = \frac{|Effect|}{\sqrt{MS_{\text{residual}} \left(\frac{1}{n_{\text{upper}}} + \frac{1}{n_{\text{lower}}} \right)}} \quad (4.3)$$

$$Effect = \frac{\sum Y_{\text{upper}}}{n_{\text{upper}}} - \frac{\sum Y_{\text{lower}}}{n_{\text{lower}}} \quad (4.4)$$

$$MS = \frac{SS}{df} \quad (4.5)$$

$$SS = \frac{(n_{\text{upper}} + n_{\text{lower}})}{4} \times (Effect)^2 \quad (4.6)$$

where the effect of each term can be computed according to Eq. (4.4); MS is the Mean Square, calculated according to Eq. (4.5) for the residual in Eq. (4.3); n is the total number of experiments in the subscript, the upper or lower level; Y is the result of the response in upper/lower levels; df is the degree of freedom defined as the number of independent pieces of information available to estimate a parameter [21]; SS is the Sum of Squares.

In the Pareto chart, the bars which cross the reference line are statistically significant. The value of the reference line is taken from the two-tailed t-table distribution which depends on the significance level denoted by α , see step 2. For example, the value 2.228 of the reference line of the Pareto chart in Figure 4-6 is taken from the statistical charts in Appendix 1.1 of the book 'DoE simplified' [21] at $\alpha = 0.05$ and $df = 10$ of the error or residual terms of the ANOVA table (page 68 of the book). Because the Pareto chart displays the absolute value of the effects, one cannot determine which effects increase or decrease the response. The normal probability plot of the standardized effects illustrates the magnitude and also the direction of the effects in one plot.

Step 2 determines whether the association between the response and the terms in the model are statistically significant using RSM and ANOVA. The terms: α and P-value are the 'significance level' and the 'probability of an event', respectively. α is the maximum acceptable level of risk for the rejection of a true null hypothesis⁶. The P-value is reported on a scale of 0 to 1 and from the computer program can be calculated to several decimals using F-statistics with Eq. (4.7). If the F-value is larger than the

⁶ A null hypothesis is a type of hypothesis used in statistics that proposes no statistical significance in a set of given observations.

critical F-value at the same df in the F-distribution table, the F-value is judged to be significant. The F-probability distribution can be found in the statistical tables of Appendix 1.3 of the book 'DoE simplified' [21]. The F-value (also called F-ratio or variance ratio) of the model can be defined assuming an equal sample size of n , as follows:

$$F \equiv \frac{nS_{\bar{Y}}^2}{S_{pooled}^2} \quad (4.7)$$

$S_{\bar{Y}}$ is the Standard Error (SE) of the mean response $\bar{Y}$ and holds as follows:

$$SE \equiv S_{\bar{Y}} \cong \sqrt{\frac{S^2}{n}} \quad (4.8)$$

S is the standard deviation and S^2 is the variance:

$$S = \sqrt{\frac{\sum_{i=1}^n (Y_i - \bar{Y})^2}{n - 1}} \quad (4.9)$$

S_{pooled}^2 is the pooled variance being estimated when there are several populations of data and the mean of each population might be different. It simply is the mean of the variances which are pooled together. Not only the model's significance but also the individual effect has to be checked for significance. Regarding the use of the F-distribution, note that in the calculation of the F-value, the df of the numerator and the denominator are different because they are referring to model and residual, respectively.

Using F-statistics for the Probability or P-value of a test: when the F-value is so large that the P-value falls below α , then with $(1-\alpha)\%$ confidence the term is affecting the measured response. In the present context, the P-value for each term is compared to the $\alpha=0.05$ to assess the null hypothesis. The null hypothesis implies that there is no association between the term and the response. $\alpha = 0.05$ works usually well and indicates a 5% risk of concluding that an association exists. Thus, the effect on the measured response occurs with 95% confidence level or more.

Therefore, if:

- P-value $\leq \alpha$: it can be concluded that the association between the response variable and the term is statistically significant.
- P-value $> \alpha$: it can be concluded that the association between the response variable and the term is NOT statistically significant. One may refit the

model without the insignificant terms which need further considerations of step 3 to carry out the model reduction.

If a model term is statistically significant, the interpretation depends on the type of term. In case a coefficient for a factor is significant, it indicates that alterations in these variables are associated with changes in the response variable. If a coefficient for a squared term is significant, the relationship between the factor and the response follows a curvature. And last, in case a coefficient for an interaction term is significant, the relationship between a factor and the response depends on the other factors in the term.

Step 3 determines how well the model fits the data. The goodness-of-fit is examined in a “model summary table” with S, R-sq, R-sq(adj), and R-sq(pred) statistics as follows:

$$S = \sqrt{MS_{residual}} \quad (4.10)$$

$$R^2 = 1 - \frac{SS_{residual}}{SS_{total}} = 1 - \frac{\sum_{i=1}^n (Y_i - \hat{Y})^2}{\sum_{i=1}^n (Y_i - \bar{Y})^2} \quad (4.11)$$

$$R_{adj}^2 = 1 - \frac{MS_{residual}}{MS_{total}} \quad (4.12)$$

$$R_{pred}^2 = 1 - \frac{SS_{PRESS}}{SS_{total} - SS_{blocks}} \quad (4.13)$$

Where the notations are listed as follows:

$\hat{Y}$: the predicted or fitted response;

$\bar{Y}$: mean response;

SS and MS according to Eqs. (4.4) and (4.5);

SS_{blocks} : SS in case of design with several blocks, in the present case only 1;

SS_{PRESS} : PRESS stands for Predicted Residual Sum of Squares (SS) being calculated by software: the model is repeatedly refitted to all the design points except the one being predicted. The difference between the predicted value and actual value at each point is then squared and summed over all points to create the PRESS.

- S represents the standard deviation, the variation of the data values from the fitted response surface. It assesses how well the model describes the response and can be used instead of the R-sq statistic to compare the fit of models; the lower the S value, the better the model. However, a low value of S alone does not indicate that the model meets its assumptions. The residual plots of step 4 should be checked to verify the assumptions.

- R-sq, also known as the coefficient of determination, is always between 0% and 100%, and one measure of how well the model fits the data. The higher the R-sq value, the better the model fits the data. R-sq is most useful when models of the same numbers of predictors are compared. Even when a model has a high R-sq, the residual plots of step 4 should be checked to verify that the model meets its assumptions.
- R-sq(adj) is used for model comparisons that have different numbers of predictors. The adjusted R-sq value incorporates the number of predictors in the model to help one choose the correct model because R-sq always increases when a predictor is added to the model, even when there is no real improvement to the model.
- R-sq(pred) shows how well the model predicts the response for new observations. Models that have larger predicted R-sq values have better predictive ability. A predicted R-sq that is substantially less than R-sq may indicate that the model is over-fit. An over-fit model occurs when terms are added for effects that are not important in the population. Predicted R-sq can also be more useful than adjusted R-sq for comparing models because it is calculated with observations that are not included in the model calculation.

Step 4 determines whether the model meets its analysis assumptions. To verify this:

- The probability plot of the residuals is required to be distributed normally and should approximately follow a straight line. The following patterns violate the assumption that the residuals are normally distributed.
- Residuals versus fits plot verify the assumption that the residuals are randomly distributed and have constant variance. Ideally, the points should fall randomly on both sides of 0, with no recognizable patterns in the points.
- Residuals versus order plots verify the assumption that the residuals are independent of one another. Independent residuals show no trends or patterns when displayed in time order.
- Patterns in the points may indicate that residuals near each other may be correlated, and thus are not independent. Ideally, the residuals on the plot should fall randomly around the centerline: If a pattern is seen, investigate the cause. The following types of patterns may indicate that the residuals are dependent, trend, shift, cycle.

Step 5 specifies whether the finalized model can predict the response for new observations. They are called confirmation runs.

4.7.2. The model reduction for DoE 2

Table 4-14: The model reduction for DoE 2, the response CC.

model	Terms omitted	DF	S	R-sq	R-sq(adj)	R-sq(pred)
Linear		3	0.105525	46.85%	36.89%	14.07%
Linear + interactions		6	0.106720	55.83%	35.45%	0.00%
Linear + quadratics		6	0.088090	69.91%	56.02%	4.89%
Full quadratic		9	0.084124	78.89%	59.89%	0.00%
Full quadratic	A*C	8	0.0802	78.89%	63.54%	10.42%
Full quadratic	A*C, A*A	7	0.078434	77.98%	65.13%	23.89%
Full quadratic	A*C, A*A, B*C	6	0.081150	74.46%	62.68%	22.14%
Full quadratic	A*C, A*A, B*B	6	0.0815002	74.24%	62.35%	30.97%

4.7.3. ANOVA analysis

An example of ANOVA analysis for the test repeatability:

Table 4-15: Five repetitions of test runs at a 15° slip angle, 65 N load, and 2.55 km/h speed for all compounds.

Run	Factors		Laboratory coefficient of frictions					Response
	Load (N)	Speed (km/h)	COF A	COF B	COF C	COF D	COF E	CC
1	65	2.55	1.115	1.320	1.388	1.107	1.165	0.93
2	65	2.55	1.195	1.316	1.369	1.069	1.109	0.92
3	65	2.55	1.131	1.318	1.357	1.057	1.099	0.98
4	65	2.55	1.104	1.330	1.364	1.040	1.115	0.99
5	65	2.55	1.094	1.331	1.366	1.043	1.097	0.98

mean: m ; Null hypothesis $H_0: m_1=m_2=m_3=m_4=m_5$; $H_1: m_1 \neq m_m$

$F_{stat} < F_{crit} \rightarrow$ Null is true

$F_{stat} > F_{crit} \rightarrow H_1$ is true (reject null) or P-value < 0.05 $\rightarrow$ reject null

Table 4-16: Summary of ANOVA analysis.

Groups	Count	Sum	Average	Variance
Column 1	5	5.7381	1.1476	2.96E-03
Column 2	5	6.6150	1.3230	4.83E-05
Column 3	5	6.8445	1.3689	1.39E-04
Column 4	5	5.3153	1.0631	7.26E-04
Column 5	5	5.5858	1.1172	7.53E-04

Source of Variation	SS	df	MS	F	P-value	F crit
Between Groups	0.3597	4	0.09	97.13	8.31E-13	2.87 (taken from F distribution)
Within Groups	0.0185	20	0.00			
Total	0.3782	24				

4.7.4. Residual plots of DoE 3

Figure 4-20: Residual plots of DoE 3, the outlier is specified.

4.7.5. The model reduction summary of the compounds A-E

Figure 4-21: Pareto chart and normal plot for compound A for DoE 3; the response is COF and $\alpha = 0.5$.

Table 4-17: The model summary for compound A; the selected model was marked in grey.

model	Terms omitted	S	R-sq	R-sq(adj)	R-sq(pred)
Full quadratic		0.0374088	92.40%	88.17%	80.37%
Full quadratic	A*A	0.0358694	92.23%	89.13%	83.93%
Full quadratic	A*A, A*B	0.0357014	91.54%	89.23%	84.98%

Compound B:

Figure 4-22: Pareto chart and normal plot for compound B for DoE 3; the response is COF and $\alpha = 0.5$.

Table 4-18: The model summary for compound B; the selected model was marked in grey.

model	Terms omitted	S	R-sq	R-sq(adj)	R-sq(pred)
Full quadratic		0.0523464	83.37%	74.13%	49.92%
Full quadratic	A*A	0.0506269	82.72%	75.81%	57.64%
Full quadratic	A*A, B*B	0.0546323	77.86%	71.83%	50.57%

Compound C:

Figure 4-23: Pareto chart and normal plot for compound C for DoE 3; the response is COF and $\alpha = 0.5$.

Table 4-19: The model summary for compound C; the selected model was marked in grey.

model	Terms omitted	S	R-sq	R-sq(adj)	R-sq(pred)
Full quadratic		0.0637445	84.35%	75.66%	62.66%
Full quadratic	A*A	0.0552638	86.98%	81.77%	66.91%
Full quadratic	A*A, B*B	0.0596103	83.28%	78.72%	73.28%
Full quadratic ⁱ A*A		0.0449955	91.80%	88.52%	80.69%

ⁱ the outlier is removed.

Compound D:

Figure 4-24: Pareto chart and normal plot for compound D for DoE 3; the response is COF and $\alpha = 0.5$.

Table 4-20: The model summary for compound D; the selected model was marked in grey.

model	Terms omitted	S	R-sq	R-sq(adj)	R-sq(pred)
Full quadratic		0.0428592	83.32%	74.05%	51.16%
Full quadratic	A*A	0.0408101	83.19%	76.47%	60.07%
Full quadratic	A*A, A*B	0.0418233	80.59%	75.29%	61.69%

Compound E:

Figure 4-25: Pareto chart and normal plot for compound E for DoE 3; the response is COF and $\alpha = 0.5$.

Table 4-21: The model summary for compound E; the selected model was marked in grey.

model	Terms omitted	S	R-sq	R-sq(adj)	R-sq(pred)
Full quadratic		0.0475569	88.84%	82.65%	69.81%
Full quadratic	A*A	0.0454323	88.69%	84.16%	74.05%
Full quadratic	A*A, A*B	0.0459566	87.27%	83.80%	75.19%

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ORIGINAL PAPER

A New Horizon for Evaluating Tire Grip Within a Laboratory Environment

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Abstract

The evaluation of tire grip on the road is costly and consumes high energy and time, but is essential for safety. Prediction of tire grip on a laboratory scale is therefore always of interest and of utmost importance for research and material developments. It mostly suffers from lack of comparison with actual tire data. To involve all influencing factors on tire grip in a laboratory-scale measurement is very complex. Therefore, it has always remained challenging to obtain a strong correlation between laboratory results and road data.

In the present study, a new test method is developed for a Laboratory Abrasion Tester, LAT100, which enables to exploit the device as a tribometer. The objective was to develop a technique on a laboratory device to mimic the common test modalities for evaluating tire grip on the road with a trailer tester: lateral (α) and longitudinal (κ) sweep tests. The new method is validated by correlating the laboratory data with the two test modalities of real tire grip on a dry road using a trailer tester for six different tire tread compositions. For the LAT100 tests, solid rubber wheels are characterized at three different normal loads. The effects are comparable with actual tire data. The outcome of the new test method is in good agreement with actual tire trailer α -sweep tests.

Keywords: Tire dry grip, Tire cornering, Rubber friction, LAT100

⁷ A self-contained chapter: the content of the paper has been transformed completely into this chapter.

5 New sweep function with LAT100 in correlation with tire data

5.1. Introduction

Tire grip or traction is a concept that describes the grasp and interaction between the tire and the road to avoid vehicle skidding or sliding which is crucial for safety. The terms grip or traction both imply the same meaning and can be used interchangeably in this context. Proper tire grip provides a good level of handling which is a prerequisite for vehicle's steering in various driving states such as cornering, braking, and accelerating. The tire grip is the result of the generated frictional forces in the aforementioned driving states which are created by the tire slippage in the contact patch. When a vehicle brakes, accelerates, or corners, the tire tread elements in the contact area move a noticeable displacement in relation to the road. This is called slippage. For instance for an Anti-lock Braking System (ABS), the braking force is optimized based on a specific region of tire slippage to provide adequate frictional forces or grip and avoid tire skidding. This has been elaborated in detail in our previous studies [1, 2]. A tire rolling at a constant speed does not move relative to the road surface in the contact patch, apart from the shear deformations owing to various forms

of rolling resistance and carcass (the tire internal structural body except for the tread layer) distortion which creates a very small slippage in the trailing edge. This is beyond the scope of this paper.

Numerous factors influence tire grip: first, the tire construction, tread formulation and the pattern of the tire; second, the road surface characteristics, its condition and structure; and last temperature [3]. The general relationship between the frictional forces and the slippage is a specific characteristic of a tire design that includes the carcass, all rubbers used in the tire structure, and most important the tread. The slippage can be classified in lateral or longitudinal directions based on the driving states. Accordingly, it creates forces that can be categorized in two main modes: lateral (side) during cornering and longitudinal while braking or accelerating, both with their own specific characteristics.

The lateral force is the force that tires transmit to the ground during cornering in parallel to the rolling axis, see Figure 5-1. The angle between the tire plane of symmetry and the actual moving direction of the vehicle in cornering is termed slip angle α ($^\circ$) which creates lateral slippage. Figure 5-1 (left) shows a typical relationship between the lateral friction coefficient μ_y and the slip angle. Various lateral forces play a role during tire service which differ by several orders of magnitude. They encompass cornering forces, camber thrust, and minor residual influences comprised of two components - ply steer (pseudo-slip angle) and conicity (pseudo-camber) [4].

The longitudinal friction force is created when the tire contact patch moves a noticeable displacement in the forward direction during braking or accelerating. The longitudinal slippage is defined by the difference between the forward driving speed and the equivalent circumferential velocity of the wheel [5, 6]. According to the Society of Automotive Engineers (SAE), the longitudinal slip ratio κ is defined as:

$$\kappa = - \frac{V_x - R_e \omega}{V_x} \quad (5.1)$$

where V_x (m/s) is the longitudinal driving speed and $R_e \omega$ is the equivalent circumferential velocity of the wheel, where ω (rad/s) is the rotational velocity of the wheel and R_e (m) is the effective tire radius. The latter is defined as the radius of the tire when rolling with no external torque applied to the spin axis. Since the tire flattens in the contact patch, this value lies somewhere in between the tire's un-deformed and the static loaded radii [7]. Figure 5-1 (middle) shows the longitudinal friction coefficient μ_x vs. the negative braking slip ratio according to Eq. (5.1). Whereas all these phenomena also occur during accelerating conditions, then conversely the slip ratio is

positive. Each tire is characterized by a specific friction curve based on its carcass, tread pattern as well as the composition of the various rubber compounds. The typical important parameters of the friction curves are specified in Figure 5-1. The slope of the lateral friction at small slip angles, typically smaller than $2^\circ - 3^\circ$ for a tire where the friction curve is still linear, represents the cornering stiffness of the tire which originates from tire carcass and tread stiffness.

Accordingly, there are two common methods for assessing tire dry grip. First, lateral (α) sweep in which the α is varied over one complete cycle of slip angle at a specific normal load to the counter-surface at steady speed and a defined camber angle. Second, longitudinal (κ) sweep which is the variation of κ , while maintaining the α equal to zero. The degree of κ is continuously varied by applying a braking force to the wheel. Combined slip is also possible which involves variation of the κ and α simultaneously [8]. During tire tests the lateral and vertical forces and self-aligning torques are collected and by employing the Magic Formula (MF) of Pacejka [9] the data are modeled and presented in the form of the typical graphs as shown in Figure 5-1. It is possible to perform the measurements at various normal loads and camber angles.

At automotive and tire industrial companies and institutes, indoor and outdoor test facilities are available for performing full-scale tire measurements to determine tire performances. The outdoor test installations are built on a truck or trailer that is equipped with a special wheel suspension and guidance system to which a measuring hub is attached. Flat track tire testing machines also evaluate tire performances in an indoor laboratory on a full tire on corundum sandpaper. The design of such systems presents a high degree of complexity in order to control the flatness of the rolling surface as well as its dynamic performance to keep it from interfering with the tire [10]. The measurements have a higher precision in comparison with the tire trailer testing because of controlled laboratory environment and conditions, but less accuracy due to the type of test track [11]. At an actual range of loads and speeds of the tire, the flat track tests are not in good agreement with the trailer testing due to the sandpaper substrate which normally becomes saturated with abraded rubber particles which fill out the asperities of the test track. The possibilities of applying a simulated road in the indoor devices have been extensively discussed to obtain proper friction properties [12].

Figure 5-1: The general relationships between the frictional forces and the slippage: (left) lateral and (middle) longitudinal braking modes; (right): a sketch of a rolling wheel as a guide for the normal, lateral, and longitudinal directions.

For material development concerning tire performance, both indoor and outdoor tire tests are enormously sophisticated, time-consuming, and costly. It would be highly desirable to predict tire grip performance in a laboratory environment before manufacturing a full tire. Therefore, it is important to evaluate the friction properties which reflect the actual tire grip in a laboratory environment. The evaluated friction properties of the tire tread compound can also be employed as input data for models and simulations for tire characteristics. The quality and parametrization of the tire models and simulations strongly depend on the correct rubber friction properties [13-17]. Tires are built up of a multitude of different elements, mostly rubber compounds, each contributing to the overall performance profile. This complex structure interacts with the road as counter-surface. Therefore, a wide range of factors influences realistic tire grip prediction with determination of the rubber friction in a laboratory environment. Considering all these factors in one laboratory device is a complex task to accomplish.

Many researchers implemented theoretical approaches to describe, understand and calculate friction with respect to influencing parameters such as temperature, rubber formulations and its properties and surface characterization [18-25]. The friction behavior has also broadly been investigated using modeling and simulation by taking parameters such as speed, load, and temperature into account in order to obtain insight into transient friction curves which are difficult to acquire in a laboratory environment [9, 13, 15, 17, 26-29]. On the other side, a large number of testing measurement systems exist that can be classified mainly in two different test set-ups: linear and rotary which are based on the specimen geometry and its relative motion to the counter-surface. The results are commonly expressed in terms of the coefficient of friction [14, 22, 30-38].

ISO 15113 standard [39] for the frictional properties of rubber was first published in 1999 and is probably the most comprehensive of all friction standards. It was developed from an earlier British standard, BS 903 Part A 61 [40]. The standard methods do not describe a specific apparatus but emphasize the importance of tight control of the various parameters and provides guidance on the factors to be considered in measuring friction. In most mentioned test procedures, the objective is to provide the best correlation with the service conditions, together with good reproducibility between laboratories.

Among all available tribometers suitable for measuring tire tread frictional properties, there is a great interest in measurements with the Grosch wheel [1, 2, 13-17, 28, 31, 41-48]. The main advantage is the rolling movement in the contact area with a renewed surface in each revolution. This helps the rubber to relax in a loading condition similar to a tire which leads to a more realistic temperature profile in the contact area. Using sliding body measurements does not provide a realistic input for tire modeling due to poor surface cooling [13]. Some researchers put more emphasize on the dynamic contact behavior of the rolling wheel because of similarities to the tire by the aid of modeling [13-17].

The Laboratory Abrasion Tester (LAT100) is one of the machines that is able to simulate a wide range of test conditions in a laboratory environment with a solid rubber sample wheel which was originally developed by Grosch in 1996 [46] to measure the lateral force and the abrasion loss as a function of various slip angles, loads, and speeds on different counter-surfaces. This apparatus has also been considered for the evaluation of rubber tribological properties due to its similarity to vehicle tire operating conditions [1, 28, 31, 41-46, 48].

For tire grip assessments, so far most attention has been paid to car ABS braking distance data, especially on wet surfaces. Good agreements have been acquired with laboratory tribometers, both linear and rotary measurements, with ABS braking data. A Linear Friction Tester (LFT) provides a strong correlation with outdoor ABS data on wet [49] and snow surfaces [50], but not for dry surfaces primarily because of poor temperature control in the contact area [49]. The portable LFTs provide the possibility to perform in-situ measurements on the actual testing fields or road surfaces. Some good agreements have been reported between LAT100 results and ABS braking data on different surfaces [1, 31, 48].

In the present study, the prediction of tire grip on a dry surface is investigated in a laboratory environment with a LAT100, compared with the friction curves acquired from actual tire grip data in the two driving states of cornering and braking. Basically,

the objective was to apply and simulate α - and κ -sweep tests of the trailer tire testing in the laboratory device. A new test method is developed for the LAT100 by which the instrument can be utilized as a tribometer to generate the full friction curve. New dynamic functions are defined which enable one to work in a semi-automatic mode of the LAT100 machine. This allows to program the LAT100 parameters: slip angle, load, and speed with time. The test method is designed in such a way that it needs substantially shorter test times in comparison with the conventional measuring method of the LAT100. This method is verified and validated with actual tire data on a dry road, evaluated with a tire trailer tester with two test modalities of κ and α sweeps for a wide range of tread compounds. Corresponding solid rubber wheels are characterized on the modified LAT100 on a specific electro-corundum counter-surface with a particular grain size based on our earlier study [1], disc 180 of the LAT100 (see later). The experiments are performed at three different normal loads and the effects are compared between laboratory and tire samples. The results are in a good agreement primarily with the actual tire trailer α -sweep tests.

The advantage of the current study is first the comparison and correlation of laboratory data with the actual tire data on the road, and secondly the design of a unique method analogous to the tire situation on a laboratory scale for the evaluation of the rubber properties and the acquisition of full friction curves. This method also reduces the change of the rubber sample surface during the measurement which stems from the inherent rubber abrasion during the measurements for acquiring the friction curves. The test results are comprehensively validated for a large variety of rubber compounds and test conditions.

5.2. Experimental

5.2.1. Tire data generation

5.2.1.1. Tread compounds

Six passenger tread compounds C1 to C6 were designed comprising various types of carbon black and silica fillers with different particle sizes; C1 to C3 were pure silica reinforced, C4 was based on a blend of silica and carbon black and C5 and C6 on pure carbon black. This was expected to provide a significant difference in tire grip. The reason for this wide selection was to strengthen the laboratory method validation in the current study.

To prepare full tires from experimental compounds for tire trailer testing on the road, large amounts of such compounds are needed. Each compound needs to be extruded to the green tread dimension according to its specification. Considering the different rheological behaviors of the compounds, various die designs would be required. Consequently, it is not time- and cost-effective to extrude the full treads in a tire production line. Therefore, experimental tires were prepared via treads which were prepared in a laboratory environment. To prepare these treads, the rubber compounds were calendered and rolled into the particular width and assembled according to the specification of the tire size 235/35R19 slick. Full-size tires were manufactured with the same construction process using the same body components, but with the prepared treads. To compare tire data and calculate the rating, an additional proprietary reference compound was used and considered as the standard tire.

5.2.1.2. Tire tests

The two types of tire tests that were performed were κ and α sweep tests as explained in section 5.1, on an asphalt testing field with a test trailer. The tests were carried out at a constant speed of 60 km/h and at three different normal load settings: 3430 N, 4910 N and 6870 N (corresponding to 350 kg, 500 kg, and 700 kg, respectively). The slip condition depending on the type of test mechanism differs. The α -sweep tests were performed from -12° to $+12^\circ$ slip angle with a slip rate of $2^\circ/\text{s}$. The κ -sweep tests were performed at a constant slip angle α of 0° and the κ varied in a range of 0 to -0.45 in the braking mode. For α -sweep tests the lateral forces F_y and for κ -sweep tests the longitudinal friction forces F_x were collected. The experiments were executed randomly for all the tires with the various compounds with 3 times repetition for the reference tire. The camber or inclination angles in these experiments are less than 0.5° , which is negligible.

5.2.2. Laboratory data generation

5.2.2.1. Test set-up

The used laboratory set-up is known with the commercial name LAT100 manufactured by VMI Holland B.V., the Netherlands. The details of the test set-up were fully described in our previous manuscript [1]. The measurement unit consists of a solid rubber test wheel which is pressed under a vertical load of F_z (N) normal to a driven disc as the counter-surface. The circumferential velocity of the test wheel V_c is induced

by the disc traveling velocity, $V_t = R * \omega$ where R is the distance of the disc center to the center of the rubber wheel contact patch and ω (rad/s) is the rotational velocity of the disc. The average slip velocity V_s in the contact area is created by combining traveling velocity and slip angle α , which for the center point of the test wheel in contact with the disc is given by:

$$V_s = V_t \sin \alpha \quad (5.2)$$

The resultant forces are generated by the slip velocity and the acting normal force. A schematic view of the measurement unit and the trigonometry of the velocities and resultant forces in the center of the contact patch: F_c counter-centrifugal (radial) force, F_y lateral force, and F_x tangential friction force are represented in Figure 3-4. The coordinate system is considered to be linked to the center of the rubber wheel.

Figure 5-2: Schematic views of the measurement unit of the LAT100.

5.2.2.2. New test design

Within this study, a new test method was designed for the aforementioned test set-up. The conventional LAT100 could only be operated with static set-points for the input parameters – disc traveling speed, normal load, slip angle – which had to be defined and imported in the machine software before each test run. The software was modified so as to be able to control the machine in a dynamic way in which the input parameters vary during the test runs based on defined functions to specified set points. The input parameters change vs. traveling distance or time with a pre-defined mathematical function with which the tire conditions on the road can be transformed to the rolling rubber wheel. For instance, Figure 5-3 shows two examples of different dynamic functions of the modified LAT100 software; (A) slip angle as a triangular function of distance; and (B) speed as a ramp function of distance. Any combinations of the three input parameters with the dynamic functions are an option. It is also possible to program the test set-up in a way which allows one to simulate non-stop runs of different test conditions on one rubber test wheel. This allows to simulate the actual tire behaviors on the road in a series of test conditions and evaluate the rubber compound properties in a set of functions and test conditions resembling real tires. The test time is greatly decreased and the experiments can be performed within minutes in comparison with at least one full day with the conventional LAT100 method. It also reduces the change of the rubber sample surface which is caused by abrasion during the measurements while acquiring the friction curves.

Figure 5-3: Modified LAT100 software illustrating: (A) a triangular function of slip angle (°); and (B) a ramp function of speed (km/h) vs. distance (m).

5.2.2.3. The test track

The employed laboratory test disc was based on our previous study [1] and provided by VMI Holland BV Tire Industry Equipment. It is composed of electro-corundum white Al_2O_3 powder bound in a ceramic binder with a weight ratio of 85%/15%. The grain size distribution of the used powder is the percentage of individually sized particles contained in the designated grit sizes and classified with the code 180 according to ISO525:2013 (E), as given in Figure 5-1. The corundum particles in the ceramic binder are pressed together and the disc surfaces flattened by a grinding process using fine steel powder. The surface root mean square S_q of the disc roughness was measured on an area of $1414\text{ }\mu\text{m} \times 1080\text{ }\mu\text{m}$ as given in Figure 5-1; A microscopic image of the disc surface was obtained with confocal laser scanning microscopy VK 970 Keyence.

Table 5-1: Characterizations of the laboratory test track: disc coded 180 of LAT100.

Disc designation	180	Microscopic image
Disc code	EKW 180 SHARP
Particle size (μm)	60	
	0%>125	
Corundum sieve analysis (μm)	Maximum 15%>90	
	Minimum 40%>63	
	Minimum 65%>53	
S_q (μm)	65	

5.2.2.4. The rubber test wheel samples

From all tire tread compounds, laboratory rubber test wheels with external diameter of 84 mm and thickness of 19 mm were vulcanized in a Wickert laboratory press WLP 1600 for 11 minutes at $170\text{ }^\circ\text{C}$ under a pressure of 160 bar, optimized on basis of t_{90} curemeter data. Finally, the samples were stored for 24 hours at room temperature, $22\text{ }^\circ\text{C} \pm 1\text{ }^\circ\text{C}$. Therefore, the initial temperatures of all samples were equal when starting the laboratory experiments.

5.2.2.5. Test conditions

For realistic and repeatable evaluations, some pre-conditioning test runs on a brand-new rubber sample and disc track were performed to remove the shiny outer vulcanization skin from the rubber wheel samples and level the sharp asperities of the experimental track, like the test conditions described in [1]. After every single test for each slip angle, disc cleaning was performed using an air pressure nozzle which removed the rubber debris and particles from the disc surface. The linear function of the slip angle vs. traveling distance of the LAT100 is the closest operating test condition compared to the described tire tests in section 2.1.2. Hence, for the angle rate ($^{\circ}/s$) which is the rate of slip angle α change with time or rather with distance traveled, the following relation with the conventional parameters of the LAT100 holds:

$$\text{angle rate } (^{\circ}/s) = \frac{\text{Slip angle } (^{\circ}) * \text{Speed } (m/s)}{\text{Distance } (m)} \quad (5.3)$$

The lateral forces were evaluated under the applied normal load, slip angle, and traveling velocity V_t to determine the friction curve which is represented as μ_y according to:

$$\mu_y = \frac{F_y (N)}{F_z (N)} \quad (5.4)$$

The ranges of speeds and loads as covered within the defined test conditions are listed in Table 5-2. The traveled distances are calculated according to Eq. (5.3) for a constant angle rate of $1.53^{\circ}/s$ for each test condition. $1.53^{\circ}/s$ of the angle rate is the optimized value based on the available machine setting to obtain a full friction curve within the range of defined variables. The speed range was based on “scaling down” the angular speed of the tested tire size (235/35R19) to the laboratory test wheel [51]. The speed, slip angle, and angle rate was calculated to get close to the tire service condition in the contact area using this scaling model. The ratio of the tire to the test wheel circumferences is around 8. Therefore, the equivalent speed for the test wheel was estimated to be of order 8 times smaller compared to a speed of 60 km/h for the tire tests. Therefore, a range of 4 speeds from 0.2 km/h to 11 km/h was employed.

Table 5-2: Experimental conditions on disc 180 in laboratory device.

Parameters	Experimental conditions
Loads (N)	55 , 75 and 95
Speeds (km/h)	0.2, 2.1, 6.5, 11
Distance (m)	0.91, 9.5, 29.5, 50
Angle rate (°/s)	1.53
Slip angle (°)	Sweep from 0 to 25

5.3. Results and discussion

5.3.1. Tire test data

5.3.1.1. κ -sweep tests

The tire μ_x value, see Figure 5-1, is obtained by dividing the longitudinal braking friction force by the normal load F_x/F_z . Table 5-3 shows the tire μ_x ratings at Peaks of the friction curves in the κ -sweep tests; the Peak point is identified in Figure 5-1. The compounds are rated based on the average values of the reference compound. The rating of the reference compound is set at 100%. The data are presented for the three different applied loads. The Coefficients of Variation (CV) - the ratio of standard deviation to the mean - of the reference compound were calculated corresponding to each load setting as also shown in Table 5-3 to provide an impression of the repeatability of the tests. An example of a κ -sweep friction curve at 350 kg for the tested compounds is illustrated in Figure 5-4 (left) which is the normalized tire μ_x versus the slip ratio κ . All the μ_x values are normalized by dividing by the largest μ_x value of the reference compound and are expressed in percentage.

Table 5-3: Tire μ_x ratings at Peaks of the κ -sweep friction curves for different loads.

	Tire μ_x at Peaks		
	350 kg	500 kg	700 kg
C2	101.4	97.0	98.4
C3	102.4	100.3	100.4
C1	100.7	98.7	96.1
Average of all Refs	100.0	100.0	100.0
C4	97.9	96.0	96.0
C5	93.8	92.8	89.6
C6	92.8	90.0	90.3
CV (%) Ref	2.2	2.1	2.1

Figure 5-4: Normalized tire friction curves of the trailer tests at 350 kg load, (left) κ -sweep and (right) α -sweeps.

5.3.1.2. α -sweep tests

Tire μ_y is obtained by dividing the lateral force to the normal load F_y/F_z versus slip angle $\alpha(^{\circ})$. Figure 5-4 (right) shows an example at 350 kg load of the α -sweep friction curves for the tested tread compounds. All the μ_y values are normalized as carried out for the μ_x values, by dividing by the largest μ_x value of the reference compound. By this approach the magnitudes of the normalized μ_y are comparable with the normalized μ_x values. The data including slope S at 0° , the minimum tire μ_y value (tire μ_y at Min.), the maximum tire μ_y value (tire μ_y at Max.) of the α -sweep friction curves, and the CVs for the reference compounds are represented in Table 5-4.

5.3.2. Laboratory results with the modified LAT100

The generated slip velocity with the LAT100 is based on varying the slip angle. Previously [1], the friction curves were provided in a static mode where the traveling speeds, loads, and slip angles remained constant during a test run. In the present study, the experiments were carried out by varying the slip angle as a single dynamic linear function of distance based on the new test modality. This present test method is similar to the tire α -sweep test which provides a higher control of the abrasion and surface temperature of the test wheel in comparison with the single point measurement method. The test time is greatly decreased and the experiments can be performed within seconds. An example of the generated friction curves by the newly designed

modality is presented in Figure 5-5. The best way for parametrization of the friction curves is applying models like the MF of Pacejka [9]. Data filtering was performed using Matlab R2017b by exploiting smoothing spline functions: see Figure 5-5. In mathematics, a spline is a function defined piecewise by polynomials. The spline is preferred over polynomial interpolation because it yields a good fitting. Accordingly, the slip angle position on the x-axis at Peak, μ_y value at Peak, and at last the slope S at small slip angles which can be derived by a linear regression below 5° for LAT100.

Figure 5-5: Example of a friction curve prepared by the modified LAT100 for C2 at speed 2.1 km/h and load 55 N; dashed line: the linear regression at small slip angles, arrows: peak position on slip angle axis and the corresponding μ_y value.

The repeatability of the laboratory sweep tests was also investigated. The CV for the μ_y at Peak for the reference compound was measured as 1.2 % which is comparable to the tire data: see Table 5-3 and Table 5-4. At higher slip angles e.g. 25° the abrasion and high temperature create 1% more variation in the laboratory results. All sweep experiments with the modified LAT100 were performed under the test conditions mentioned in Table 5-2. After data smoothing for the output friction curves, the results were correlated with the collected tire data, see section 5.3.4. Examples of smoothed μ_y vs. slip angle (°) for the modified LAT100 for the various tested compounds at speed 6.5 km/h and 55 N load are presented in Figure 5-6.

Table 5-4: Tire μ_y ratings at Min. and Max. points and the slope S at 0° of the tire α -sweep friction curves for different loads.

	Tire μ_y at Max.	Tire μ_y at Min.	S (slope) at 0°
350 kg			
C2	92.8	92.9	76.5
C3	101.1	100.2	84.4
C1	93.3	94.0	85.0
Average of all Refs	100.0	100.0	100.0
C4	101.0	102.5	100.4
C5	101.3	101.3	116.1
C6	97.3	99.2	119.4
CV (%) Ref	1.0	0.5	2.0
500 kg			
C2	92.7	93.5	74.9
C3	101.0	98.3	79.9
C1	93.3	93.3	80.2
Average of all Refs	100.0	100.0	100.0
C4	99.2	99.3	93.8
C5	97.4	98.5	109.7
C6	95.2	98.4	111.2
CV (%) Ref	1.6	0.1	3.2
700 kg			
C2	93.0	91.9	78.6
C3	101.1	99.2	82.9
C1	92.9	91.5	85.0
Average of all Refs	100.0	100.0	100.0
C4	102.2	100.8	95.4
C5	103.1	100.4	100.1
C6	97.2	98.3	114.2
CV (%) Ref	0.4	0.5	3.1

Figure 5-6: Examples of laboratory μ_y vs. slip angle ($^{\circ}$) at speed 6.5 km/h and 55 N load for all compounds in the modified LAT100.

5.3.3. Tire data analysis

Before validating the new technique and correlating the laboratory results with the actual tire data, it is necessary to investigate the relation - if any - between the two different test modalities of α - and κ -sweep of the tire data for all 6 tread compounds. Both test modalities were outdoor types and the friction behaviors were measured on full-scale tire interfaces with the same road asphalt surface. Therefore, the comparison between the acquired tire data demonstrates the difference in the two tire test modalities. As discussed in section 5.1, the differences between these test modalities are the slippages based on different displacement types and directions of tread elements: longitudinal and lateral, with their generated forces. For the κ -sweep test, the slip ratio is calculated based on Eq. (5.1) and for the α -sweep the slip angle is considered.

In order to check eventual correlations, the tire μ_y values at the Max. and Min. points of the α -sweep curves were plotted on a linear scale vs. the μ_x values at Peak of the κ -sweep curves, as derived from Figure 5-4. The regression coefficients R^2 were calculated for different normal loads. The R^2 values for the correlations between α - and κ -sweep tests are very poor: see Table 5-5. This indicates that the compound with the superior longitudinal grip in the κ -sweep test does not offer the best behavior in the lateral direction of the α -sweep test: see Figure 5-4. It suggests different contributions of friction mechanisms: adhesion and hysteresis, that govern the contact between the

tire and asphalt. It is therefore worthwhile to investigate how the different test mechanisms adversely affect the grip properties of longitudinal and lateral slippage mainly after the peak. It demonstrates that the two test modalities need to be looked at each on their own. This will be covered in a separate manuscript.

Table 5-5: Regression coefficients R^2 between the tire data ratings at three different tire loads.

Tire loads	350 kg	500 kg	700 kg
R^2 between κ -sweep μ_x at Peak & α -sweep μ_y at Max.	0.08	0.11	0.03
R^2 between κ -sweep μ_x at Peak & α -sweep μ_y at Min.	0.20	0.02	0.04

5.3.3.1. Symmetry of tire α -sweep tests

In order to properly analyze the tire data, it is important to judge whether the α -sweep curves are symmetrical. The tire μ_y values in the mirrored α -ranges: negative and positive slip angles, per each load for all compounds were correlated. Figure 5-7 shows an example of the correlated μ_y values at the positive slip angle vs. the negative slip angle for the reference compound at load 350 kg. If the curves were perfectly symmetrical, the R^2 values should be 1.0. The R^2 value of 1.0 value before the Max. and Min. points of the friction curves demonstrates full symmetry. Given the uncertainties in performing the tests beyond the Max. and Min. points where sliding occurs, the curves are not fully symmetrical; although the ratings of the respective compounds in the positive and negative ranges of slip angle are still correlated: see Figure 5-7 (a).

Another approach is to correlate the tire data, comparing the μ_y values for different applied normal loads. The correlation coefficients R^2 are given in Table 5-6, which demonstrates that even with different loads the friction behaviors remain reasonably well correlated.

Table 5-6: Correlation coefficients R^2 between μ_y values at positive and negative slip angles for different loads.

Loads (kg)	R^2
350, 500	0.90
350, 700	0.91
500, 700	0.89

Figure 5-7: Tire μ_y at negative vs. positive slip angles for the reference compound at load 350 kg;(a) for μ_y before and after Max. (b) for μ_y before Max..

5.3.3.2. Tire temperature

The tire temperature profiles in a half circle sweep from -12° to 12° for different applied loads are shown in Figure 5-8. The temperature drops by the sweep from -12° to 0° and rises again when it reaches slip angle 12° . By increasing load the temperature profile shifts upward to higher temperatures. The temperatures from the trailer α -sweep test at different loads for all tread compounds are presented in Figure 5-9. For every sweep from -12° to 12° , the temperature profiles are similar for each compound but slightly higher for C5 due to its special application. When the tire direction is changed from negative to positive slip angles, the abrasion pattern of the tread compounds also changes. This leads to non-symmetrical α -sweep curves. The temperature profiles presented in the current context were recorded with outdoor tire testing within two days. The reader also is referred to Table 5-10 in the supplementary data for further comparisons.

Figure 5-8: Tire tread surface temperatures in a sweep from -12° to 12° for the reference compound.

Figure 5-9: Temperatures from the trailer α -sweep tests at different loads for all tread compounds: 350 kg, 500 kg, and 700 kg from left to right, respectively; tire testing at constant speed of 60 km/h according to section 5.2.1.

5.3.4. Validation of the laboratory technique with the tire-road data

5.3.4.1. The validation approach

After all the preparatory work, the laboratory method can now be validated with the analyzed tire data. The approach to validate the new technique is by correlating the two groups of the collected data in the laboratory and on the road; the 6 test wheels vs. the 6 tire samples at all test conditions. The fitting parameters derived from the friction curves of the tire data in Table 5-3 and Table 5-4 and the laboratory results in e.g. Figure 5-5 were correlated linearly. Examples of linear regression between the tire trailer α - or κ - sweep tests and the modified LAT100 results for all compounds are shown in Figure 5-10. The high correlation coefficient R^2 for the α -sweep test indicates that the relative ranking of the measured μ_y for the 6 compounds on laboratory scale with the new technique is similar to the tire data on the road. The regression slopes of the figures for α - and κ -sweep tests with laboratory results are opposite of one another, indicating that the new test set-up of the modified LAT100 only correlates with the tire trailer α -sweep tests with positive slopes; the higher the μ_y in the laboratory, the higher the μ_y on the road. The positive slopes between the LAT100 and tire α -sweep data emphasize the similarities between both test mechanisms. The P-value of the linear regression between the tire μ_y and laboratory μ_y is smaller than 0.05 which attributes to a statistically significant correlation within a 95% confidence interval; this is not the case for the regression between tire μ_x and laboratory μ_y as indicated in Figure 5-10. The negative slopes of the correlation with the κ -sweep does not have a physical meaning in the present context. All slopes for all tested conditions are summarized in Table 5-11 the supplementary data.

It should be noted though that the data of the full-scale tire tests, i.e. α - and κ -sweeps interfacing with the same asphalt as the counter-surface, already could not be correlated as explained in section 5.3.3 of the tire data analysis, see Table 5-5. As indicated before in section 5.2.1, the slippage in the contact area of the laboratory test wheel and the tire in α -sweep test are both lateral. Still considering the vast selected range of rubber tread compounds for the present study, the prediction of actual tire lateral grip on the road with a solid test wheel within a laboratory environment is highly successful. However, the argument of the effect of different types of slippage by the two test modalities on friction should be investigated further.

Figure 5-10: Examples of the correlation between friction coefficients at peaks of the normalized tire α -sweep (a) and κ -sweep (b) tests with laboratory results.

5.3.4.2. Correlations between laboratory results and tire data

To explore the interaction between load and speed in laboratory measurement conditions in Table 5-2 on prediction of the tire α -sweep tests, the validation approach of section 5.3.4.1 was employed to all laboratory and tire test data. The regression coefficients R^2 between the μ_y values at Max./Min. of the tire α -sweeps and the μ_y values at Peak for the modified LAT100 sweep tests are summarized in Table 5-7.

Each value in Table 5-7 shows the calculated R^2 for all 6 compounds like in Figure 5-10. 3D graphical representations of the correlation coefficients are presented in Figure 5-11 and Figure 5-12; which shows where the interaction of two factors of load and speed results in occurring the strong correlations.

Table 5-7: Calculated R^2 values for comparison between the tire μ_y values from α -sweep data and laboratory μ_y values at Peak for all test conditions; R^2 values are calculated according to Figure 5-10.

		Tire α - sweep tests						
		Figure 11	a	b	c	d	e	f
		Tire load (kg)	350	500	700	350	500	700
Laboratory results	Speed (km/h)	Load (N)	R^2 values:					
	0.2	55	0.61	0.57	0.58	0.55	0.34	0.56
		75	0.53	0.51	0.5	0.45	0.25	0.47
		95	0.61	0.56	0.56	0.54	0.33	0.51
	2.1	55	0.81	0.71	0.71	0.71	0.54	0.63
		75	0.77	0.71	0.69	0.72	0.58	0.62
		95	0.70	0.69	0.64	0.54	0.37	0.53
	6.5	55	0.85	0.85	0.79	0.67	0.54	0.63
		75	0.75	0.71	0.7	0.69	0.52	0.63
		95	0.48	0.5	0.39	0.27	0.18	0.21
	11	55	0.07	0.05	0.07	0.19	0.28	0.12
		75	0.33	0.19	0.32	0.44	0.29	0.51
		95	0.14	0.09	0.12	0.26	0.34	0.17

Figure 5-11: Graphical representations of Table 5-7 Rows a, b, c: the regression coefficients R^2 (z-axis) vs. load and speed of the laboratory results; the R^2 values are based on correlations between the tire μ_y values at Min. of the α -sweep tests at different tire loads: (a) 350 kg, (b) 500 kg, and (c) 700 kg, and the μ_y values at Peak for the laboratory results.

Figure 5-12: Graphical representations of Table 5-7 Rows d, e, f: the regression coefficients R^2 (z-axis) vs. load and speed of the laboratory results; the R^2 values are based on correlations between the tire μ_y values at Max of the α -sweep tests at different tire loads: (d) 350 kg, (e) 500 kg, and (f) 700 kg, and the μ_y values at Peak for the laboratory results.

Where the R^2 values are the highest, the relative rankings of the measured μ_y for the 6 compounds with the new laboratory technique are closest to tire data on the road. For the laboratory experimental parameters, the speed has a quadratic influence on the R^2 values, vs. the load with a linear impact. Therefore, the speed is the most significant and the effect of normal load is less pronounced compared to the influence of speed. The dependency of the coefficient of friction on speed was also demonstrated by Grosch

to be quadratic, where he showed the similarity to the viscoelastic properties $\tan \delta$ of the rubber [52].

The R^2 values are largest for speed around 6.5 km/h. The scaling down of the traveling speed of the tested tire size (235/35R19) to the laboratory test wheel, as explained in section test conditions, was estimated to be around 8 times smaller compared to a speed of 60 km/h for the tire tests. The present results validate the scaling down of the tire radius to the laboratory test wheel. The contact and relaxation times of the rubber elements in each revolution of the tire and laboratory wheel are similar. This concept is more clear by elaborating on basis of the well-known tire brush model [53]: the volume of rubber of the tread in the contact area is assumed to be a number of deformable elastic bristles that touch the counter-surface. The time lapse between when a bristle leaves the contact area until it comes in contact again, will approximately be the same between the tire and the test wheel. That provides a bristle or rather a volume element of the tread the same time to relax from deformation at similar angular velocity. It implies that analogous test mechanisms and dynamics on the road and in the laboratory scale are generated. Therefore, the comparison can be narrowed down mainly to the rubber tribological properties of the tread compound of the tire and test wheel.

5.3.4.3. Correlations of cornering stiffness

Additional to correlating the tire μ_y values at Max./Min. and laboratory μ_y value at Peak, the so-called cornering stiffness represented by the S (slope) at 0° of the tire α -sweep data and the S at small slip angles for the laboratory μ_y curves can also be correlated: see Figure 5-5 for the slope. The calculated R^2 values are presented in Table 5-8. The tire cornering stiffness stems mainly from the tread compound stiffness because the structure of all tested tires is the same. For the solid test wheels, the radial thickness represents the cornering stiffness. Also, higher load directly affects R_e which leads a drastic change for the calculated R^2 values. So, the correlations for slope S become worse at the higher loads of tires and test wheels, which shows the limit to the load ranges that can be applied.

Table 5-8: Calculated R^2 values for comparison between the tire α - sweep data and laboratory results for slope S at all test conditions.

	Laboratory results for S at small slip angles												
Tire load	Speed (km/h)	0.2			2.1			6.5			11		
	Load (N)	55	75	95	55	75	95	55	75	95	55	75	95
350 kg	R^2 values:	0.91	0.86	0.41	0.87	0.84	0.78	0.86	0.86	0.56	0.89	0.90	0.77
500 kg		0.94	0.86	0.47	0.90	0.88	0.68	0.90	0.89	0.61	0.90	0.89	0.74
700 kg		0.70	0.79	0.14	0.61	0.57	0.58	0.60	0.60	0.26	0.71	0.79	0.80

5.3.4.4. Visual comparisons between road and laboratory

A point to mention still is the fact that the R^2 values for correlation with the μ_y at Max. are somewhat lower than the μ_y at Min. The reason is that the friction curves for the α -sweep tests are not fully symmetrical as already explained in section 5.3.3.1, Figure 5-7. This is a consequence of the way the α -sweep tests on the real tires are executed. It comes from the temperature and the abrasion pattern in the contact area of the tire with the road in the sliding regions of the friction curves. However, there are not only the correlations between the μ_y at Max. and Min. of the tire data and μ_y at Peak of the laboratory results, the whole friction curves are comparable with the tire data, compare Figure 5-6 and Figure 5-13.

For a better visualization, the α -sweep curves of Figure 5-4 (right) were split in two parts: positive and negative slip angles and replotted in a right-handed Cartesian coordinate system. An example is shown in Figure 5-13 for the 6 compounds for load 350 kg. By comparing the laboratory results and the tire α -sweep data at different loads, the effect of the normal load on the tires and the LAT100 test samples is shown to be similar. An example of this trend at speed 6.5 km/h is shown in Figure 5-14 for compound C4; see the supplementary data for the rest of compounds at different experimental test conditions.

Figure 5-13: Splitting and replotting the α -sweep curves of the normalized tire data in a right-handed Cartesian coordinate system for the load 350 kg.

As to the influence of different tire loads on the R^2 values in Table 5-7, the laboratory results are correlating satisfactorily with the minimum tire α -sweep data at all loads at the speed of 6.5 km/h. This is due to the fact that the tire data themselves already correlated well among themselves at different loads: see Table 5-6. The effect of normal load is less pronounced compared to the influence of speed. This corresponds with Schallamach and Grosch who showed that the proportionality of the friction coefficient to the normal load is of a power -1/3 for a smooth surface to -1/9 for a rough one [54, 55]: so the effect is limited than speed. Moreover, the larger load also decreased the R_e in Eq. (5.1) of the test wheel. That explains the poorer correlation at higher load in the laboratory results, e.g. for 95 N.

Figure 5-14: Comparing (up) the normalized tire data with (down) laboratory results at speed 6.5 km/h at different loads for compound C4.

5.3.4.5. Temperature effect

Figure 5-15 demonstrates the temperature profiles for LAT100 sweeps at load 55 N and speed 6.5 km/h for all compounds. The final surface temperatures of the test wheels for different compounds as presented in Table 5-9 are slightly lower than the tire temperatures in Table 5-10; The speed has again more influence on temperature rise rather than load.

Figure 5-15: An example of the surface temperature profiles of the 6 compounds at speed 6.5 km/h and 55 N load.

Table 5-9: The final sample temperatures of the laboratory wheels at different speeds and loads.

Load (N)	Speed (km/h)	C1	C2	C3	C4	C5	C6
55	0.2	23	24	24	23	24	24
	2.1	27	29	28	29	32	30
	6.5	39	40	38	41	43	41
	11	49	50	50	49	51	46
75	0.2	25	24	24	24	24	24
	2.1	28	29	29	30	32	31
	6.5	41	45	42	46	45	45
	11	54	56	56	54	59	53
95	0.2	23	24	23	23	23	23
	2.1	29	30	30	31	33	32
	6.5	42	42	45	49	54	45
	11	54	59	57	55	61	56

5.4. Concluding remarks

Above all, the lateral forces measured with the laboratory test method are solely expected to be the cornering forces; compared to tire α -sweep tests the total collected lateral forces are comprised of major cornering forces and minor contributions of camber thrust and residual influences of ply steer and conicity from the tire structure. Given the large differences between the full-scale tires with air inside and the solid rubber test wheels in the laboratory, and the many influencing factors involved, the correlation coefficient of 0.84 in Figure 5-10 (a) for the modified LAT100 test method on the laboratory track i.e. the disc 180 for this vast range of rubber compounds is a surprisingly satisfactory correlation and as discussed statistically significant. Provided the proper experimental conditions are applied based on the scaling down model as presented here above. This shows the utmost importance of modeling the small solid rubber wheel in the lab and compare it with the tire load distribution to obtain an insight into the different ranges of slip angles in x-axes of Figure 5-6 and Figure 5-13. This will be the subject of a future manuscript.

5.5. Conclusions

Prediction of tire grip on a laboratory scale is always of interest and of high importance for research and material developments. In the present study, a new test method was developed for a Laboratory Abrasion Tester (LAT100) in which the device is exploited as a tribometer with the aim of mimicking the common test modalities of evaluating the tire grip on the road with a trailer tester: lateral (α) and longitudinal (κ) sweep tests. For validation, the new test method was compared with tire grip data collected with the two modalities of a trailer tester on a dry test track for six different tire tread compositions.

- Correlation for the α -sweep tests (lateral slip, i.e. “cornering grip”) were found to be strong (R^2 value > 0.84 ; P-value < 0.05 and a 95% of confidence interval).
- The R^2 values for the correlations between tire α - and κ -sweep tests on the road are poor. Consequently, the correlations for the κ -sweep tests (longitudinal slip, i.e. “acceleration and braking grip”) were not found in agreement with the laboratory test set-up.
- Benchtop testing for ranking of lateral slip performance over a range of loads and speeds can be conducted in only 1 minute, compared with months of preparation and trailer testing of real tires.

- The correlation coefficients are the strongest, based on scaling down of the tire radius to the laboratory test wheel. The condition of appr. 7 km/h and 55 N load on the benchtop tester gives the best correlation with 60 km/h and 350 kg load for α -sweep tests on the tire trailer tester.
- Given the large differences between the full-scale tires with air inside and the solid rubber test wheel in the laboratory, the obtained correlations for this vast range of rubber compounds is surprisingly satisfactory, provided the proper experimental conditions are applied based on the scaling-down model as presented.
- Additional work is required to better understand the two different modalities involved in tire slippage in lateral and longitudinal directions on the road, to be able to also propose new scaling rules for the longitudinal slip on a laboratory benchtop test.
- The approach described to predict actual dry tire grip as a function of speed and load with this laboratory device offers a positive new horizon to evaluate tire grip.

5.6. References

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5.7. Supplementary data

Table 5-10: Tire temperatures from the trailer α -sweep tests at different angles and loads for all tread compounds; at constant speed of 60 km/h according to section 5.2.1.

Tire load (kg)	Compound	Slip angle (°)								
		-12	-9	-6	-3	0	3	6	9	12
350	C1	62	58	52	44	41	43	51	58	63
	C2	60	57	51	43	39	39	48	56	60
	C3	65	59	51	42	40	41	50	59	64
	C4	64	59	52	44	42	44	53	61	66
	C5	71	64	55	45	43	45	54	64	74
	C6	59	53	46	38	36	38	47	56	65
500	C1	61	57	51	45	42	43	49	56	62
	C2	60	56	49	43	41	42	49	56	61
	C3	63	58	50	43	41	42	49	58	64
	C4	63	58	50	44	42	44	51	59	65
	C5	76	68	58	50	47	47	55	66	75
	C6	57	53	47	40	38	38	45	53	60
700	C1	65	61	54	45	42	44	53	61	64
	C2	63	59	51	42	39	42	51	60	65
	C3	68	63	54	43	41	42	53	63	69
	C4	67	61	54	44	42	44	54	64	70
	C5	87	77	65	54	51	52	64	77	87
	C6	64	59	53	43	39	39	51	59	65

Table 5-11: The regression slopes of the linear correlations of the Min. and Max. points of the tire data and the Peak values of the laboratory results.

			Laboratory results												
Tire α - sweep tests	Figure 11	Tire load (kg)	Speed (km/h)	0.2			2.1			6.5			11		
			Load (N)	55	75	95	55	75	95	55	75	95	55	75	95
	a	350	Regression slopes	0.5	0.4	0.3	0.2	0.3	0.4	0.8	0.6	0.6	0.3	0.5	0.8
	b	500		0.3	0.2	0.2	0.2	0.2	0.3	0.5	0.4	0.4	0.1	0.2	0.4
	c	700		0.4	0.3	0.3	0.2	0.3	0.4	0.7	0.5	0.5	0.2	0.4	0.7
	d	350		1.2	1.2	1.6	2.9	2.1	1.3	0.9	1.2	0.6	0.4	0.7	0.2
	e	500		0.3	0.2	0.2	0.2	0.3	0.3	0.5	0.4	0.3	0.4	0.4	1.1
f	700	0.4		0.4	0.3	0.2	0.3	0.4	0.7	0.5	0.4	0.3	0.6	0.9	
k- sweep		350	-41.7	-35.4	-34.8	-23.0	-32.2	-38.2	-59.4	-50.7	-66.3	-14.9	-27.8	-67.2	
		500	-24.8	-21.6	-20.9	-13.6	-19.3	-24.7	-39.2	-30.8	-47.0	-1.7	-4.4	-23.0	
		700	-28.5	-25.2	-22.3	-12.5	-20.0	-26.9	-42.7	-34.7	-42.7	-5.2	2.7	-21.7	

Table 5-12: Full table of R² values, κ-sweep tests addition to Table 7.

			Laboratory results												
Tire α- sweep tests κ- sweep	Figure 11	Tire load (kg)	Speed (km/h)	0.2			2.1			6.5			11		
			Load (N)	55	75	95	55	75	95	55	75	95	55	75	95
	a	350	R ² values:	0.61	0.53	0.61	0.81	0.77	0.70	0.85	0.75	0.48	0.07	0.33	0.14
	b	500		0.57	0.51	0.56	0.71	0.71	0.69	0.85	0.71	0.50	0.05	0.19	0.09
	c	700		0.58	0.50	0.56	0.71	0.69	0.64	0.79	0.70	0.39	0.07	0.32	0.12
	d	350		0.55	0.45	0.54	0.71	0.72	0.54	0.67	0.69	0.27	0.19	0.44	0.26
	e	500		0.34	0.25	0.33	0.54	0.58	0.37	0.54	0.52	0.18	0.28	0.29	0.34
	f	700		0.56	0.47	0.51	0.63	0.62	0.53	0.63	0.63	0.21	0.12	0.51	0.17
		350	0.58	0.52	0.71	0.81	0.80	0.61	0.57	0.71	0.69	0.03	0.12	0.10	
	500	0.50	0.47	0.61	0.68	0.69	0.62	0.60	0.63	0.83	0.00	0.01	0.03		
	700	0.61	0.58	0.64	0.53	0.68	0.67	0.65	0.74	0.63	0.01	0.00	0.02

Figure 5-16: Normalized tire α -sweep data of the 6 compounds at load 500 kg, replotted in a right-handed of Cartesian coordinate system.

Figure 5-17: Normalized tire α -sweep data of the 6 compounds at load 700 kg, replotted in a right-handed of Cartesian coordinate system.

Figure 5-18: Comparing the normalized tire data (left) with laboratory results at speed 6.5 km/h (right) at different loads for compounds C1, C2, C3, C5, C6 respectively.

6 Characterization of counter-surfaces

6.1. Introduction

Tire grip is the interaction between the tire and the road as counter-surface. On studying tire friction, the characteristics of the roads/counter-surfaces are of pivotal importance as well. The excitations on the tire come from the road asperities. A proper characterization of the counter-surface texture is, therefore, an absolute necessity. The present chapter provides the required knowledge over the counter-surfaces employed in the study throughout this thesis. All surfaces were scanned and characterized by laser microscopic apparatuses. The surface micro and macro roughness/textures are evaluated to provide insight into the friction studies.

6.2. Classification of road surface texture

The texture or roughness of the road counter-surface plays a dominant role in tire friction. Various textures contribute differently to friction components and involve

different mechanisms. PIARC⁸ has defined a scale based on the texture wavelength of the contact surface to classify the characteristics and the impact of the roughness on tire performance. Figure 6-1 shows the influence of the texture wavelength on the tire track interaction/performance. The micro-texture (below about 0.5 mm) is of fundamental importance on dry roads and interacts directly with the tire friction on a molecular scale [1].

Figure 6-1: Influence of the texture wavelength on the tire-surface interaction, redrawn based on [1].

The road surface is commonly engineered in such a way that by adjusting the micro- and macro-texture the tire performance is balanced for a wide range of tires, see Figure 6-1. Macro-texture occurs on a scale of 0.5 mm – 20 mm and the micro-texture refers to asperities less than 0.5 mm across the surface, see Figure 6-2. A locally specified anti-skid top layer is usually applied on asphalt roads whose surface is gritty and resistant to abrasion. The surface profile can be characterized in positive and negative structures [2] as depicted in Figure 6-3. The irregularities of the road surface which are lower than 10 mm (100 cycle/m wavenumber), and their impacts are largely absorbed by the tire before they reach the springs of a car suspension system.

⁸ Permanent International Association of Road Congress

Figure 6-2: Macro- and micro-texture representation and wave terminology.

Figure 6-3: Positive and negative structures and distributions resulting in positive or negative skewness of the counter-surface [2, 3].

6.3. Characterization of texture

Numerous roughness parameters are available that can be extracted from texture measurement methods which can be carried out in the lab or outside, such as contactless optical microscopic methods and contact-type profile measuring instruments. The most basic and common roughness parameters are R_a , R_q which are calculated from the amplitudes of the asperities. The amplitude parameters characterize the surface based on the height deviations of the roughness profile from the mean line; the mathematical definitions are as follows:

$$R_a = \frac{1}{L} \int_0^L |z(x)| dx \quad (6.1)$$

$$R_q = rms = \left(\frac{1}{L} \int_0^L z(x)^2 dx \right)^{1/2} \quad (6.2)$$

where R_a (m) is an arithmetic average of the absolute values of the height z (m) of the asperities relative to the mean line over a distance x (m) on the length of measurement L ; and R_q (m) is the root mean square (rms) of the surface heights. R_{sq} is another useful parameter for the shape of the asperities: the skewness which can indicate a positive or negative structure of the counter-surface [4]. Two surfaces with the same R_q can have different shapes [3].

$$R_{sq} = \frac{1}{LR_q^3} \int_0^L z(x)^3 dx \quad (6.3)$$

A surface with a large R_a value, or a positive R_{sq} produces usually high friction and wears quickly. More parameters should be considered: the form and waviness of both amplitudes and frequency of the asperities, their slopes and spacing's. A finite number of surface parameters to describe the real surface geometry is mainly scale-dependent and provides only limited information [3].

In 1971, Nayak [5] modeled rough surfaces as two-dimensional isotropic Gaussian random processes and analyzed them with the aid of random process theory. The surface statistics such as the distribution and density of heights, and the rms height and rms slope were related to the power spectral density of the surface profile; it is a combination of several important surface parameters in one tool. The use of random process analysis has been reported as a breakthrough in recent years, based on the idea of the autocorrelation function⁹ first used in 1946 [6].

The Power Spectral Density (PSD) contains statistical characteristics of the surface topography regardless of the choice of the particular scan size. It decomposes the surface into a superposition of numerous sine and cosine signals of different amplitudes and wavelengths. Using the Fourier transform technique, a wavelength with a specific

⁹ $A(\tau) = \frac{1}{L} \int_0^L z(x)z(x+\tau)dx$: it describes the height similarity of two points on the surface at a distance τ . At $\tau = 0$, $A(\tau) = R_q^2$.

amplitude can be converted to a wavenumber with a specific power. The wavenumber is the number of cycles per meter representing the frequency of the asperities. The term ‘power’ originated from electronic engineering and is a common practice to measure the power content associated with different wavelengths rather than just the amplitudes [2]. Mathematically, a PSD is the Fourier transform of the autocorrelation function of the signals with different spatial frequencies (wavevectors) [6]. There are variations in the mathematical definitions and units of the PSD [7]. The 2D power spectrum $C_{2D}(\mathbf{q})$ in m^4 holds as follows¹⁰ [8]:

$$C_{2D}(\mathbf{q}) = \frac{1}{(2\pi)^2} \int h(\mathbf{x})h(\mathbf{0})e^{-i\mathbf{q}\cdot\mathbf{x}} d^2x \quad (6.4)$$

where the wavenumber $q = 2\pi/\lambda$ is the magnitude of the wavevector $\mathbf{q}$ (m^{-1}) with the wavelength of λ (m) and $h(\mathbf{x})$ is the height coordinate at the point $\mathbf{x} = (x, y)$. The roughness amplitude rms can be written as follows [9]:

$$R_q^2 = 2\pi \int_{q_0}^{q_1} q C_{2D}(q) dq \quad (6.5)$$

where q_0 is the small cut-off wavevector (long roll-off wavelength) and q_1 is the large cut-off wavevector (short wavelength). The slope and curvature rms will be first and second derivatives of eq. (6.5).

To characterize a fractal surface, an understanding of the concepts of self-similarity and self-affinity is required. A fractal is either self-similar or self-affine. The physical appearance of a fractal is the same regardless of the scale at which it is observed. In mathematics self-affinity is a feature of a fractal whose pieces are scaled by different amounts in the x - and y -directions; the y -direction must be rescaled by a^H to preserve the x -direction defined by a , where H is a value between 0 and 1, see later [10]. In nature, a variety of surfaces are associated with self-affine fractal scaling, defined probably first by Mandelbrot in terms of fractional Brownian motion. Self-affine surfaces in three dimensions can be distinguished from self-similar ones by an asymmetry in the scaling behavior orthogonal to the surface xy -plane i.e. z -axis [11]. For a self-affine surface the power spectrum has a power-law behavior [9]:

$$C_{2D}(q) \sim q^{-2(H+1)} \quad (6.6)$$

¹⁰ The bold letter are considered as a vector.

where the Hurst exponent H is related to the fractal dimension by $D_f = 3 - H$ which is usually < 2.3 . A fractal dimension $D_f \approx 2.2$ ($H \approx 0.8$) is typical for asphalt and concrete roads [9]. Theoretically, it is possible to convert a 1D PSD obtained with a linear roughness measurement such as a stylus to a 2D PSD. Figure 6-4 depicts typical 2D PSD and 1D PSD graphs which differ in the slope values. The relationship between a 1D and 2D PSD for an isotropic surface is as follows [12]:

$$C_{2D}(q) \approx \frac{\pi}{q} C_{1D}(q) \quad (6.7)$$

The cut-off wavevectors should be taken from reliable data which depends on the accuracy of the measurement [12]. There are challenges for accurate determination of the 2D-PSD measured with optical microscopy due to various types of surface topography techniques and resolution limits. A deviation from a 1D stylus linear scanner and a 2D-optical scanner typically takes place in the higher frequency regime than 10^4 [13].

Figure 6-4: Theoretical log-log representations of 2D- and 1D-PSD for an isotropic self-affine fractal surface; with q_0 the small cut-off wavevector (long roll-off wavelength) and q_1 the large cut-off wavevector (short wavelength).

Engineering surfaces may also have anisotropic statistical properties, and therefore frictional properties depend on the direction of sliding. For surfaces polished or ground in one direction, contact mechanics depend on the radially averaged power spectrum [9, 14].

6.4. Experimental

The test tracks which are employed for the friction experiments and contact area evaluations in Chapters 3-5 are characterized in the present chapter:

- Road samples: asphalt and concrete;
- Sandpaper grade P120;
- LAT100 counter-surface discs 60, 180, 320, and 180B;
- A comparably smooth surface such as an A4 paper sheet.

The asphalt road sample is a dense asphalt prepared for wet grip tire labeling. A 30 cm² tile of concrete was selected for the contact area measurements in the laboratory which is comparable to the asphalt sample. The sandpaper P120 is a fine grade of Silicon Carbide with an average particle size 125 µm which is classified according to ISO 6344/FEPA Grit Designation. The employed LAT100 counter-surface discs were provided by VMI Holland B.V. - Tire Industry Equipment. They are composed of electro-corundum white Al₂O₃ powder bound in a ceramic binder with a weight ratio of 85/15%. They are classified according to ISO 525:2013(E) as given in Table 6-1, of which the size-fraction of the corundum-particles is the most important variable in the present context. Each disc code represents the disc grain size; a higher code number indicates a smaller corundum particle size. The corundum particles in the ceramic binder are pressed together and the disc surfaces flattened by a grinding process using fine steel powder. The letter “B” for disc 180B indicates “Blunt”, which means that the disc 180 was smoothened one step further in the finishing/flattening process using a grinding disc containing diamond powder. This smoothening process does not influence the grain size, however, it affects the surface structure and sharpness of the asperities as shown in Figure 6-3.

Table 6-1: LAT100 disc characterizations.

Disc designation	60	180	320	180B	
Disc code	EKW 60 SHARP	EKW 180 SHARP	EKW 320	EKW 180 BLUNT	
Average particle size (µm)	225	60	32	↗ ----- Same as disc 180	
Corundum sieve analysis (µm)	0%>425	0%>125	Max. of 52*	----- ↓	
	Max. 30%>300	Max. 15%>90	Min. of 19*		
	Min. 40%>250	Min. 40%>63			
	Min. 65%>212	Min. 65%>53			

* Photosedimentometer analysis (µm)

Microscopic images of the experimental tracks were obtained with two different laser scanning microscopes at various resolutions to obtain the micro- and macro-texture of the counter-surfaces:

1. Keyence confocal¹¹ laser scanning microscopy VK 970 as indoor testing. A maximum scanning area of 2 cm² can be acquired by stitching and combining multiple measurements into one image. Three lens magnifications of 10x, 50x, and 100x at different resolutions were employed.

2. Stemmer imaging laser scanner made with a Micro-Epsilon optoNCDT2300 triangulation laser sensor with a maximum height range of 20 mm. The XY-stage has a range of 100 mm *100 mm and a maximum resolution of 10 µm/pixel. This device was utilized to assess the macro-texture.

To compare both apparatus settings, the devices were evaluated at a specific surface. Since the Keyence device has a certain limit regarding the sample dimension and weight, the Stemmer device was used as a non-destructive method for the counter-surface samples which also enables to scan a larger area and provide information over the larger asperities.

¹¹ The measurements with a confocal scanning microscope cannot be compared accurately at different measurement conditions such as lens magnification, optical zoom magnification, measurement mode, scan settings, scan format (single, double), measurement pitch, XY calibration, and Z calibration (ref: manual of Keyence confocal VK 970)

6.5. Results and discussion

Macro and micro-textures can approximately be distinguished in a PSD graph as depicted in Figure 6-5 for wavenumbers, which correspond to the defined wavelengths in Figure 6-1. The magnitude of $C(q)$ is associated with the power of the amplitudes of the asperities; the higher the $C(q)$, the larger the amplitude. Subsequently, the steeper the slope the rougher the surface. In the current section, the macro and micro texture are compared utilizing the available laser scanners.

Figure 6-5: The distinction of macro- and micro-textures on the PSD graph.

6.5.1. Macro-texture characterization

In the first step, the Keyence and Stemmer scanners are compared by scanning the asphalt sample which is known as a typical self-affine surface. The Keyence device with 10x magnification provided an image by stitching 20*20 small images at 5524 nm/pixel to a final dimension of 24.9 mm * 17.8 mm as shown in Figure 6-6. The Stemmer scanner provided a reciprocal linear scan of an area of 50 mm * 50 mm at the highest possible resolution of 10 μ m/pixel. The 2D-PSD of both measurements are compared in Figure 6-7. The PSD ends at the limits of the resolution of the scanners. The middle regions are in good agreement with basic principles of a linear fit of the PSD; the region at high wavenumbers which deviates from the line is considered as an unreliable region as specified in Figure 6-4 which is due to the outliers of the measurement (the spikes in Figure 6-6). Due to the larger area scan of 5 cm² by the Stemmer device, the PSD graph is extended to the lower wavenumbers. A Hurst exponent $H = 0.8$ for typical road asphalt is illustrated in Figure 6-7. Therefore, the Keyence and Stemmer scanners both give comparable results within the adjusted range of resolutions.

Figure 6-6: 3D view of the asphalt sample scanned by the Keyence device with a resolution of 5524 nm/pixel and dimension of 24.9 mm * 17.8 mm; axes in mm.

Figure 6-7: 2D-PSD graphs of the asphalt surface obtained with the Keyence and Stemmer laser scanners.

To evaluate the macro-texture, all the counter-surfaces under study were characterized by the Stemmer scanner; the images were acquired at the resolution of 10 μm /pixel, the calculated 2D-PSDs are presented in Figure 6-8. Long-distance roll-off wavelengths are observed for LAT100 discs and sandpaper P120: at the beginning of the PSD the $C(q)$ becomes invariant. Corundum particles produced by sintering and a sandpaper surface typically have this property of a long-distance roll-off wavelength [8]. The PSD graphs of the LAT100 discs are in accord with the particle size distribution in Table 3-2; however, the nominally flat surfaces of LAT100 discs and sandpaper still have amplitudes shorter than the diameters of the particle sizes. By considering the

reliable region for wavenumbers smaller than $6 \cdot 10^4 \text{ m}^{-1}$ from Figure 6-7, two bumps can still be detected in the PSD graphs of disc 180 and P120 surfaces. It is a point of discussion whether the data is trustworthy due to reaching the resolution limit. It should be noted that the nature of corundum material compared to asphalt and concrete is shiny and more reflective which could make the optical laser scanner techniques less effective for characterization. Lorenz and et al. [13] have reported that the 2D PSD obtained by the contact-less optical method deviates at wavenumbers larger than 10^4 m^{-1} from 1D PSD assessed by a Stylus scanner. The reliability cut-off wavenumber for optical measurements is also elaborated well in detail in Jacobs's work [7]. According to his explanation, even for concrete the region between 10^4 m^{-1} and 10^5 m^{-1} is not reliable due to the small bend on the PSD line.

Figure 6-8: PSD graphs of all surfaces obtained by the Stemmer scanner at $10 \mu\text{m}/\text{pixel}$ resolution.

Figure 6-9: Image processing and filtering on the microscopic image of disc 180.

To clarify this, the image of LAT100 disc 180 was examined by pixel and image processing as presented in Figure 6-9. Some pixels in a scanning line of order $10\ \mu\text{m}$ have shifted in the images during scanning measurements; this could occur due to environmental disturbances. The shifts were modified by removing the outliers and also with a Gaussian filter with 1σ (standard deviation), which offered a cut-off frequency of $5 \cdot 10^4\ \text{m}^{-1}$, see Figure 6-9. The heart of the matter is that the region between $10^4\ \text{m}^{-1}$ and $10^5\ \text{m}^{-1}$ depends on the surface type and the quality of the image. However, disc 180 and P120 are surfaces with particular particle size distribution and are polished/ground and tend to show asymmetric behavior. For instance, the average particle size $125\ \mu\text{m}$ of P120 gives a $q = 2\pi/(2 \cdot 125 \cdot 10^{-6}) = 2.5 \cdot 10^4\ \text{m}^{-1}$ where is the onset of the second bump; and it is followed by the slight depression. Overall, the counter-surface could be properly compared in the region lower than $q = 2.5 \cdot 10^4\ \text{m}^{-1}$ in macro-texture according to Figure 6-8.

6.5.2. Micro-texture characterization

Second, microscopic images of all surfaces were scanned with the Keyence scanner. Figure 6-10 shows some examples, disc 180, sandpaper P120, and an A4 paper sheet and their 3D representations. Their 2D-PSD graphs are calculated and depicted in Figure 6-10. The resolution of the images is $276\ \text{nm/pixel}$, accordingly, the largest q value on the plot should be of order $10^7\ \text{m}^{-1}$. However, again the reliability of the high-frequency data based on the device type should be taken with care. The PSD graphs of

disc 180 and P120 surfaces start roughly from the same $C(q)$ value; as can be seen in Figure 6-11, the amplitude of the asperities is of the same order of magnitude (between $50\text{ }\mu\text{m}$ to $100\text{ }\mu\text{m}$). The slope of the PSD graph for P120 is steeper than for disc 180 due to the size and distributions of the particles on the surface.

Figure 6-10: Microscopic images by the Keyence scanner at 50x magnification (left) and their 3D views (right) of sandpaper P120, disc 180, and A4 paper, respectively; the scale of the images is $100\text{ }\mu\text{m}$ with the resolution of 276 nm/pixel ; the axes scales are in μm .

Figure 6-11: 2D-PSD of the microscopic images of Figure 6-10; sandpaper P120, disc 180 compared with an A4 paper sheet.

The microscopic images of the LAT100 discs obtained with the Keyence scanner at 10x magnification with 0.05 μm vertical pitch are illustrated in Figure 6-12; with increasing disc code: 60 – 180 – 320, the grain asperities decrease in size. The blunting process of the disc 180B has resulted in a much smoother, flattened surface, in which the individual corundum particles can still be discerned to be of about the same size as in disc 180, but with the asperities reduced to about half the height and a lower rms comparable to disc 320. The PSD graphs and roughness parameters are given in Figure 6-13 and Table 6-2, measured on an area of $1414\text{ }\mu\text{m} \times 1080\text{ }\mu\text{m}$.

Figure 6-12: Roughness surface analyses with confocal laser scanning microscopy of the LAT100 discs employed; height numbers and scales in μm .

Table 6-2: Surface roughness parameters, measured with the Keyence scanner, 10x magnification over an area of $1414 \mu\text{m} \times 1080 \mu\text{m}$, the notation S is being used rather than R because the measurements are done over a surface.

surface	Sa (μm)	Sq (μm)	Ssq (μm)
Disc 60	146	175	-0.44
Disc 180	47	65	-1.37
Disc 320	39	48	-0.33
Disc 180B	35	44	-0.56

Figure 6-13: PSD graphs of LAT100 discs.

6.5.3. Combined macro and micro characterization: overall picture

The cut-off wavenumber q_1 commonly applied in calculations of the viscoelastic contribution of rubber friction on the road surface is around 10^6 m^{-1} [13, 15]. This frequency typically marks the transition to the glassy behavior of rubber. To obtain the overall PSD, the measurement with the Stemmer scanner is combined with the Keyence. The concrete surface is shown in Figure 6-14, the 3D view in Figure 6-15; the combined PSD is represented in Figure 6-16. For q above 10^6 m^{-1} deviation starts to occur from a straight line for the reason explained earlier.

Figure 6-14: The concrete surface at two different magnifications of 10x and 50x by the Keyence device; scales 2 mm and 100 μm .

Figure 6-15: 3D view of the concrete surface, scale is μm .

Figure 6-16: Combined PSD for the concrete surface.

Figure 6-17 depicts the overall PSD graphs for the LAT100 discs combined. The unreliable regions at the higher frequencies are removed. The graphs could be extrapolated to a higher frequency which is commonly done in a linear manner, based on other techniques like Atomic Force Microscopy (AFM) [9, 16]. However, above 10^6 m^{-1} is out of the scope of the current research.

Figure 6-17: The combined PSD graphs of LAT100 discs.

6.6. Conclusions

Tire grip is the interaction between the tire and the road as counter-surface. The counter-surfaces employed in the present thesis were characterized by optical methods utilizing laser scanners: Keyence and Stemmer. The surfaces micro- and macro-roughness/structure were evaluated as an important element for the friction studies in previous chapters. The Power Spectral Densities (PSD) of the surfaces were analyzed and compared.

The reliable cut-off wavenumber of the PSD was discussed. From the PSD graphs of the surfaces in comparison with asphalt, it can be deduced that:

- There is a shift upwards for disc 60 which suggests the asperities with the same wavenumbers in comparison with other discs to have larger amplitudes representing the micro-texture of that surface.
- Disc 320 shows smaller amplitudes for macro-texture compared to disc 180 and asphalt; however, the micro-texture is in the same range as for disc 180 and 180B.
- The extra flattening for disc 180B affects only the macro-texture and the structure or the shape of the asperities compared to disc 180. The micro-texture is the same.
- Sandpaper P120 has a surface with positive R_{sq} , while disc 180 has a negative one. It also has a higher H exponent indicating a rougher surface than disc 180.

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7 Characterization of contact area

7.1. Introduction

Understanding what a tire experiences in the contact with the counter-surface is a key to mimic the situation in the laboratory and from there predict tire grip. Tire grip or friction is an interface phenomenon between two surfaces. Unless the shape and magnitude of the real contact area between rubber and the counter-surface are fully understood, it is not possible to calculate or comprehend the tire grip concept. In the present chapter, the tire contact patch which is the result of the interaction of rubber material and a counter-surface is being investigated. The real or actual area of contact depends on the surface characteristics, applied load, tire inflation pressure, tire speed, and rubber properties. An investigation of the contact area is carried out, measuring with pressure-sensitive films in a stationary configuration. A calculation method is applied in which a wide range of pressures is covered to analyze the contact area and pressure of real tires and the Grosch wheels on different counter-surfaces. The effect of different surface roughness and normal load for various tread compounds is considered. The output is compared by applying the Persson friction theory. And finally, a comparison is carried out between the solid Grosch wheels and the tires.

7.2. Background

It is known that the real area of tire contact with the road is smaller than the nominal or apparent one due to the asperities and irregularities on both surfaces. Some researchers have investigated the shape and magnitude of the tire contact area by FEM computation. They also analyzed tire interaction to understand the stress distribution in the contact area [1-3]. For instance, the measurements of tire contact areas were associated with different tire inflation pressures, loads, and deflections on the ground [4]. In 1994, Fujikawa et al. [5] measured the contact area by transparent replicas of pavements in interaction with a rubber block which was cut from a tire tread. He differentiated the actual contact areas from the nominal ones by visualization using a video camera where the actual contact area appeared as dark regions in the captured images. The ratio of the actual to the apparent contact area was 0.1 for a nominal contact pressure range of 0.3 MPa to 0.5 MPa. Maegawa et al. [6] used a developed friction tester to visualize the real contact regions distributed within the transparent contact interface between Poly-DiMethyl Siloxane (PDMS) and glass surfaces: an elastic hemisphere in contact with a smooth rigid plane. Based on experimental results, an adhesion friction model was developed to explain the normal load dependence of the friction coefficient. This model provided a simple technique that could estimate the contact area and shear stress. The shape and the magnitude of the actual contact area were investigated which was the projected area of the hemispherical rubber PDMS sample on the glass surface with a circular patch.

7.3. Experimental

7.3.1. Sample preparations and rubber properties

Five different tread compounds were selected to prepare Grosch wheels: A-E. Summer tires with size 225/50R17 were built from compound A and B, see Table 7-1. The tire inflation pressure was 2.4 bar or 0.24 MPa. The Grosch test samples are solid rubber wheels with a thickness of 19 mm and external and internal diameters of 84 mm and 35 mm, respectively. The test samples were vulcanized in a Wickert laboratory press WLP 1600 for 11 minutes at 170 °C under a pressure of 160 bar, optimized on basis of t_{90} curemeter data. The curing times of test wheels and tire treads were comparable. The samples were afterward stored for 24 hours at room temperature before testing. For the experiments, the samples were brand-new ones without removing the vulcanization skin from the surface. The rubber physical and mechanical

properties are presented in Table 7-1 and the Dynamic Mechanical Thermal Analyses (DMTA) illustrated in Figure 7-1.

Table 7-1: The physical and mechanical properties of the 5 tread compounds.

Properties	Comp A	Comp B	Comp C	Comp D	Comp E
Hardness (°Sh A)	71	67	68	66	61
Modulus at 300% strain (MPa)	9.0	16.3	9.1	9.1	7.1
M 300/M100	3.4	3.9	4.0	4.1	4.3
Tensile strength (MPa)	15.7	16.8	18.6	20.4	17.7
Elongation at break (%)	500	315	565	580	585

Figure 7-1: DMTA characterizations, storage modulus G' in strain, frequency, and temperature sweeps; and tensile properties of the tread rubber compounds.

7.3.2. Measurement conditions

Grosch wheels corresponding to the 5 tread compounds A-E, and the two full tires A-B were subjected to various test conditions. The ranges of loads and the types of counter-surfaces are listed in Table 7-2. Herewith, the effect of the counter-surface roughness, in-rubber properties, load, and the comparison between the structure of the Grosch wheel and tire were investigated. The test tracks of P120, disc 180, and concrete were fully characterized in Chapter 6 with the PSD graphs; additionally, the aluminum surface is presented and compared in Figure 7-2.

Figure 7-2: The PSD graphs of the applied counter-surfaces.

Table 7-2: Measurements conditions for Grosch wheel and tires.

	Grosch wheel	Tire
Sample designations	A, B, C, D, E	A, B
Loads (N)	35, 55, 75, 95, 115	1500, 3500, 5000, 7000, 9500
Counter-surfaces	Al, P120, disc 180	Al, P120, Concrete

7.3.3. The measurement method

In the tire industry, there are some common ways to measure contact pressure. The examples are thin-film interface pressure mappings like Fuji Prescale pressure-sensitive films and force-sensing technology like tactile sensors of Tekscan or Topaq. The measurements are mainly carried out in two modes: stationary and non-stationary

so-called 'static' and 'dynamic', respectively. The technique of tactile sensors can provide both dynamic or static pressure distributions. It is easy to use and instantly shows the shape and the pressure magnitude and distribution as soon as the object touches the sensor surface. However, it cannot be used on asphalt or any rough surfaces because the roughness damages the expensive sensor, and the resolution for micro-scale purposes is low. The method with thin-film interface pressure mapping works in the static mode according to different pressure ranges: see later. The films with the pressure range of interest are comprised of two measuring films A and C which are polyester based. Film A is coated with a micro-encapsulated color-forming material and film C with a color-developing material. During measurements, the two films are placed with the coated surfaces facing each other, see Figure 7-3. By applying pressure, the microcapsules burst and the color-forming material reacts with the color-developing material. The product of this reaction forms various densities of magenta color on the C-film. The intensity of this color corresponds to the amount of pressure applied; the more intense the color, the larger the pressure. A complicating factor is that these films are sensitive to humidity and temperature. Based on the combination of relative humidity and temperature during the test, Figure 7-4 defines which calibration curve must be employed for the pressure/color intensity correlation, see Figure 7-5 [7].

Figure 7-3: Mechanism of indicating the pressure by sensitive films [8].

Figure 7-4: Example of the graph of humidity and temperature ranges for LLLW (ultra-super low-pressure film) [8].

Figure 7-5: Correlating the color density to pressure for each temperature and humidity range for the LLLW films [8].

In the present study, Fuji Prescale pressure-sensitive measuring films were employed according to two pressure ranges: (i) an ultra-super low pressure (LLLW films) for the range of 0.2 MPa - 0.6 MPa, (ii) an upper low pressure (LLW films) for the range of 0.5 MPa - 2.5 MPa. Momentary pressure measurements were performed with both films at each test condition i.e. the applied pressure on Grosch wheels or tires was raised gradually for five seconds and then maintained at this level for a further five

seconds. The laboratory device for this purpose was the LAT100 for the Grosch wheels. The device for tire footprint measurements was a Zwick/Roell tensile tester at the mentioned loads in Table 7-2, see also Figure 7-6. To calculate the pressure distribution over the range of 0.2 MPa to 2.5 MPa, two measurements according to the conditions given in Table 7-2 were carried out, one with LLLW and one with LLW type films.

Figure 7-6: Pictures of a Grosch wheel and tire during contact area measurements.

7.4. Results and discussions

7.4.1. Evaluation method

The measurements performed with LLLW and LLW type films were combined to map the stress probability distribution. There is an overlap between the pre-defined ranges of pressure for LLLW and LLW type films in the range of 0.5 MPa to 0.6 MPa. As an example, the images of the Grosch wheel with compound A taken with LLLW and LLW films, on the Al surface at a load of 75 N are shown in Figure 7-7, column 1. The images were digitized with MatlabR2017b, Figure 7-7, column 2. The contact area and mean pressure were calculated based on the spectrum of the magenta color; the darker the magenta color, the higher the pressure. The sharp edges and imperfections in the images are because the measurements were carried out on brand-new samples without removing the curing skin, an artifact from the vulcanization. This is also the cause of a certain waviness along the width of the contact patch. The third column of Figure 7-7 provides the histogram of the vertical average pressure in equal steps of 0.1 MPa for each pressure film according to their specific ranges.

Figure 7-7: Images of Grosch wheel footprints on both pressure films, including the digitized pictures and pressure distributions at load 75 N for compound A on AL surface; upper: LLLW films, lower: LLW films.

For the LLLW films, all pressures larger than 0.6 MPa (and outside of the pre-defined range) are accumulated in the large bar which is also visible in the digitized image in the form of dark red color. In this example, around 78% of the whole area has a pressure above 0.6 MPa, but how high exactly is not known. LLW films were utilized to provide information on the pressures larger than 0.6 MPa. The lower bar chart of the third column in Figure 7-7 shows the LLW pressure histogram which can be merged with the first film to create an overall pressure histogram distribution, see Figure 7-8.

Figure 7-8: Pressure histogram distribution based on the combined pressure films for the Grosch wheel with compound A on AL and at load 75 N.

Similarly, for tire A on Al the combined pressure distribution based on the above approach is presented in Figure 7-9 and Figure 7-10. The first impression from these results is that the local pressure reaches a higher value of 0.51 MPa and 0.91 MPa than a typical average contact pressure of 0.3 MPa for a passenger car tire; see above the digitized images of Figure 7-9. Based on film accuracy, the lower pressure limit was 0.2 MPa. Hence, the actual mean pressure should be slightly lower than what is calculated on basis of the pressure films.

Figure 7-9: Tire A footprints at load 5000 N on the Al surface, the digital images and pressure histogram distributions on both films; upper: LLLW films, lower: LLW films.

Figure 7-10: Pressure histogram distribution based on the combined pressure films for tire A on Al at 5000 N.

Figure 7-8 and Figure 7-10 are the results of image analyses on the whole contact areas which illustrate both macro and micro pressures of the Grosch wheel and tire on the Al surface. The macro distribution differs substantially between the tire and the Grosch wheel. This is due to the fact that the Grosch sample is a solid wheel with no air inside, while the tire is pneumatic with specific construction. This leads to a different deflection of the rubbers in the contact areas of the Grosch wheel and tire as shown in Figure 7-11. The side-walls deform under the vertical load on the tire. Within the contact patch between the tire tread and the counter-surface, the resulting bending moments tend to bend the shoulders outwards as shown in Figure 7-11 (left) [9]. This results in a higher pressure on the shoulders of the tire rather than on the tread blocks towards the center as can be seen in Figure 7-9. This explains why the footprint on the pressure films of the tire is sharper and has a narrower distribution than the Grosch wheel. On the other hand, the solid Grosch wheel deflects and flattens in the center of the contact area as presented in Figure 7-11 (right). On the Al surface, the macro pressure is prevailing. Overall, the main difference between the tire and the Grosch wheel is observed in the high-pressure tail of the combined histogram distributions; the sudden drop from LLLW pressure-sensitive film to an LLW one looks unusual: see below.

Figure 7-11: Tire and Grosch wheel deflection in the contact area under vertical load.

7.4.2. Comparison with the real area of contact in Persson theory

In the next step, the captured images by the pressure films are compared with the real area of contact and stress probability distribution calculated by Persson's theory, see Chapter 2. To determine the nominal pressure σ_0 , a region of the contact patch must be selected wherein with a good approximation σ_0 remains constant or is distributed evenly. Despite the micro-stress probability induced by the counter-surface asperities, the macro pressure of the Grosch wheel varies over the contact length in a parabolic manner. Figure 7-9 demonstrates that the pressure is not distributed evenly over the tire contact width and the pressures in the shoulder regions are higher. However, the

σ_0 for the blocks towards the center of the tire contact patch can be estimated to be constant. Figure 7-12 and Figure 7-13 illustrate center blocks of tire A on concrete and sandpaper surfaces at load 5000 N for the pressure-sensitive LLLW and LLW films, respectively. The nominal contact area of the tire sample can be calculated from the applied vertical load divided by the inflation pressure. The tire is inflated to 0.24 MPa and carries a load of about 5000 N. Therefore, the expected tire nominal area is 208 cm² if the tire was a balloon and only if there were no patterns on the tread. A nominal area of 128 cm² excluding the tire grooves was estimated by visual aids software of ImageJ 1.52a. From this and the applied vertical load, the average nominal pressure σ_0 was calculated to be 0.39 MPa.

Figure 7-12: Contact areas of a center block of tire A on concrete and sandpaper surfaces at tire load 5000 N captured by LLLW films, 200 dpi.

To compare this with theory, two blocks from the center region of the tire in contact with concrete with an equal nominal area A_0 of 4.4 cm² were chosen, see Table 7-3; the nominal area was cut small enough to avoid the tire grooves.

Figure 7-13: Contact areas of a center block of tire A on concrete and sandpaper surfaces at tire load 5000 N captured by LLW films, 200 dpi.

Table 7-3: The footprints of two blocks from the center region of the tire in contact with concrete on LLLW and LLW pressure films; compound A.

Sample	LLLW film	LLW film
1		
2

Figure 7-14: Stress-strain behavior of the pressure-sensitive film performed by a Zwick-Roell tensile tester according to ISO 37.

It should be noted that in the contact between the rubber and the counter-surface substrate two semi-rigid polyester films are present. Therefore, the properties of these films should be taken into account. The onset of the non-linear region of the stress-strain curve of the pressure-sensitive films in Figure 7-14 is about 50 MPa, before the plastic yield stress of the polyester films. It impacts the large cut-off wavenumber q_1 (see chapter 6) which is estimated to be 10^4 m^{-1} , indicating a limitation on the capability of rubber to penetrate into the irregularities and asperities of the counter-surfaces. The modulus $E = 3.5 \text{ MPa}$ can be extracted from Figure 7-1, the DMTA properties of compound A. The stress probability distributions can then be calculated for the LLLW and LLW films from the experimental evaluation and the theoretical approach, see Figure 7-15. From this, the relative contact areas A/A_0 were determined to be 0.51 and 0.24, respectively. The pressure distribution probability evaluated with the experiment is very sharp compared to the theoretical calculation. The main discrepancy between the theoretical and the experimental calculations comes from the LLLW film which is far from what is predicted by theory, see later.

Figure 7-15: Pressure probability distribution according to pressure-sensitive films in comparison with Persson theory for compound A on the concrete surface at load 5000 N [the graph obtained by Dr. Persson]; paper 1 = LLLW and paper 2 = LLW films.

On the one hand, the rigidity of the polyester pressure films limits the capability of the rubber to penetrate the irregularities and asperities of the counter-surfaces, which impacts the accuracy of the measurements. This is considered, to some extent, in the calculations in the large cut-off wavenumber q_1 . On a very smooth counter-surface, this discrepancy should be negligible, but it is more pronounced for rough counter-surfaces like concrete or disc 180. On the other hand, (i) the color-forming mechanism of pressure-sensitive films is not fully understood i.e the wetting process of bubbles on the color-forming (ii) the size and distribution of color-forming bubbles on film A of the pressure-sensitive films affect the resolution. Figure 7-16 to Figure 7-19 are microscopic images of film A of the LLLW and LLW pressure-sensitive films and representation of the color-forming bubble distribution obtained by 3D light Keyence microscopy at 200x and 900x magnification, respectively. The bubbles in the LLW film are distributed and dispersed nearly evenly, while in the LLLW film they are agglomerated in spots.

Figure 7-16: Microscopic image of film A of the LLLW film and representation of the color-forming bubble distribution obtained by 3D light Keyence microscopy at 900x magnification, scale is 100 μm.

Figure 7-17: Microscopic image of film A of the LLW film and representation of the color-forming bubble distribution obtained by 3D light Keyence microscopy at 900x magnification, scale is 100 μm.

Figure 7-18: Microscopic image of film A of the LLLW film and representation of the color-forming bubble distribution obtained by 3D light Keyence microscopy at 200x magnification, scale is 200 μm .

Figure 7-19: Microscopic image of film A of the LLW film and representation of the color-forming bubble distribution obtained by 3D light Keyence microscopy at 200x magnification, scale is 200 μm .

To obtain some more insight into the discrepancy in footprints collected experimentally and calculated theoretically, the depth of indentation of a Shore A

indenter was measured on the LLLW pressure-sensitive film. The mechanism of contact is seemingly analogous considering the penetration of one single asperity of the counter-surface into rubber compounds. The measurements were carried out using the standard compounds for hardness calibration: the compounds with the designated hardness of 73.2 °Sh A and 61.9 °Sh A, which are close to the range of hardness in the current study.

The indenter of Shore A durometer is a 35° truncated cone (frustum) as shown in Figure 7-21. According to ASTM D2240, attached to a spring force of 8.05 N and the hardness value depends on the depth of penetration applied for a certain time on the rubber block of at least 6 mm thickness. The hardness H is scaled from 100 to 0 in a linear relationship to the depth of penetration h from 0 mm to 2.54 mm. Theoretically, at a depth of $h_2 = 0.97$ mm in Figure 7-21, the corresponding H will be 61.9 °Sh A. Conversely, the h for $H = 72.3$ °Sh A is calculated, see Table 7-4. At a depth of h with a radius of r , the projected area is $A_p = \pi * r^2$ and geometrical area A_c could be the summation of the base area A_b and the lateral area of A_l of the frustum, see Figure 7-21. The difference between the geometrical area and the projected area of the indent has been investigated in the literature [10, 11]. The geometrical area is obtained if the indentation produces 100% plastic deformation. In the case of elastic-plastic behavior, the measurement also involves an amount of elastic recovery of the material.

Suppose under the applied load the rubber reaches full contact with the conical frustum, indicating that no rubber sink-in occurs, then the real area of contact between the rubber and the indenter is A_c . The areas on the LLLW pressure film A_{llw} calculated from the digitized pictures are twice higher than A_c , see Table 7-4. This is because the pressure film encompasses the needle by folding and pleating during the measurement to be able to fit at the depth of h between the rubber and the indenter. The expected radii and areas on the film due to folding are given in Table 7-4 which are close to the calculated A_{llw} . It shows that the areas measured with the pressure-sensitive films are overestimated relative to A_c .

Figure 7-20:Pile-up and sink-in phenomena [12].

Table 7-4: Characterization of standard calibration compounds for hardness based on the depth h and radius of indentation r .

H (°Sh A)	73.2	61.9
h (mm)	0.68	0.97
r (mm)	0.63	0.70
A_p (mm ²)	1.27	1.54
A_l (lateral) (mm ²)	2.34	3.49
A_c (actual) (mm ²)	2.83	3.98
A_{llw} (area on film LLLW) (mm ²)	5.89	7.19
The expected radius on film (mm)	1.36	1.71
The expected area on film (mm ²)	5.80	9.20

Figure 7-21: Geometry of a Shore A durometer.

Nevertheless, the real area of contact is something between the defined A_c and A_p due to neglecting the sink-in effect of the rubber. In total, two scenarios might occur in the real area of contact: pile-up or sink-in [13]. In the purely elastic contact solution, the material always sinks in. While for an elastic/plastic contact, the material may either sink-in or pile-up. The heart of the matter is that in both scenarios, the actual geometric areas in contact with the rubber are larger than the projected ones. For elastomeric materials with viscoelastic behavior, mainly the sink-in phenomenon is expected to take place. The difference between the areas depends on how much stress is applied to rubber and whether rubber can manage to fill out the cavity fully or partially with the given applied stress [10].

7.4.3. Comparison between the tires and Grosch test wheels in pressure histograms

Irrespective of the overestimations of the pressure probability distribution in the measurements with the LLLW film, the pressure histograms of the tires and Grosch wheels are qualitatively compared. Figure 7-22 shows the pressure histograms of tire A and Grosch wheel A on different counter-surfaces at various vertical loads. It should be noted that this comparison is the overall pressure of macro and micro distributions. By increasing load the pressure distribution plots move upwards which means the areas of contact expand over the counter-surface as expected. The difference between the deflections of the solid rubber wheel and tire structure can best be illustrated in the pressure distributions on the Al surface. As to the Grosch wheel, the larger pressure zone grows with increasing load and the smaller zone remains almost unchanged, whilst this is not the case for the corresponding tire. The trends on the P120 surface look more alike compared to the Al surface. Last, the comparison between the concrete surface and disc 180: the distributions look alike.

Figure 7-22: Pressure histograms at different vertical loads of the tires and Grosch wheels on all surfaces.

7.4.4. Effect of in-rubber properties and surface roughness

Figure 7-24, row 1 shows the Grosch wheel pressure histograms at various normal loads for each compound on the Al surface. According to the in-rubber properties in Table 7-1, the compound hardness ranks as follows: A>C>B>D>E; the largest hardness difference is between compound A and E with 10 °Sh A. The harder the compound the more it resists to wheel deflection. Hence, the larger pressure zone grows. But a softer compound allows the rubber to expand over the surface and growth in the lower pressure zone also becomes visible. The trend of the load variation for each compound can be seen for the other counter-surfaces of P120 and disc 180 as well, see Figure 7-24, row 2, and 3. In contrast to the Al surface, the increase in the smaller pressure zone is now comparable with the larger one for the harder compounds as well. This indicates that the rubber starts to fill out the cavities between the asperities rather than just expand over the surface.

The pressure distributions on the Al surface can be roughly taken as a reference to deduct the effect of macro pressure on the two other surfaces, therefore, it is possible to qualitatively focus on micro pressure and how rubber fills out the cavities between

asperities. In total, by increasing load in the combined pressure distribution plots, three possibilities can be observed: first, the growth of the low-pressure zone dominates the higher zone; second, the higher pressure zone progresses more; last, both zones progress simultaneously. These different trends can roughly describe how the rubbers fill in the cavities on the counter-surface. In the *first* case, it is more likely that rubber fills the longer wavelength cavities with a higher amplitude corresponding to smaller q and higher C in the PSD graphs. It is noteworthy that on a very smooth counter-surface substrate like the Al, the macro pressure due to wheel deflection is dominant over the micro pressure due to the surface irregularities. If the *second* case occurs, it indicates that the rubber is hard and has less chance to fill the cavities. This case is mainly visible on the Al surface and it is because of the physical deflection of the wheel itself. In the third case which happens most often, the rubber penetrates the larger wavelength cavities, it already fills in the shorter wavelength cavities superposed on the summits of the large wavelength ones. After that, it starts to fill out the larger cavities deeper until it reaches a certain depth. By doing so, extra pressure is applied to the shorter wavelength cavities superposed on the summits of the large wavelength, therefore a rise can be seen in the higher pressure zone.

The data are replotted in Figure 7-25 for different loads and three counter-surfaces of Al, P120, and disc 180 with the compounds as variables. In these graphs, mainly the first case, growth in smaller pressure zones occurs. The pressure histogram distributions are in correlation with rubber hardness or G' at 10 Hz, 6% strain, and 40° C. For instance, the best representative examples can be observed at loads 55 N and 75 N on disc 180.

To observe also the influence of in-rubber compound properties on the tire contact area, the pressure distributions are plotted in Figure 7-23. These plots also confirm the effect of hardness on the penetration of rubber into the cavities on the counter-surfaces.

Figure 7-23: Effect of compound hardness on pressure distributions of tire A and B.

7.5. Summary and concluding remarks

In the present chapter, the objective was to obtain insight into the shape and magnitude of the contact between the rubber samples and the counter-surfaces to be able to comprehend the tire grip concept. The magnitude of the contact patch was measured by pressure-sensitive films supplied in two different ranges from LLLW: 0.2 MPa - 0.6 MPa and LLW: 0.6 MPa - 2.6 MPa. The stress probability distributions in the contact patches were evaluated and compared with the Persson theory. In the contact between the rubber and the counter-surface substrate, the rigid polyester films impact the large cut-off wavenumber q_1 which directly affects the calculated area of contact in Persson's friction theory; indicating a limited capability of rubber penetration into the irregularities and asperities of the counter-surfaces. The areas measured with the pressure film LLLW were overestimated compared to the ones calculated by theory. The discrepancy in footprints collected experimentally and calculated theoretically was investigated by measuring the depth of indentation of the Shore A indenter on the pressure film LLLW. The pressure films encompass the needle by folding and pleating to be able to fit at the depth h between the rubber and the indenter. It leads to a larger color-formed region on the pressure film compared to the actual scenario. Furthermore, the pressure probability distribution of the Grosch wheel and tire were qualitatively compared. The overall shape of the pressure probability distributions of the Grosch wheels is similar to those of the tires based on the same compounds. The pressure histogram distributions are in correlation with rubber hardness or storage modulus at 10 Hz, 6% strain, and 40 °C

7.6. References

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Figure 7-24: Grosch wheel pressure profiles for various loads on Al, P120, and disc 180 for different tread compounds.

Figure 7-25: Grosch wheel pressure profiles for all compounds at various loads on Al, P120, and disc 180.

8 Understanding the LAT100 as a tribometer for tire friction with modeling

The LAT100 as the main apparatus used in the present thesis – despite its name – was re-designed and employed as a laboratory tribometer for prediction of tire friction. It was demonstrated that the LAT100 can indeed be exploited particularly to simulate the α -sweep tire tests, but not the κ -sweep. The correlations were supported and verified with statistical analyses. Nevertheless, validation was performed for one set of tread compositions. Obtaining more insight into the wheel rolling dynamics contributes to the understanding of the involved friction phenomena and mechanisms in the contact patch. Hence, the conclusion that the LAT100 can be a reliable predictor for tire grip becomes concrete. Therefore, the present chapter is dedicated to understanding the dynamics and physics of a rolling rubber wheel on a counter-surface disc of the LAT100 test step-up utilizing the renowned physical “brush model” for tires. Therewith, the behavior of the rolling wheel in lateral and longitudinal movements, as opposed to a real tire, can be investigated. And more comprehension of the intricate differences between the α and κ sweep test configurations is obtainable.

8.1. LAT100 test set-up

8.1.1. The LAT100 as a rolling friction tester

Among all available tribometers suitable for measuring tire tread frictional properties, great interest exists towards measurements with solid rubber wheels [1-17]. The main advantage is the rolling movement of the rubber wheel sample known as the Grosch wheel which is analogous to a tire. The LAT100 manufactured by VMI Holland B.V. [18], the Netherlands, is a compact and versatile machine that is designed to simulate tire service conditions to measure the abrasion loss and the frictional forces as a function of various slips, loads, and speeds on different counter-surface substrates. The friction properties as the output of the machine can be employed as input data for tire models and simulations. Using a rolling wheel sample provides a more realistic input for tire modeling due to better surface cooling, compared to sliding body measurements with continuous contact to the counter-surface [1].

Figure 8-1 shows a schematic view of the LAT100 test set-up. The machine consists of a driven disc with a diameter of 350 mm onto which a solid rubber wheel with outside and inside diameters of 84 mm and 35 mm and a thickness of 19 mm, known as Grosch wheel, is pressed under a given normal load F_N (N) and run at a defined slip velocity of V_s (km/h). The slip velocity is created by combining the disc traveling velocity V_t and a slip angle α (°). The basic principle is that the disc spinning with a certain traveling velocity induces rotation of the Grosch wheel with a circumferential velocity V_c . Side (lateral) and tangential friction forces are generated by slip velocity and the normal force. Counter-surface discs with various surface roughness can also be mounted. All three force components acting on the wheel during the tests are recorded with the load cell measuring hub. The test set-up is arranged in a vertical alignment which eases the removal of the abrasion debris. The vertical configuration does technically occupy less space than a horizontal one.

Figure 8-1: Schematic view of the LAT100 test set-up.

The frictional forces as direct outputs of the measurements with the LAT100 enable to rank and compare various rubber compounds. This apparatus has been considered for the evaluation of rubber friction properties due to its similarity to vehicle tire operating conditions on wet and ice surfaces [1, 6, 10, 12, 13, 15]. Good correlations between tire data and laboratory test results have been obtained for abrasion and rolling resistance [9, 11-13].

8.1.2. Effect of disc curvature

The curvature of the driven disc of the LAT100 causes a variation in the slip velocity in the contact area between the Grosch wheel and the disc counter-surface. Based on the gradient of the radial distance between leading and trailing edge of the Grosch wheel to the disc center point o , the traveling velocity alters: $V_t = r * \omega$, the further the point from the center, the larger the traveling velocity. Here R in Figure 8-1 and Figure

8-2 is the distance between the center of the contact area and the disc center point o , and r varies around R value dependent on the slip angle and disc curvature. Consequently, the slip velocities vary; this is not the case if no curvature exists. Figure 8-2 and Figure 8-3 depict the schematic view of the contact area of the Grosch wheel with and without a disc curvature at a slip angle α ($^\circ$).

Figure 8-2: Schematic view of the contact area of the Grosch wheel at slip angle α ($^\circ$) with disc curvature. The black, green, and red vectors are disc traveling V_t , wheel circumferential V_c , and slip V_s velocities in the contact points, respectively.

Figure 8-3: Schematic view of the contact area of the Grosch wheel at slip angle α ($^\circ$) running without disc curvature. The black, green, and red vectors are disc traveling V_t , wheel circumferential V_c , and slip V_s velocities in the contact points, respectively.

The relationship between the traveling and slip velocities in the center point of the wheel contact with the counter-surface disc is expressed in Eq. (8.1).

$$V_s = V_t \sin \alpha = R\omega \sin \alpha \quad (8.1)$$

At any other point in the contact area, the slip velocity varies according to Eq. (8.2).

$$V_s = V_t \frac{r(x, y, \alpha)}{R} \sin \alpha \quad (8.2)$$

where r can be estimated as follows:

$$r(x, y, \alpha) = R^2 + x^2 + y^2 - 2R(y \cos \alpha + x \sin \alpha) \quad (8.3)$$

The derivation of r is presented in the Appendix. On an actual road, the impact of a bend in cornering depends on the radius of the curvature which is most notable in sharp turns.

8.1.3. Wheel deflection

A deflection δ for a rubber wheel-shaped sample occurs as shown in Figure 8-4. As the normal load F_N increases, the wheel flattens more in the center of the contact area. The r_e is defined as the effective radius of the rubber wheel when a spinning without an external torque is applied to the rolling axis. Its value lies somewhere in between the unloaded radius r_w and the static loaded r_l [19].

Figure 8-4: Wheel deflection under normal load F_N , vertical deflection δ , loaded radius r_l , effective radius r_e , unloaded wheel radius r_w , and α is half of the contact length.

8.2. Rolling friction of a tire

The tire function on the road is very complicated. The viscoelastic nature of the rubber material of the whole tire determines the function attainable. This nature is the main reason for the complexity. Tires are composed of various elements which are mainly rubber compounds, each contributing to the overall performance profile. This complex structure interacts with the road as an intricate system, which was already discussed in Chapter 6. Consequently, the tire is considered as a non-linear function of a multitude of inputs and outputs, see Figure 8-5. To predict tire grip or friction, considering all influential factors in one laboratory device, seems an elaborate task to accomplish.

Figure 8-5: Tire function on the road, taken from ref. [20].

However, the tread composition itself is the main factor influencing tire friction beside the tire construction and the tread pattern. The latters contribute only around 25% of all estimated on a dry surface, see Figure 8-6 [21].

Figure 8-6: Contributions of the influential factors on tire friction for dry/wet surfaces [21].

The attempt within the present thesis was to predict dry tire grip with the determination of the rubber friction in a laboratory environment. To mimic the situation in the laboratory environment, it is required to understand what a tire experiences in the contact patch. In Chapter 2, some introductory concepts of the interface of the tire with the road, frictional forces, slippage, and friction curves were described in two situations of longitudinal and lateral slips. Considering the simplified schematic view of tread blocks in the contact patch in longitudinal slippage in Figure 8-7, tire grip is generated by the frictional forces which are created by the relative movements of these tread blocks on the ground known as slippage. Slippage occurs only during cornering, braking, and accelerating when the tread block has a displacement; otherwise, the relative speed of the tire in the contact line would be zero.

Figure 8-7: Tire tread blocks in the contact area in longitudinal slippage with identifications of regions: stationary and non-stationary (also known as adhesion and sliding regions according to Pacejka); and friction components: adhesion and hysteresis; based on [20, 22, 23].

Two regions can be recognized in each tire revolution: stationary and non-stationary, also known as adhesion and sliding regions according to Pacejka [23]. What every single tread block experiences in the contact area, first, starts with the front non-stationary region in which the tread blocks adhere in a perpendicular direction to the

counter-surface and undergo deformation and shear stress; it is expected to predominantly have adhesion as friction component. Then, the blocks enter into the non-stationary region, and leave the contact area with a noticeable displacement under a range of frequency excitations which originate from the surface roughness of the ground; it is expected to have hysteresis friction as the dominant component. Without any kind of slippage and disregarding the possible rolling resistance, the tread block would leave the contact area still in a perpendicular direction to the counter-surface. The friction curve in Figure 8-7 describes the transition from the stationary to the non-stationary regions by describing the frictional force vs. the slippage: where the tire mostly adheres to the counter-surface; then reaches a maximum in friction force; and finally starts to slide. The ABS braking principle is commonly optimized at the highest friction.

In the case of lateral movement, the displacement occurs in parallel to the rolling axis of the tire. Both will be explained in detail by exploiting a physical model: the so-called brush model. In the coming sections, the tire behaviors are qualitatively and comparatively described. The pure lateral slip is closer to the governing slip mechanism of the LAT100. A combined slip condition is not discussed in the present context.

8.2.1. The tire brush model

The relation between frictional force and slippage is of high importance in tire dynamics and mechanics. Figure 8-7 depicts schematic and graphical representations of this relationship; the mathematical relation, which is a point of great interest for many researchers [4, 19, 20, 23-34], requires a sophisticated model. The concept of the 'brush' model originates from Fromm and Julien as stated by Pacejka [23]. The brush model employed in the present thesis describes qualitatively the main features of a slipping tire based on Pacejka and Besselink. In the present section, the nomenclature is based on Besselink [20] and the sign convention on Pacejka [23] and the equations on both. The model can be extended to better accuracy depending on the types of assumptions. Tire behavior can be described in greater detail through more complex models, computer simulations, and the finite element method [28, 35, 36].

The brush model is typically known as a row of elastic bristles attached to a rigid body called 'carcass'. Every bristle comes to the counter-surface plane, touch it, and can deflect and leave it until the next revolution. The bristles may be called tread volume or elements and their compliance corresponds to the overall elasticity of the real tire including carcass, belt, and tread layers. In the present context the brush model is described based on the following assumptions [20, 23, 32]:

- (i) The carcass is rigid and flattened in the contact region with a length of $2a$;
- (ii) tread elements attached to carcass are elastic and the deformation of the rubber volume between the ground and the tire carcass generates the slip;
- (iii) the rubber part is approximated as a single row of equally-spaced deformable bristles like a brush i.e. the lateral contact width $2b$ is assumed to be a thin slice of the tire;
- (iv) the bristles are compliant i.e. can deflect either in longitudinal or lateral directions parallel to the counter-surface;
- (v) a bristle enters the contact region perpendicular to the counter-surface;
- (vi) the bristle leaves the contact region after a certain time but in a presence of slip;
- (vii) the slip causes a bristle deflection $\varepsilon(x)$ (m) and force per unit length $q(x)$ (N/m) generated in the contact point which follows Hooke's law:

$$q(x) = k_b \cdot \varepsilon(x) \quad (8.4)$$

with k_b (N/m²) the bristle stiffness per unit length. This is valid either for lateral $q_y(x), \varepsilon_y(x)$ or longitudinal $q_x(x), \varepsilon_x(x)$ directions; a Cartesian coordinate system is introduced with x longitudinal, y lateral, z perpendicular directions attached to the contact center C where the coordinate system is situated;

(viii) every bristle can deform independently of one another i.e. at a specific time both stationary and non-stationary (or adhesion and sliding) regions in the contact line can occur;

(ix) the longitudinal, lateral, and vertical forces are determined by summations of individual bristle contributions. The forces are the integrals over the contact length and read as follows:

$$F_x = \int_{-a}^a q_x(x) dx \quad (8.5)$$

$$F_y = \int_{-a}^a q_y(x) dx \quad (8.6)$$

$$F_z = F_N = \int_{-a}^a q_z(x) dx \quad (8.7)$$

Figure 8-8: The tire brush model drawn according to the assumptions and [20].

Figure 8-8 depicts the tire brush model according to the mentioned assumptions. Pure slip conditions can occur:

1. when the velocity of travel, vector V shows an angle α with the symmetry plane of the tire normal to the rolling axis; it creates lateral slip;
2. when the forward component of the wheel velocity $V_x = V \cos \alpha$ is not equivalent to $V_r = r_e \Omega$, where Ω (rad/s) is the angular speed of the tire; it creates longitudinal slip.

The tip of each bristle moves from the leading to the trailing edge and stays in contact with the counter-surface as long as friction allows in each revolution. The deflections of the bristles increase in a direction parallel to the velocity V towards the trailing edge. This velocity vector could be parallel to V_x in longitudinal slip or form an angle with V in the case of lateral slip. The base points of the bristles fixed to the carcass move backwards with a velocity of V_r relative to the contact center C ; while with respect to the counter-surface, the base points travel with a slip speed V_s , see Figure 8-9.

Pure longitudinal slip-Side view

Pure lateral slip-Top view

Figure 8-9: Pure longitudinal and lateral slip conditions, based on [20, 23].

By assuming a parabolic function for $q_z(x)$, F_z in equation (8.7) can be obtained by inserting the following expression:

$$q_z(x) = \frac{3F_z}{4a} \left(1 - \left(\frac{x}{a}\right)^2\right) \quad (8.8)$$

The maximum deflection of the bristles $\varepsilon_{max}(x)$ is limited by the presence of the coefficient of friction μ assumed constant. The magnitude of the generated frictional force per unit length $q(x)$ is defined according to Coulomb's equation:

$$|q(x)| \leq \mu q_z(x) \quad (8.9)$$

Equation (8.9) is valid whether the bristle moves in x or y direction i.e. longitudinal $q_x(x)$ or lateral $q_y(x)$ direction. From equations (8.4) Hooke's law and (8.9) Coulomb's

law, the maximum bristle deflection $\varepsilon_{max}(x)$ can be derived in equation (8.10) and has a parabolic shape as well because it is a function of $q_z(x)$:

$$|\varepsilon_{max}(x)| = \frac{\mu q_z(x)}{k_b} \quad (8.10)$$

This concept is a good basis for the following discussions of the mathematical calculations of the frictional forces and 'break-away' point which is the transition between the stationary and non-stationary regions. In the next section, they will be discussed for two scenarios of pure lateral and longitudinal slips.

8.2.2. Pure lateral (side) slip

In the stationary region, the contact line between the bristles and the counter-surface is straight, the bristles follow the straight line of contact which is parallel to V , the velocity of travel. By applying the slip angle α , at the moment that the straight line of contact intersects with the maximum deflection parabola ε_{max} , the non-stationary region begins from the intersection or the break-away point in Figure 8-10A. The bristles coincide with the parabola for the maximum possible deflection. The deflection is not linear anymore and is curved indicating that available frictional forces become lower than the forces that would have continued along the straight line, Figure 8-10B. Due to asymmetry of the deflection, a self-aligning torque or moment M_z arises. The moment is the product of multiplying the lateral force F_y by the moment arm, or the 'pneumatic trail' t_p , which is the distance of the lateral force line of action from point C . As the slip angle increases, the deformation profile becomes more symmetric. The point of intersection shifts to the leading edge, the lateral force escalates and t_p decreases; until the contact line becomes the tangent to the parabola at the foremost of the leading edge, Figure 8-10C. The deformation becomes fully symmetric and the stationary (adhesion) region disappears and full sliding occurs. The friction curve turns flat, the t_p becomes zero, and the self-aligning moment vanishes. By further increasing the slip angle, the situation for bristles has to stay theoretically unchanged, see Figure 8-10D.

Figure 8-10: Pure lateral slip condition and resulting force and aligning moment by increasing slip angle α , based on [20, 23].

The mathematical equations for the above description of the pure lateral slip are derived for a steady-state situation $\partial \epsilon / \partial s = 0$, i.e. deflection does not change with distance s . Consider Δt the time that the bristle already spent in the contact region:

$$\Delta t = \frac{a - x}{V_r} \quad (8.11)$$

then the pure lateral deflection derived from the trigonometry of velocities in the contact region [20] is as follows, see also Figure 8-9:

$$\varepsilon_y(x) = -V_{sy}\Delta t = (a - x) \tan \alpha \quad (8.12)$$

Based on the last assumption of the model in the previous section, the lateral or cornering force F_y is the integral of $q_y(x)$ over the contact length $2a$; by substituting equation (8.4) and (8.12) for very small slip angle, it results:

$$F_y = \int_{-a}^a q_y(x) dx = k_b \int_{-a}^a \varepsilon_y(x) dx = k_b \tan \alpha \int_{-a}^a (a - x) dx \quad (8.13)$$

By solving the integral for very small $\alpha \approx \tan \alpha$, the relationship between frictional force and slippage i.e. F_y as a function of α in the stationary region follows the expression below:

$$F_y = 2k_b a^2 \alpha = C_{F\alpha} \alpha \quad (8.14)$$

where $C_{F\alpha}$ (N/rad) is the cornering stiffness. Note that the cornering stiffness is the slope of F_y vs. α :

$$C_{F\alpha} = (\partial F_y / \partial \alpha)_{\alpha \rightarrow 0} \quad (8.15)$$

which depends on tread stiffness and the length of contact $2a$. Subsequently, the self-aligning moment M_z is calculated with the moment arm:

$$M_z = \int_{-a}^a q_y(x) x dx = -\frac{2}{3} k_b a^3 \alpha = C_{M\alpha} \alpha \quad (8.16)$$

where $C_{M\alpha}$ (Nm/rad) is the self-aligning stiffness. Accordingly, the pneumatic trail t_p can be calculated using equations (8.14) and (8.16) as follows:

$$M_z = F_y \cdot t_p \rightarrow (t_p)_{\alpha \rightarrow 0} = \frac{1}{3} a \quad (8.17)$$

These calculations were made for the stationary region. To find the distance of the breakaway point x_t to the center C (Figure 8-10), the linear bristle deflection in the stationary region has to be equal to the maximum of the parabolic bristle deflection. Hence, from equations (8.10) and (8.12), two solutions are obtained for x_t :

$$\varepsilon_y(x)|_{at x_t} = \varepsilon_{ymax}(x)|_{at x_t} \Rightarrow$$

$$\text{solution 1: } x_{t1} = \frac{4k_b a^3}{3\mu F_z} (\tan \alpha) - a, \quad (8.18)$$

$$\text{solution 2: } x_{t2} = a$$

Solution x_{t1} is the breakaway point, describing that from $-a$ to x_{t1} is the stationary region and from x_{t1} to a is the non-stationary one. If $x_{t1} = a$, the sliding slip angle $\alpha_{sliding}$ can be deduced:

$$\alpha_{sliding} = \tan^{-1} \left(\frac{3\mu F_z}{2k_b a^2} \right) \quad (8.19)$$

This is the slip angle when the stationary region vanishes and full sliding occurs, where solution 2 is indicated, Figure 8-10C. Similar to equation (8.13), F_y and M_z can be written for the contact length when both regions exist. For the stationary region equation (8.4) and the non-stationary region equation (8.9) are relevant, hence F_y is the summation of two integrals:

$$F_y = \frac{3\mu F_z}{4a} \int_{-a}^{x_t} \left(1 - \left(\frac{x}{a} \right)^2 \right) dx + k_b \int_{x_t}^a (a - x) \tan \alpha dx \quad (8.20)$$

Correspondingly, by considering the torque arm, the aligning moment becomes:

$$M_z = \frac{3\mu F_z}{4a} \int_{-a}^{x_t} \left(1 - \left(\frac{x}{a} \right)^2 \right) x dx + k_b \int_{x_t}^a (a - x) \tan \alpha x dx \quad (8.21)$$

The solutions of the integrals of the above expressions depend on x_t .

- If $x_t = x_{t2} = a$, it indicates that the $\alpha \geq \alpha_{sliding}$, then:

$$F_y = \mu F_z \quad (8.22)$$

$$M_z = 0 \quad (8.23)$$

- If $x_t = x_{t1}$, i.e. $\alpha \leq \alpha_{sliding}$ then the integral has to be solved based on x_{t1} in equation (8.18), which results in:

$$F_y = \mu F_z \tan \alpha (3\theta - 3\theta^2 \tan^2 \alpha + \theta^3 \tan^3 \alpha) \quad (8.24)$$

$$M_y = -\mu F_z a \theta \tan \alpha (1 - 3|\theta \tan \alpha| + 3(\theta \tan \alpha)^2 - (\theta \tan \alpha)^3) \quad (8.25)$$

Where the parameter θ is defined as follows [23]:

$$\theta = \frac{2k_b a^2}{3\mu F_z} \quad (8.26)$$

And subsequently, t_p holds as follows:

$$t_p = \frac{a}{3} \left(\frac{1 - 3|\theta \tan \alpha| + 3(\theta \tan \alpha)^2 - (\theta \tan \alpha)^3}{(1 - |\theta \tan \alpha| + \frac{1}{3}(\theta \tan \alpha)^2)} \right) \quad (8.27)$$

8.2.3. Pure longitudinal slip

The basic principles for lateral and longitudinal slip are similar, except that the longitudinal deflection follows equation (8.28) [20]:

$$\varepsilon_x(x) = -V_{sx} \Delta t = (a - x) \frac{\kappa}{1 + \kappa} \quad (8.28)$$

where κ is the slip ratio as described in Chapter 3. According to the coordinate system, it holds that:

$$\kappa = -\frac{V_{sx}}{|V_x|} = -\frac{V_x - r_e \Omega}{|V_x|} \quad (8.29)$$

From the similarity between the bristle deflections in x and y directions, i.e. equations (8.12) and (8.28), it can be deduced that by replacing $\tan \alpha$ in equation (8.24) for F_y by $\kappa/(1 + \kappa)$, the force F_x can be derived as follows:

$$F_x = 3\mu F_z \theta \frac{\kappa}{1 + \kappa} \left(1 - \left| \theta \frac{\kappa}{1 + \kappa} \right| + \frac{1}{3} \left(\theta \frac{\kappa}{1 + \kappa} \right)^2 \right) \quad (8.30)$$

And accordingly the breakaway point x_{t1} can be described as follows:

$$x_{t1} = \frac{4k_b a^3}{3\mu F_z} \left| \frac{\kappa}{1 + \kappa} \right| - a \quad (8.31)$$

In Section 8.3, the implementation, possible improvement, and limitations of the model for the LAT100 compared with the tire is investigated.

8.2.4. Accuracy of the model

The presented model is already complex but still contains numerous assumptions. Besselink [20] calculated the errors of the presented brush model compared to actual tire measurements for F_y , F_x , and M_z in pure slip conditions to be 15%, 11%, and 73%, respectively. The error for the aligning moment M_z is notably high. To reduce the errors and improve the accuracy of the presented model for a real tire, the assumptions have to be enhanced, see later in section 8.3.3.

8.3. Implementation of the brush model

8.3.1. Model inputs

The presented brush model is now used to explain the physics and dynamics of the Grosch wheel as the test specimen of the LAT100 machine as shown in Figure 8-11. The internal structure of a real tire is much more complicated compared to the simple Grosch rubber wheel. The Grosch sample specification is illustrated in Figure 8-1; a solid rubber wheel with no air inside, while an actual tire is pneumatic with specific construction. This leads to different pressure profiles in contact with the counter-surface as explained in Chapter 7. For the Grosch wheel, the parabolic pressure distribution should be a good basis as a start to apply the brush model.

Due to the geometry of the wheel-shaped sample, the contact length $2a$ depends on the vertical wheel deflection δ as shown in Figure 8-4. By rising the normal load, δ increases. Assuming as an approximation a linear relation between F_N and δ with a spring constant k_z [20]:

$$F_N = k_z \delta \quad (8.32)$$

k_z can be termed the vertical stiffness factor of the tread compound which varies based on the elastic modulus of the rubber material. Shallamach [33] suggested a proportionality between the square of the contact length of the wheel with the counter-surface, and the ratio of normal load divided by Young's modulus of the rubber compound, i.e. $a^2 \propto (F_N/E)$. Besselink [20] introduced a pragmatic approach that matches better with the measurement data of vertical deflections of tires.

Figure 8-11: The velocities and the resultant forces of the Grosch wheel sample on the LAT100 counter-surface disc at slip angle α according to brush model and sign convention of Pacejka.

In the present chapter, the vertical stiffness factor is specified semi-empirically as shown in Table 8-1 as an example for compound A. It is based on the alteration of the contact length with the normal load using Fuji film pressure-sensitive film measured in 'static' mode, as described in detail in Chapter 7. From Figure 8-4 it can be derived that:

$$a = \sqrt{r_w^2 - (r_w - \delta)^2} \quad (8.33)$$

By substitution of a in equation (8.32), k_z becomes:

$$k_z = \frac{F_N}{r_w - \sqrt{r_w^2 - a^2}} \quad (8.34)$$

Figure 8-12 is a graphical representation of the variations of the experimental values of the contact length $2a$ and width $2b$ vs. normal load for compound A. Figure 8-13 describes the semi-empirical relation between F_N and δ , where the k_z can be extracted from the slope of a linear fit of the data. The variation of the width $2b$ with load is neglected, see Figure 8-12.

Table 8-1: Alteration of contact length with normal load, measured with pressure-sensitive film, type LLLW; 2a is identified.

Load	35 N	55 N	75 N	95 N	115 N
CompA

Figure 8-12: Alteration of the experimental values of the contact length $2a$ and width $2b$ with varying normal load for compound A.

Figure 8-13: A semi-empirical plot of normal load F_N vs. tread deflection δ .

The values for bristle stiffness k_b are taken as the storage moduli G of the rubber compounds from frequency sweep measurements with DMTA at 0.1% strain, assuming the geometry factor of the bristles is of order unity. The coefficient of friction μ is taken from the maximum value of the friction curve obtained with the LAT100. A summary of the modeling parameters for compound A is listed in Table 8-2. A similar set of parameters was also derived for the other compounds.

Table 8-2: Modeling parameters for the Grosch wheel from compound A, at speed 3.7 km/h.

Parameter	Symbol	Value	Unit
Wheel radius	r_w	41	mm
Wheel width	b	9	mm
Vertical stiffness	k_z	3.8×10^2	N/mm
Bristle stiffness	k_b	7.6	N/mm ² or MPa
Coefficient of Friction	μ	1.2	-
Half of the contact length at load 65 N	a	3.7	mm

8.3.2. Model outputs

Modeling was performed in two parts:

First, half of the contact length a was taken from ‘static’ measurements of contact area as illustrated in Figure 8-12; the input values for compound A are presented in Table 8-2. Figure 8-14 represents a comparison of the modeling data and the measurements with the LAT100: the lateral force F_y vs. slip angle α . Although the stiffness of the rubber compound and normal load were considered in the evaluation of the ‘static’ a , the outcome of the model is far from the experimental data: the red curve vs. the black. The model is very sensitive to bristle stiffness and therefore, the rubber modulus governs the friction curve up to the $\alpha_{sliding}$. The model accuracy can be altered significantly by the G value: for instance, assuming an hypothetical value of 15 MPa for G , as shown in Figure 8-14 with the dotted curve. Hence, obtaining a proper value for G is very important, as the modulus can change based on the type of test and its condition, see Table 8-4 in the Appendix.

Figure 8-14: Measurement results and modeling outputs of F_y vs. α for a 'static'; the modeling inputs are taken from Table 8-2. The dotted curve shows the influence of the bristle stiffness or G value of the rubber compound on modeling for compound A.

Second, an estimate of the a values is obtained using the LAT100 experimental data which is expected to represent the 'dynamic' contact length. From equation (8.14) at a very small $\alpha < 1^\circ$, half of the contact length a can be extracted:

$$a = \sqrt{\frac{C_{F\alpha}}{2k_b}} \quad (8.35)$$

By implementing the 'dynamic' approximate of a in the model, the outcome is presented in Figure 8-15 in comparison 'static'; the modification of a improves the model accuracy for the prediction of F_y remarkably. Note that beyond $\alpha_{sliding}$, the model follows equation (8.22) where the Coulomb approximation of the sliding region is clearly poor.

Figure 8-15: Lateral force F_y vs. slip angle α for compound A; implementation of the ‘a dynamic’ compared to the ‘a static’ with LAT100 measurement data at G of 7.6 MPa.

Figure 8-16 illustrates the outcome of the model, lateral force F_y vs. slip angle α , for the softest compound E presented in Table 8-4; the modeling parameters for compound E are listed in Table 8-5, Appendix. There is a vertical shift upwards for the experimental friction curve F_y vs. α at $\alpha = 0^\circ$ for compound E, which could indicate the presence of a pseudo-camber angle. Wheel camber or inclination angle γ is the angle between the planes of wheel and the normal to the counter-surface, see chapter 5. The shift can stem from the non-uniformity of the vulcanized sample. The Grosch wheel sample is 19 mm thick which inevitably leads to a gradient in crosslink density and consequent inhomogeneity of the specimen [16]. Still, by using the ‘a dynamic’ in the calculation, the deviation of the modeling output for compound E from the measurement result is larger than for compound A. Figure 8-17 shows a comparison of the lateral forces F_y , aligning moments M_z , and pneumatic trails t_p vs. slip angle α for compounds A and E.

Figure 8-16: lateral force F_y vs. slip angle α for compound E.

Figure 8-17: Lateral force F_y , aligning moment M_z , and pneumatic trail t_p vs. slip angle α for compounds A and E.

However, there is a subtle point concerning introducing the ‘ α dynamic’ in equation (8.35). The model outcome F_y vs. α becomes insensitive to variation of the G value for a particular compound. Similar to Figure 8-14, the model outcomes are presented for various hypothetical G values for compound A in Figure 8-18. The G value of 7.6 MPa was taken from Table 7-1; the data is presented as well for 3.8 MPa (0.5 G) and 15.2 (2 G). No change of F_y can be observed. This can be traced back to the $\alpha_{sliding}$ value calculated by equation (8.19). By substituting (8.35) in (8.19), the $\alpha_{sliding}$ value becomes invariant to the G modulus, it reads as follows:

$$\alpha_{sliding} = \tan^{-1}\left(\frac{3\mu F_z}{C_{F\alpha}}\right) \quad (8.36)$$

While the self-aligning moment M_z , and the pneumatic trail t_p differ based on the G values; the stiffer compound has a smaller pneumatic trail and needs a lower aligning torque. Based on this outcome, it can be concluded that F_y before reaching the $\alpha_{sliding}$ point is not a function of the tread stiffness. However, in reality, the tread stiffness affects the friction curve and $\alpha_{sliding}$ value as elaborated in Chapter 5. This is a limitation of the current assumptions for the model. To get closer to the experimental data, some ways are explained in the next section.

Figure 8-18: Lateral force F_y , aligning moment M_z , and pneumatic trail t_p vs. slip angle α for compound A; at various hypothetical bristle stiffness of 0.5G, G, and 2G presented with colors dark to light green, respectively; G value is 7.6 MPa.

8.3.3. Potential model improvements

As explained in section 8.2, the assumptions for the model may be enhanced to reduce the errors and improve the accuracy, e. g. by introducing:

- an elastic carcass;
- non-linear bristle deflections;
- the rubber volume as multiple-rows of deformable bristles;
- realistic pressure distribution in the contact area;
- the dependency of the coefficient of friction μ on slip speed and normal load by implementing available friction theories;
- deal with non-uniform carcass deflection;

vii. inclusion of inclination (camber) angle or path curvature.

In ‘Treadsim’ which is an improved brush model by Pacejka [37], some of the above improvements have been further developed for tires. The modeling in the present thesis was performed to create a basic understanding of the physics and dynamics of the Grosch wheel. These are the possibilities to improve the theory further.

8.3.4. Attribution of friction components

With the presented model, some insights into the dynamics and physics of the Grosch wheel as the specimen for the LAT100, the newly developed tribometer, were obtained. The attempt was not purely to discover the most accurate model, but to acquire a deeper understanding of the slippage regions i.e. stationary and non-stationary. This bridges the gap between wheel dynamics and rubber friction mechanisms. It creates a clear and intelligible profundity of friction curves in connection with rubber characterization and frictional mechanisms.

The stationary and non-stationary regions were identified in the model, as marked in Figure 2-9 and Figure 8-10B with magenta and blue colors, respectively. Each region encompasses specific contributions of frictional mechanisms, where different components are expected to govern and play the main role. In the **stationary** region, the tread deforms and stresses under shear which predominantly contributes to the adhesive friction component. The classical description of long-chain rubber polymers at the interface with a counter-surface dictates that by applying shear stress on the tread block, the rubber chains stretch, detach, relax, and re-attach to the counter-surface whereupon the cycle repeats itself [38]. In the **non-stationary** region due to the displacement, the tread blocks are subjected to a range of frequency excitations originating from the asperities of the surfaces, where viscoelastic or hysteresis friction could prevail. This can be measured with a sliding friction measurement and also Persson’s friction theory can calculate the viscoelastic part as explained in Chapter 2. The adhesion contribution can be obtained with a separate measurement designed for adhesion evaluation [39].

The friction curve can be split into three main sections concerning characteristic shape factors: linear, transitional, and frictional, see Figure 8-19. The linear section indicating cornering stiffness represents basic linear handling and stability of the tire; the transitional section is mainly affected by tread stiffness; the frictional range depends primarily on the viscoelastic properties. The advantage of this categorization

of the friction curve is that it provides information regarding the slippage regions and accordingly distinguishes the prevailing friction components.

Figure 8-19: Characteristic shape factors of the friction curve: 1. linear, 2. transitional, and 3. frictional; after Pacejka [40].

Full sliding starts at the peak of the friction curve and after that, the Coulomb equation in the model is a poor representation. For comparison of α and κ sweep tests, the main focus has to be dedicated to the sections before the peak occurs where the contributions of both regions i.e. stationary and non-stationary, envisage a high likeliness for different friction mechanisms. In reality, in ordinary driving, the passenger car tire gets an angle of $2^\circ - 6^\circ$ during cornering in bends and the peak typically occurs at $4^\circ - 7^\circ$ [22], therefore, both regions are important for tire friction. Also, another fact is that wider tires provide more grip because of the possibility to enlarge the stationary region which results in enhanced static friction.

8.3.5. Comparison of α and κ sweep tests

The sweep tests need to be compared at an equivalent scale or measure. This measure is defined as 'theoretical slip' for longitudinal and lateral slips. Table 8-3 shows a comparison of both slip types.

Table 8-3: Comparison of the equations derived from the brush model in longitudinal and lateral slip conditions.

Slip type	Pure longitudinal (x direction)	Pure lateral (y direction)
Characteristic parameter	κ	α
Slip definition	$\kappa = -\frac{V_{sx}}{ V_x }$	$\tan \alpha = -\frac{V_{sy}}{ V_x }$
Bristle deflection	$\varepsilon_x(x) = -V_{sx}\Delta t = (a - x) \frac{\kappa}{1 + \kappa}$	$\varepsilon_y(x) = -V_{sy}\Delta t = (a - x) \tan \alpha$
Theoretical slip, s	$s_x = \frac{\kappa}{1 + \kappa}$	$s_y = \tan \alpha$
At small slip	$F_x = C_{F\kappa}\kappa, \quad C_{F\kappa} = 2k_b a^2$	$F_y = C_{F\alpha}\alpha, \quad C_{F\alpha} = 2k_b a^2$
At large slip	$F_x = \mu F_z s_x (3\theta - 3(s_x\theta)^2 + s_x\theta ^3)$	$F_y = \mu F_z s_y (3\theta - 3(s_y\theta)^2 + s_y\theta ^3)$
$ s_{sliding} $	$1/\theta = \frac{3\mu F_z}{2k_b a^2}$	$1/\theta = \frac{3\mu F_z}{2k_b a^2}$

Theoretically, by assuming an equal coefficient of friction and bristle stiffness at both pure slip types, i.e. supposedly $\mu_x = \mu_y$ and $k_{bx} = k_{by}$, then:

- The equation for the longitudinal stiffness $C_{F\kappa}$ is identical to the cornering stiffness $C_{F\alpha}$;
- The theoretical slip at sliding occurs at the same θ for both types.

Hence, the equality of the friction curves is described in Figure 8-20.

Figure 8-20: Equality of friction curves at pure slip types plotted against theoretical slip; (left) longitudinal, and (right) lateral [23].

In reality, a notable discrepancy between the measured values for longitudinal and cornering stiffness is observed. Pacejka [23] indicated that $C_{F\kappa}$ is typically 50% larger than $C_{F\alpha}$ due to the lateral (torsional) compliance of the carcass of an actual tire. Besselink [20] reported the measured values of C_F in longitudinal and lateral tests as shown in Figure 8-21. As can be seen, the C_F for longitudinal and lateral slip conditions are influenced by the vertical load. Cornering stiffness may differ with tire vertical load, from 6 to 30 times according to Pacejka [23].

Figure 8-21: Comparison between $C_{F\kappa}$ and $C_{F\alpha}$ [20].

Still, it is expected that the qualitative similarity of both pure slip characteristics remains. By comparing actual tire data of α and κ sweep tests from Chapter 5, the portions of stationary and non-stationary regions in each test can be approximated by the model. An example of the comparison between both α and κ tests for the same tread compound is shown in Figure 8-22 where all other variables remained the same. The results are plotted vs. the characteristic slip parameters and the theoretical slips for the modeling and the measurement data. The right-side plots of each row describe the disparity between the α and κ tests: at equivalent theoretical slip, the curves show different characteristic shapes. This becomes more pronounced by plotting the portions of the stationary and non-stationary regions vs. theoretical slips in Figure 8-23. The ratios of the stationary to the non-stationary at equivalent theoretical slips for both α and κ tests are presented in Figure 8-24. For instance, at 5% slip, the stationary region in the α test is around 50% larger than for the κ test.

Figure 8-22: Comparison of α and κ tests at characteristic parameters and theoretical slips, first row: lateral sweep test, F_y vs. α and s_y , second row: longitudinal sweep test, F_x vs. κ and s_x ; for the same tread compound, C3 of Chapter 5.

Figure 8-23: Comparison of stationary and non-stationary regions at the theoretical slips for both α and κ tests for the examples of Figure 8-22.

Figure 8-24: Stationary/Non-stationary regions for α and κ tests for the examples of Figure 8-22.

The theoretical slip as employed here before was solely a simple approach to demonstrate the basic but important underlying reason for no correlation between α and κ sweep tests for tires. In reality, more factors play a role. When at equal theoretical slip the frictional forces differ, it implicitly suggests that the slip velocities in the contact area differ. Referring back to the theoretical slip equations in Table 8-3, these defined measures do not imply equivalent slip velocities. For instance, at a theoretical slip of 0.1, the slip velocities of the tire in α and κ tests are 6 km/h and 10 km/h, respectively. Furthermore, temperature and abrasion play a role. Figure 8-25 presents the

temperature profiles of α and κ tests at equivalent theoretical slip. Above 10% slippage, the temperature difference becomes significant. This impacts the whole scenario of comparison.

Figure 8-25: Temperature profiles of α and κ tests at equivalent theoretical slip for the examples of Figure 8-22.

Based on the current scenario, it was deduced so far that the α and κ sweep tests of tires are not correlating theoretically and experimentally, and respectively as the tire α data and LAT100 sweep measurements do correlate, see Chapter 5, a correlation between tire κ test and LAT100 sweep seems beyond the realm of possibilities for the acquired data.

8.3.6. Comparison of α sweep test of Grosch wheel and tire

The possibility of the tire grip prediction for both α and κ with the small rubber wheel is still highly attractive. If the underlying reasons for the correlation between LAT100 sweep and tire α test are fully comprehended, it might open the possibility to mimic the situation for tire κ test in a laboratory environment as well. Therefore, in the present section, the fundamental reasons for the correlations between the α sweep test of the Grosch wheel and tire are being investigated. It benefits the subject in two ways: first, the correlations are not purely experimental by relying on test conditions, which makes it more reliable to acquire correlations with a new set of tire data; and second,

the advantage of using a wheel shape specimen becomes more concrete to raise the possibility to also predict κ sweep.

The similarity between the test modalities of lateral sweeps of the tire and LAT100 seems to be the silver lining for the acquired correlations in Chapter 5. In the previous section, the comparison between tire α and κ tests was elaborated in-depth, the same approach can now be employed for the α friction curves of the Grosch wheel and the tire at an equal theoretical measure of s_y . Referring back to the acquired correlations of Chapter 5, the slip angle sweep ranges up to the peak points in the measured friction curves F_y vs. α of the LAT100 and the tire were different. The α friction curve of LAT100 approaches a plateau roughly above 20° , which can be seen for example in Figure 8-26 for the curves of F_y vs. α or s_y for compound C3 at a speed of 6.5 km/h and a normal load of 55 N. Accordingly, Figure 8-27 shows the stationary and non-stationary regions resulting from the modeling for the Grosch wheel. For the tire, the α friction curve reaches full sliding at around 7° , see Figure 8-22: F_y vs. α . The question is why the sweep ranges of the tire and LAT100 vary and more particularly with such a large difference, the α test of the LAT100 and the tire did still correlate.

With the aid of the previously described model, it should be possible to reply to this question or at least provide some basic understanding, as was implemented for explaining α and κ sweeps in the previous section. First, the beginning of the measured friction curve of the Grosch wheel shows a slight upward shift along the y-axis: see Figure 8-26. This could indicate implicitly the presence of a pseudo-camber angle as mentioned in section 8.3.2, which could be due to a non-uniformly vulcanized sample or an induced inclination angle owing to the path curvature. Second, due to this deviation, the $\alpha_{sliding}$ value for the Grosch wheel can be computed to be 30° (or $s_y = 0.53$), but the measurements suggested that full sliding occurred at 20° (or $s_y = 0.45$).

Figure 8-26: α sweep friction curve of the LAT100: F_y vs. α or s_y for the Grosch wheel at 6.5 km/h and 55 N for compound C3.

Figure 8-27: Comparison of stationary and non-stationary regions at the theoretical slip s_y for the Grosch wheel at 6.5 km/h and 55 N for compound C3.

The correlation between the tire α and the LAT100 sweep tests were achieved for the cornering stiffness C_f and peak values of friction curves F_y vs. α . For C_f and peak values, the main regions in the contact area are to a large extent stationary and non-stationary, respectively. The correlation between the LAT100 and tire α tests for the C_f values are because of the similarities between the friction mechanisms in the stationary regions of wheel samples, see section 8.3.4. By recapping the comparison between the tire α and κ tests, the ranking of the tread compounds for the stiffness parameters $C_{f\alpha}$

and $C_{f\kappa}$ are comparable in order; suggesting that the tire tests of α and κ in the primarily stationary regions correlate and have similar friction mechanisms, see Figure 8-28.

Figure 8-28: Ranking of tire α and κ sweep tests at small slip.

The theoretical slip s_y values of c.a. 0.17 and 0.53 for the tire and Grosch wheel do differ at the peak or α_{sliding} points that indicate that both samples are in the non-stationary region; it theoretically implies that no correlation can occur! Despite that, the α tests of LAT100 and tire experimentally did correlate at the peak values. This paradox can be solved by comparison of the slip velocities of the LAT100 Grosch wheel and the corresponding tire. The slip velocity in lateral test modality of the tire is defined as follows:

$$-V_{sy} = V_x \tan \alpha \quad (8.37)$$

Based on this equation and the tire data, at the 60 km/h velocity of travel and theoretical slip of 0.1, the tire slip velocity is around 5.8 km/h in the lateral direction. The velocity of the Grosch wheel 6.5 km/h which delivers the correlation with the tire data was 8 times smaller than the tire speed: see Chapter 5. Therefore, the measurement at lower velocity is compensated with sweeping to a larger range of slip angles to maintain a level of slip velocity that can induce full sliding or a full non-stationary region for the Grosch wheel. At an equivalent slip velocity of 5.8 km/h for the Grosch wheel, the slip angle for the traveling speed of 6.5 km/h can be calculated according to Eq. (8.37) to be 41° . This shows why LAT100 peaks at larger slip angles. It should be noted that in the measurements full sliding begins at around 25° slip angle. This discrepancy is due to the assumptions of the brush model. It is noteworthy that the

modeling was done under numerous assumptions. The contact width of the Grosch wheel, $2b$, is larger than the contact length $2a$; an implementation of multi-rows of bristles is more realistic and can provide more precise information regarding the deformation of the Grosch wheel. Furthermore, the curvature of the disc counter-surface path should be taken into account. The counter-surface disc curvature creates a distribution of slip velocities within the contact patch of the Grosch wheel as a function of α and r , the radius of an arbitrary point in the contact area to the center of the counter-surface disc, see Figure 8-2 and Eq. (8.3). The range of influence of the disc curvature on the variation of contact patch radius and slip velocity can be approximately computed for the Grosch wheel according to section 8.1.2, as shown in Figure 8-29; the farther the contact point from the center of the counter-surface disc, the larger the slip velocity. The difference between the highest and the lowest slip velocities ΔV_s in the contact area is influenced by input parameters of α and disc traveling velocity, see Figure 8-30. The ΔV_s can even reach 10 km/h at a very high speed and slip angle. It should be noted that the deflection of rubber with the sweep is disregarded in this calculation.

Hence, it is possible to highlight the underlying reasons for the obtained correlations between the Grosch wheel and tire by comparison of the slip velocities and normal load ranges (see also Chapter 7) and friction mechanisms concerning the slippage regions. And if these assumptions are correct, it should be possible to predict κ sweep with a small solid rubber wheel just by choosing the right test configuration and conditions. This could be obtained even though the experimental samples of the Grosch wheel were compared with tires with 25% contribution of the carcass construction and tread pattern (see Figure 8-6) for friction and the counter-surfaces being corundum-based vs. asphalt.

Figure 8-29: Influence of the path curvature on the variation of the radius of contact points to the center of disc counter-surface and slip velocity, at disc traveling speed 6.5 km/h and slip angle 25°; $2a$ Grosch contact length, $2b$: contact width.

Figure 8-30: The ΔV_s between the highest and the lowest slip velocities in the contact area vs. disc traveling speed and slip angle.

Last but not least, the radial thickness of around 25 mm for the LAT100 sample should be reduced, thus, the implementation of the brush model with the assumption of a rigid carcass for the Grosch wheel would be more realistic. This could be further optimized to be able to exploit the LAT100 as a versatile predictor for tire friction test modalities.

8.4. Conclusions

In the present chapter, by using the renowned physical ‘brush model’, the dynamics and physics of the Grosch rubber wheel on the LAT100 counter-surface were modeled. Its behavior in lateral and longitudinal movements, as opposed to a real tire, were investigated. The differences between the α and κ sweep test modalities were explained at a defined equivalent measure called theoretical slip. It was deduced that the type of test modality leads to different friction mechanisms in the contact patch even at similar test conditions. The underlying reasons for the correlation between LAT100 and tire α sweep tests were highlighted. The importance of a laboratory wheel sample geometry compared to a pure sliding body specimen could be substantiated by recognizing the two regions: a stationary and non-stationary status in the contact area similar to the tire. This could result in similar friction mechanisms for the Grosch wheel as a rolling tire. In reality in ordinary driving, the passenger car tire gets an angle of $2^\circ - 6^\circ$ on the road and the peak typically occurs at $4^\circ - 7^\circ$. This explains the importance of friction curves studied with the Grosch wheel because it provides insight into the stationary friction.

It was deduced that the α and κ sweep tests of tires are not correlating theoretically and experimentally, and respectively as the tire α data and LAT100 sweep measurements do correlate, a correlation between tire κ test and LAT100 sweep seems beyond the realm of possibilities. However, the present chapter provided information that it should be possible to predict κ sweep with a small solid rubber wheel just by choosing the right test configuration and conditions.

The present chapter suggests that modifications of the Grosch wheel could enhance the predictability of tire friction on the road, such as improvement of the geometry of the specimen. The radial thickness and width of the Grosch wheel can be optimized by utilizing a modified brush model where the assumptions are tailored to be closer to reality. Therefore, by using such an optimized small rubber wheel, it should be possible to predict tire grip in the laboratory for material development which is cost and time effective rather than testing tread compounds by building full-scale tires.

8.5. References

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8.6. Appendix: tread compound characterizations

Figure 8-31: The results of strain sweep measurements with DMA at 10 Hz and 40 °C for the five compounds, for more properties see Figure 7-1.

$$G = \frac{E}{2(1 + \nu)}$$

(8.38)

Table 8-4: The Physical and mechanical properties of the five tread compounds.

Properties	Comp A	Comp B	Comp C	Comp D	Comp E
Hardness (°Sh A)	71	67	68	66	61
Tensile strength (MPa)	15.7	16.8	18.6	20.4	17.7
Elongation at break (%)	500	315	565	580	585
G' (MPa) at 10Hz, 40°C, S6% ¹²	2.3	2.3	2.5	2.5	1.6
G' (MPa) at 10Hz, 40°C, S0.1% ¹³	13.8	7.4	9.5	9.3	5.4
G' (MPa) at 10Hz, 40°C, S0.1% ¹⁴	7.8	6.6	6.8	9.0	3.7
G' (MPa) at 1Hz, 40°C, S0.1% ¹⁵	7.6*	5.5	5.6	6.7	3.1
G' (MPa) at 10Hz, 40°C, S6% ¹⁵	3.1	2.7	3.1	2.7	2.3
G' (MPa) at 1Hz, 40°C, S6% ¹⁵	2.9	2.4	2.7	2.5	2.0

*The grey row is used for the modeling calculation.

Table 8-5: Modeling parameters for compound E.

¹² The data is taken from temperature sweep measurement.

¹³ The data is taken from strain sweep measurement.

¹⁴ The data is taken from frequency sweep measurement.

Parameter	Bristle stiffness	contact length/2
symbol	k_b (N/mm ²)	a (mm)
value	3.1	4.5

Derivation of Eq. (8.3) based on Figure 8-32: based on cosine law on the oblique triangle of (OAH):

$$R_A^2 = R^2 + |AH|^2 - 2R|AH| \cos(\alpha + \beta) \quad (8.39)$$

The magnitude of AH based on the coordinate of A point can be defined as follows:

$$|AH|^2 = x^2 + y^2 \quad (8.40)$$

The compound angle formula of $\cos(\alpha + \beta)$ reads as:

$$\cos(\alpha + \beta) = \cos \alpha \cos \beta - \sin \alpha \sin \beta \quad (8.41)$$

By substituting Eq. (8.40), Eq. (8.41), and definition of $\sin$ and $\cos$ based on the coordinate of point A, r can be derived as follows:

$$r(x, y, \alpha) = R^2 + x^2 + y^2 - 2R(y \cos \alpha + x \sin \alpha) \quad (8.42)$$

The deformation of rubber in the contact area is not considered.

Figure 8-32: An arbitrary point A with distance R_A from the center of the disc in the contact area of the Grosch wheel.

9 Summary

Tire grip is the primary determining factor in transmitting all the forces and moments originating from the vehicle to the road because tires are the only contact of a vehicle with the ground and the key consideration for driving safety. The tread compound is the eventual contact of a full tire with the ground. In order to improve the frictional properties of these rubber material to optimize tire grip: traction and slip properties, there is a continuous drive to find methods that allow a better simulation of the tribological characteristics of the tread compounds in early development stages in a laboratory environment. Full-scale tire tests are enormously sophisticated, time-consuming, and costly. It is, therefore, more sustainable and highly desirable to predict tire performance in a laboratory environment before having to manufacture a full tire, only for the determination of the frictional properties of the rubber materials used in the tire.

Considering all the different conditions that a tire may encounter during service, their simulation in one laboratory apparatus seems an impossible task. Many different test configurations have been developed over time with varying success. Among the various available tribometers suitable for assessing rubber frictional properties, measurements with a solid rubber wheel are most promising. The big advantage is the

rolling motion of the rubber specimen which resembles a tire on the road. This is due to the physical fact that the relative speed of a wheel's contact point to the ground is zero. The Laboratory Abrasion Tester (LAT) 100 is used for the prediction of tire performance making use of a rolling rubber wheel known as the Grosch wheel. The equipment was designed to simulate a wide range of tire service conditions to measure abrasion loss and frictional forces as a function of various tire-road service conditions such as slip, normal load, and speed on different counter-surfaces. The first purpose for the LAT100 was to measure rubber abrasion; therefore, its designation "Laboratory Abrasion Tester". Gradually it was realized that the apparatus could also be used as a tribometer to characterize frictional properties. Consequently, there is a steadily increasing interest to develop this concept further, which was the main purpose of the research presented in this thesis.

In previous studies, it has been demonstrated that tire performance properties i.e. **wet** grip, abrasion, and rolling resistance can be characterized with the LAT100, provided the right conditions for testing are employed. The question was raised whether testing of various tire compounds under **dry** conditions could also be employed to predict tire frictional properties in longitudinal as well as lateral conditions. This was taken as the original objective of the present work: various tire tread formulations, as well as different counter surfaces were included in the study to fulfill this goal. The results of LAT100 tests were to be correlated with actual tire tests with the same rubber tread formulations on standardized roads, under ABS-conditions, and different test modalities for the tires. Later on in the project, a new test configuration was designed and further developed by introducing a dynamic interface to the LAT100. This was set as a new objective to understand the machine as a tribometer because it provided tire service and operating conditions with a wheel-shaped specimen.

Chapter 1 encompassed a basic introduction to the work performed in this thesis. *Chapter 2* presented a survey of the literature on friction and tire grip concepts and a short introduction to the Persson's contact theory. Also, the available laboratory tribometers and tire testing methods were introduced and compared.

In *Chapter 3*, preliminary studies were conducted on the LAT100 to characterize the apparatus possibilities and limitations to be exploited for dry tire friction evaluations. The frictional forces of four different tread compounds were determined by collecting individual measurement points at defined test conditions on various counter-surfaces with different surface roughness properties. The results were correlated with tire ABS braking distance data. One of the LAT100 counter-surface discs with the designation

code 180 – composed of corundum particles based on sieve analyses according to ISO 525:2013(E) – reflected the present road surface conditions of the tire test in the best way with good correlation for the four tread compounds. The excitation frequency of this disk 180 was estimated to be closest to the range of hysteresis friction or road indentation. The current results showed that the lateral friction force decreases with increasing measured rms. Disc 180 was therefore chosen as standard counter surface among other LAT100 disc grades for further experiments.

In *Chapter 4*, the LAT100 was employed as a sliding friction tester with blocked wheels for deriving a predictive mathematical model using Design of Experiments for the input factors. Various operating parameters of the LAT100 as a tribometer on tire friction were investigated: normal load, speed, and slip angle. Measurements were performed on sandpaper P120 and LAT100 disc 180 counter-surfaces, using 5 tread compound formulations with a wide range of grip properties. The former surface is the test track that is widely employed for tire indoor flat-track tests. The variation of the coefficient of friction versus speed on the disc 180 was in good agreement with the ABS braking distance data of actual tires on the road, but not for the sandpaper substrate. At the specific frequency range, the power spectral density of disc 180 was closer to asphalt than to sandpaper P120. The optimal test conditions of load and speed were in similar ranges of a car tire. The mutual interactions of factors on tire friction were quantified, where the effects of single factors alone could not adequately be identified. The interaction of normal load and speed influences the coefficient of friction of tire tread compounds most clearly.

In *Chapter 5*, the LAT100 was re-designed and a new dynamic interface was introduced which opens the possibility to perform and mimic tire test modalities in a laboratory environment with enormously reduced test time duration. Comprehensive tire testing was performed to provide sufficient data for method validation. Strong correlations were obtained between the LAT100 results and the lateral tire test modality called α -sweep, but not for the longitudinal modality or κ -sweep. The approach described how to predict actual dry tire grip as a function of speed and load with this laboratory device which offered a positive new horizon to evaluate tire grip.

In *Chapters 6 and 7*, various counter-surfaces were analyzed using power spectral density. The real contact area was characterized using Persson's friction theory together with some measurements with pressure-sensitive films to obtain insight into the shape and magnitude of the contact between the rubber samples and the counter-

surfaces. These provided better explanations for the acquired correlations and also more comprehension of the tire grip concept.

And finally, *Chapter 8* was dedicated to understand the dynamics and physics of a rolling rubber wheel on a counter-surface disc of the LAT100 test step-up utilizing the physical “brush model” for tires. Therewith, the behavior of the rolling wheel in lateral and longitudinal movements was modeled in comparison to a real tire. A better comprehension of the intricate differences between the α and κ sweep test configurations with respect to slippage distribution in the contact area was obtained in association with rubber friction components: adhesion vs. hysteresis. It was deduced that both types of test modalities lead to different friction mechanisms in the contact patch even at similar test conditions. The underlying reasons for the correlation between the LAT100 and tire α sweep tests were highlighted. The importance of a laboratory wheel sample geometry compared to a pure sliding body specimen was substantiated by recognizing the two regions: a stationary and non-stationary status in the contact area analogues to the tire which results in similar friction mechanisms for the Grosch wheel as for a rolling tire.

The present thesis demonstrated that the idea of using a small rubber wheel in a right test modality smoothens the way to predict tire grip in the laboratory for material development which is cost and time effective, rather than testing tread compounds by building full-scale tires.

Samenvatting

Banden grip is de primaire bepalende factor voor de overdracht van alle krachten en momenten die door het voertuig op de weg worden overgedragen. Banden nemen het enige contact met de ondergrond voor hun rekening en spelen een hoofdrol bij de veiligheid van het rijden. Het rubberen loopvlak mengsel vertegenwoordigt het feitelijke contact van een volledige band met de grond. Teneinde de wrijvings/frictie eigenschappen van deze rubber materialen voor banden grip te verbeteren: de tractie en slip eigenschappen, is er een voortdurende streven om methodes te vinden die een betere simulatie mogelijk maken van de tribologische karakteristieken van loopvlak mengsels in vroege stadia van de ontwikkeling binnen een laboratorium omgeving. Grootschalige banden testen zijn enorm veelomvattend, tijdsverslindend en kostbaar. Het is daarom meer houdbaar en hogelijk gewenst om het gedrag van een band te kunnen voorspellen in een laboratorium omgeving, alvorens een volledige band te moeten maken enkel en alleen voor de bepaling van de frictie-eigenschappen van de rubber materialen gebruikt in de band.

Vanwege al de verschillende omstandigheden die een band doormaakt tijdens z'n gebruik, lijkt het simuleren daarvan in één laboratorium apparaat een onmogelijke opgave. Veel verschillende test configuraties zijn al ontwikkeld in het verleden met wisselend succes. Onder de verschillende beschikbare tribometers geschikt om de frictie-eigenschappen van rubber te karakteriseren zijn metingen met solide rubber wielen het meest veelbelovend. Hun grote voordeel is de rolbeweging van het rubber monster, lijkend op een band op de weg. Dit als gevolg

van het natuurkundige feit dat de relatieve snelheid van het contact punt van een wiel met de ondergrond nul is. De Laboratory Abrasion Tester (LAT) 100 wordt gebruikt voor voorspelling van banden-gedrag door gebruik te maken van een rollende rubber wiel, Grosch-wiel genaamd. Het apparaat is oorspronkelijk ontworpen om een brede spanne aan functionele omstandigheden van banden te simuleren door slijtage-verlies en frictie krachten te meten als functie van verschillende band-weg omstandigheden zoals slip, normaal kracht en snelheid op onderscheiden tegen-oppervlakken (counter-surfaces). Het eerste doel voor de LAT100 was om slijtage van rubber te meten; vandaar de aanduiding "Laboratory Abrasion Tester". Langzamerhand realiseerde men zich dat het apparaat ook gebruikt kon worden als tribometer om frictie-eigenschappen te karakteriseren. Dientengevolge is er momenteel een gestaag groeiende interesse om dit concept verder te ontwikkelen, hetgeen het hoofddoel van de research was zoals verwoord in het huidige proefschrift.

In eerdere studies is al aangetoond dat banden eigenschappen zoals natte grip, slijtage en rolweerstand kunnen worden gekarakteriseerd met de LAT100, mits de juiste testcondities worden toegepast. De vraag werd gesteld of testen van verschillende rubberen banden-mengsels onder droge omstandigheden ook kon worden toegepast om de frictie-eigenschappen van banden te voorspellen in longitudinale zowel als zijwaartse/laterale omstandigheden. Dit was dan ook het oorspronkelijke doel van het huidige werk: verschillende rubberen loopvlak-formuleringen naast verscheidene "counter surfaces" werden meegenomen in het onderzoek om aan het doel te voldoen. De resultaten van LAT100 testen moesten worden gecorreleerd met feitelijke banden testen met gebruikmaking van dezelfde rubberen loopvlak-formuleringen op genormeerde wegen onder ABS omstandigheden en verschillende test modaliteiten voor de banden. Later in het project werd een nieuwe test configuratie ontworpen en verder ontwikkeld door inbrengen van een dynamische interface in de LAT100. Dit werd vastgesteld als een nieuwe doelstelling, teneinde de machine als tribometer beter te begrijpen en omdat dit de mogelijkheid bood van simulatie van banden-onderhoud en bedrijfsomstandigheden met een wiel-vormig monster.

Hoofdstuk 1 omvatte een fundamentele inleiding in het verrichte werk in dit proefschrift. Hoofdstuk 2 gaf een overzicht van de literatuur op het gebied van frictie en banden grip-concepten, en een korte inleiding in de contact-theorie van Persson. Ook de beschikbare laboratorium tribometers en banden test methodes werden geïntroduceerd en vergeleken.

In Hoofdstuk 3 werden voorlopige studies uitgevoerd op de LAT100 om de mogelijkheden en beperkingen van het apparaat te karakteriseren, bedoeld om te worden gebruikt voor droge banden frictie evaluaties. De frictie krachten van vier verschillende loopvlak mengsels werden bepaald door het verzamelen van individuele meetpunten bij bepaalde test condities op verscheidene "counter surface" met diverse oppervlakte-ruwheid eigenschappen. De resultaten werden gecorreleerd met metingen van de droge ABS rem-afstand. Eén van de LAT100 "counter surface" schijven met de aanduiding 180 - samengesteld uit corundum-deeltjes gebaseerd op zeef-analyses volgens ISO 525:2013 (E) - stemde het beste overeen met de weg-oppervlak

omstandigheden tijdens de banden testen, resulterend in goede correlatie voor de vier loopvlak-mengsels. De excitatie-frequentie van deze schijf 180 lag het dichtst bij het gebied van hysteresefrictie of weg-indentatie. De resultaten toonden dat de zijwaartse frictie kracht afneemt met toenemende gemeten rms (wiel-omwentelingssnelheid). Schijf 180 werd daarom gekozen als standaard “counter surface” voor verdere experimenten uit de verschillende andere LAT100 schijven.

In Hoofdstuk 4 werd de LAT100 gebruikt als een glijdende frictie-tester met geblokkeerde wielen om een voorspellend mathematisch model af te leiden, met gebruikmaking van Design of Experiments voor de input factoren. Verschillende operationele parameters van de LAT100 als tribometer werden onderzocht voor banden frictie: normaal belasting, snelheid en sliphoek. Metingen werden verricht op schuurpapier P120 en de LAT100 schijf 180 als “counter-surface”, met vijf loopvlak mengsel-formuleringen over een breed gebied van grip eigenschappen. Het P120 schuurpapier is het test-oppervlak dat breed wordt toegepast voor banden testen in laboratoria met flat-track opstellingen. De variatie van de frictiecoëfficiënt versus snelheid op de 180-schijf kwam goed overeen met de ABS remweg gegevens van feitelijke banden op de weg. Dit was niet het geval voor het schuurpapier substraat. In het specifieke frequentie gebied lag de “power spectral density” van schijf 180 dichter bij dat van asfalt dan voor schuurpapier P120. De optimale test condities als belasting en snelheid lagen in vergelijkbare bereiken als voor een voertuigband. De onderlinge wisselwerkingen van factoren op de banden frictie werden gekwantificeerd, waarbij de effecten van afzonderlijke factoren alleen niet adequaat konden worden vastgesteld. De interactie van normaal belasting en snelheid tezamen beïnvloedt de wrijvingscoëfficiënt van een banden-loopvlak het duidelijkst.

In Hoofdstuk 5 werd de LAT100 aangepast en voorzien van een nieuwe dynamische interface, die de mogelijkheid opende voor het uitvoeren en simuleren van banden testmogelijkheden in een laboratorium omgeving met een enorme reductie in de duur van de testen. Uitgebreide banden testen werden uitgevoerd om voldoende gegevens te verzamelen voor validatie van de methode. Sterke correlaties werden verkregen tussen de LAT100 resultaten en de zijwaartse banden test modaliteit, α -sweep genaamd, maar niet voor de longitudinale modaliteit of κ -sweep. De werkwijze gaf aan hoe feitelijke droge grip te voorspellen als functie van snelheid en belasting met het huidige labortorium apparaat, hetgeen een positieve nieuwe horizon opende om banden grip te evalueren.

In Hoofdstukken 6 en 7 werden verschillende “counter surfaces” geanalyseerd gebruikmakend van hun “power spectral density”. Het werkelijke contact oppervlak werd gekarakteriseerd met Persson’s frictie theorie samen met enkele metingen met druk-gevoelige films, om inzicht te verkrijgen in de vorm en grootte van het contact tussen de rubber monsters en de “counter surfaces”. Deze gaven een betere verklaring voor de verkregen correlaties en meer begrip voor het banden grip concept.

En tenslotte, Hoofdstuk 8 was bedoeld om de dynamica en fysica van een rollend rubberen wiel te begrijpen op een “counter surface” schijf van de LAT100 test opstelling, door gebruik te

maken van het fysische “brush model” voor banden. Daarmee werd het gedrag van het rollende wiel gemodelleerd in zijwaartse en longitudinale bewegingen, in vergelijking met een echte band. Een beter begrip van de specifieke verschillen tussen de α - en de κ -sweep test configuraties werd verkregen met betrekking tot de verdeling van mate van slip in het contact oppervlak betrokken op de onderscheiden rubber frictie componenten: adhesie vs. hysteresie. Er werd afgeleid dat beide types test modaliteiten leiden tot verschillende frictie mechanismen in het contact vlak, zelfs bij vergelijkbare test omstandigheden. De onderliggende redenen voor de correlatie tussen de LAT100 en banden α -sweep testen werden afgeleid. Het belang van een laboratorium wiel geometrie vergeleken met een zuiver glijdend monster werd bevestigd door het herkennen van twee regionen: een stationaire en een niet-stationaire toestand in het contactvlak, analoog aan de band, hetgeen resulteert in vergelijkbare frictie mechanismen voor een Grosch wiel als voor een rollende band.

Het huidige proefschrift heeft aangetoond, dat het idee om gebruik te maken van een klein rubberen wiel met een juiste test modaliteit de weg effent naar voorspelling van banden grip in het laboratorium. Dit ten behoeve van materiaal ontwikkeling tegen lagere kosten en in minder tijd, vergeleken met testen aan loopvlak mengsels door complete banden te bouwen.

List of symbols

Chapter 2

a or α	Slip angle	(°)
A, A_c	Real or actual contact area	(m ²)
A_0	Nominal or apparent contact area	(m ²)
$C(q)$	Power spectra	(m ⁴)
$E(\omega)$	Viscoelastic modulus	(Pa)
E	Young's modulus	(Pa)
F	Force	(N)
H	Amplitude of asperity	-
L	Length of contact	(m)
F_N	Normal force	(N)
P	Nominal or apparent pressure	(Pa)
q	Wavevector or spatial frequency	(m ⁻¹)
R_e	Effective radius	(m)
$P(\zeta)$ or $p(q)$	Relative area at magnification ζ	-
V	Sliding velocity	(m/s)

V_x	Longitudinal driving speed	(m/s)
γ	Empirical parameter	-
κ	Longitudinal slip ratio	-
λ	Wavelength of asperity	(m)
μ	Coefficient of friction (COF)	-
μ_s	Static coefficient of friction	-
μ_k	Kinetic coefficient of friction	-
μ_A	Adhesion friction coefficient	-
μ_D	Deformation friction coefficient	-
μ_H	Hysteresis friction coefficient	-
ζ	Magnification parameter	-
σ_0	Mean perpendicular pressure	(Pa)
ν	Poisson's ratio	-
ω	Excitation frequency	(s ⁻¹)
Ω	Wheel rotational velocity	(rad/s)

Chapter 3

d	distance	(m)
F_f or F_x	Tangential friction force	(N)
F_s or F_y	Lateral (Side) force	(N)
F_c	Counter force to centrifugal force (radial)	(N)
F_z	Normal force	(N)
R	Radius	(m)
R_e	Effective radius	(m)
R^2	Linear regression coefficient	-
SFC	Side Force Coefficient	-
V_c	Circumferential velocity	(m/s)
V_t	Traveling velocity	(m/s)
V_x	Longitudinal driving speed	(m/s)
V_s	Slip velocity	(m/s)
α	Slip angle	(°)
Ω	Wheel rotational velocity	(rad/s)

Chapter 4

α	Slip angle	(°)
A	Factor A: load	(N)
$ANOVA$	Analysis of Variance	-

B	Factor B: speed	(km/h)
C	Factor C: Slip angle	(°)
CC	Correlation Coefficient	-
COF	Coefficient of Friction	-
DF or df	Degree of Freedom	-
F_N	Normal force	(N)
F_f	Friction force	(N)
F -ratio	Variance ration	-
MS	Mean Square	-
n	Number of experiments	-
P -Value	Probability Value	-
RSM	Response Surface Modeling	-
R -sq	R-squared	-
R -sq(adj)	R-squared adjusted	-
R -sq($pred$)	R-squared predicted	-
S	Standard deviation	-
SS	Sum of Squares	-
S^2	variance	-
SE	Standard error	-
$S_{\bar{y}}$	Standard Error of the mean response	-
t -value	dimensionless statistical scale	-
X	Input factor	-
XX	2-way interaction term	-
X^2	Quadratic term	-
Y	Predicted response	-
$\bar{Y}$	Mean response	-
V_t	Disc traveling velocity	(m/s)
α	Significance level	-
$(1-\alpha)$	Confidence interval	(%)
β	Model coefficient	-
μ	Coefficient of Friction	-
ω	Disc rotational velocity	(rad/s)

Chapter 5

CV	Coefficient of variance or relative standard deviation	(%)
F_c	Counter force to centrifugal force (radial)	(N)

F_f or F_x	Tangential friction force	(N)
F_s or F_y	Lateral (Side) force	(N)
F_z	Normal force	(N)
R^2	Linear regression coefficient	-
R_e	Effective radius	(m)
V_c	Circumferential velocity	(m/s)
V_s	Slip velocity	(m/s)
V_t	Traveling velocity	(m/s)
V_x	Longitudinal driving speed	(m/s)
α	Slip angle	(°)
κ	Longitudinal slip ratio	-
μ_x	longitudinal coefficient of friction	-
μ_y	lateral coefficient of friction	-
Ω	Wheel rotational velocity	(rad/s)

Chapter 6

$A(\tau)$	Auto-correlation function	-
$C_{2D}(\mathbf{q})$	2D power spectrum	(m ⁴)
$C_{1D}(\mathbf{q})$	1D power spectrum	(m ³)
H	Hurst exponent	-
L	The length of the measurement	(m)
PSD	power spectral density	-
$\mathbf{q}$	Wavevector	(m ⁻¹)
q	wavenumber	(m ⁻¹)
q_0	Small cut-off wavevector (long roll-off wavelength)	(m ⁻¹)
q_1	Large cut-off wavevector (short wavelength)	(m ⁻¹)
R_a	The arithmetic average of the absolute values of the distance of the asperities to the mean line	(μm)
R_q	Root mean squared of the distance of the asperities to the mean line	(μm)
R_{sq}	Skewness	-
x	Distance in x-direction	(m)
z	Height of asperities	(m)
ΔM	Difference between the maximum and minimum torque	(dNm)
λ	Wavelength	(m)
τ	Distance defined in the autocorrelation function	(m)

Chapter 7

A_0	Nominal or apparent area of contact	(mm ²)
A_b	The area of the base of frustum	(mm ²)
A_c	Actual area	(mm ²)
A_l	Lateral area	(mm ²)
A_p	Projected area	(mm ²)
A_{llw}	area on film LLLW	(mm ²)
A/A_0	the relative contact areas	-
E	modulus	(MPa)
f	frequency	(Hz)
h	Depth of penetration	(mm)
h_2	Height of the frustum	(mm)
H	Hardness	(°Sh A)
LLW	upper low pressure	-
$LLLW$	ultra-super low-pressure film	-
P_0	Nominal pressure	(MPa)
q_1	large cut-off wavenumber	(m ⁻¹)
r	Frustum radius	(mm)
S	Lateral length of frustum	(mm)
T	Temperature	(°C)
σ_0	Nominal pressure	(MPa)

Chapter 8

a	Half of contact length	(m)
b	Wheel width	(m)
$C_{F\alpha}$	Cornering stiffness	(N/rad)
$C_{F\kappa}$	Longitudinal stiffness	(N)
Δt	Time interval	(s)
E	Young's modulus	(MPa)
F_f or F_x	Tangential friction force	(N)
F_c	Counterforce to centrifugal force (radial)	(N)
F_s or F_y	Lateral (Side) force	(N)
F_z or F_N	Normal force	(N)
G	Shear modulus	(MPa)
k_b	Bristle stiffness	(N/m ²)
k_z	Vertical tread stiffness	(N/m)

M_z	Self-aligning torque or moment	(N.m)
$q_y(x)$	Force per unit in y direction	(N/m)
$q_x(x)$	Force per unit in x direction	(N/m)
$q_z(x)$	Force per unit in z direction	(N/m)
r	Radius of an arbitrary point in the contact area to the center of the counter-surface disc	(m)
R	Distance between the center of the Grosch wheel contact area and the disc center point in static mode	(m)
r_e	Effective radius of the rubber wheel	(m)
r_l	Static loaded radius	(m)
r_w	Unloaded radius	(m)
s	Traveled distance	(m)
s_x	Theoretical slip in longitudinal direction	-
s_y	Theoretical slip in lateral direction	-
T	Temperature	(°C)
t_p	Pneumatic trail	(m)
V	Velocity of travel	(m/s)
V_c	Circumferential velocity	(m/s)
V_r	Linear velocity of carcass	(m/s)
V_s	Slip velocity	(m/s)
V_{sx}	Slip velocity in x direction	(m/s)
V_{sy}	Slip velocity in y direction	(m/s)
V_t	Traveling velocity	(m/s)
V_x	Longitudinal driving speed or forward component of V	(m/s)
x_t	Break-away point	(m)
α	Slip angle	(°)
$\alpha_{sliding}$	The slip angle when the stationary region vanishes and full sliding occurs	(°)
δ	Deflection	(m)
$\varepsilon_x(x)$	Bristle deflection in x direction	(m)
$\varepsilon_y(x)$	Bristle deflection in y direction	(m)
$\varepsilon_{max}(x)$	Maximum bristle deflection	(m)
γ	Camber or inclination angle	(°)
κ	Longitudinal slip ratio	-
μ_x	Longitudinal coefficient of friction	-
μ_y	Lateral coefficient of friction	-
μ	Coefficient of friction	-

θ	A dimensionless parameter defined for simplification	-
ω	Counter-surface disc rotational velocity	(rad/s)
Ω	Wheel rotational velocity	(rad/s)

List of tire codings

Tire data set	1	2	3
Number of treads	4	5	6
Tire sizes	205/55R16	225/50R17	235/35R19
Types of test modality	ABS braking	ABS braking	α and κ sweeps
Codings	Tread1, Tread2, Tread3, Tread4	A, B, C, D, E	C1, C2, C3, C4, C5, C6

Chapter	Used tire data set	Codings
3	1	Tread1, Tread2, Tread3, Tread4
4	2	A, B, C, D, and E
5	3	C1, C2, C3, C4, C5, C6
7	2	A, B, C, D, and E
8	2 and 3	A and E, and C1 to C6

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8. Salehi, M., J.W.M. Noordermeer, L.A.E.M. Reuvekamp, A. Blume, Characterization of the real contact area of solid rubber wheels vs. real tires with pressure-sensitive films in comparison with Persson theory, under submission.

Conferences and presentations

9. Salehi M., M. Heinz, T.V. Tolpekina, T.G.G. Teunissen, Louis A.E.M. Reuvekamp, Wilma K. Dierkes, and A. Blume. Prediction of Tire Dry Grip by Using the Laboratory Abrasion Tester (LAT100). in 12th Fall Rubber Colloquium 2016. 22-24 Nov., Hannover, Germany.
10. Salehi, M., J.W.M. Noordermeer, L.A.E.M. Reuvekamp, and A. Blume. Correlation between the tire grip and LAT100 results. in 4th International Conference Science meets Tires 2018: Perspectives for Tire Technology. 24- 25 April, Institute for Automotive Engineering (ika) and RWTH Aachen University, Aachen, Germany.
11. Salehi, M., J.W.M. Noordermeer, L.A.E.M. Reuvekamp, and A. Blume. Prediction of the Dry Grip, using LAT100. in Tire Technology Expo & Conference 2017. 14-16 Feb, Hannover, Germany.
12. Salehi, M., J.W.M. Noordermeer, L.A.E.M. Reuvekamp, and A. Blume. A New Method to Rapidly Predict Tyre Grip In a Laboratory Environment. in International Rubber Conference (IRC). 3-5 Sep 2019. London, UK.
13. Salehi, M., J.W.M. Noordermeer, L.A.E.M. Reuvekamp, and A. Blume. What happens in the tire contact area? in Tire Technology Expo 2020, 24-26 Feb. Hanover, Germany.
14. Salehi, M., J.W.M. Noordermeer, L.A.E.M. Reuvekamp, and A. Blume. Evaluation of Tire Grip in a Laboratory Environment: Introducing a Rapid Test with LAT100 in Correlation with Actual Tire Data on Dry Road. in ICTL 2020: 14th

International Conference on Tribology and Lubrication. June 11-12, Tokyo, Japan.

15. Salehi, M., J.W.M. Noordermeer, L.A.E.M. Reuvekamp, and A. Blume. An investigation into the effect of surface roughness with LAT100 in correlation with tire data. in Dutch Polymers Days (DPD), 14-15 March 2017. Lunteren, Netherlands.
16. Salehi, M., J.W.M. Noordermeer, L.A.E.M. Reuvekamp, and A. Blume. Prediction of the dry grip, using Laboratory Abrasion Tester (LAT100) in DKG West meeting (Deutsche Kautschuk-Gesellschaft), 3rd April 2017, Bad Neuenahr, Germany.
17. Salehi, M., J.W.M. Noordermeer, L.A.E.M. Reuvekamp, and A. Blume. A new method to rapidly predict tire grip in a laboratory environment in 4th Annual Innovative All-season and Winter Tire Development Forum, 27th – 28th May 2019, Amsterdam, Netherlands.

Collaborations

18. Dirk Schuttel (author), Salehi, M. (interviewee), 'Advancing abrasion testing', Tire Technology International, 2019, The annual review of tire materials and tire manufacturing technology, pp 88-10.
19. Sekar, P., Salehi, M., Anyszka, R. P., and A. Blume. 'HydroThermally Carbonized lignin as sustainable filler for tire-tread application' in Tire Technology Expo 2020, 24th -26th Feb. Hanover, Germany.
20. Sekar, P., Salehi, M., Anyszka, R. P., and A. Blume. 'Hydrothermally Carbonized Lignin as Sustainable Bio-based Filler for Tire-Tread Applications', under submission.

Honors and Awards

1. Honored Cum Laude for the PDEng thesis, Professional Doctorate in Engineering, University of Twente, July 2017.
2. Awarded for the best poster prize in IRC 2019, London, UK.
3. The paper "A New Horizon for Evaluating Tire Grip Within a Laboratory Environment" was selected as best from the Society of Tribologists and Lubrication Engineers (STLE) in the July edition of Tribology & Lubrication Technology magazine in the article Editors' Selections.

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خانواده گلم، شما عزیزای دلمین، خیلی دوستتون دارم، خیلی عاشقتونم. شما بهترین خانواده‌ای هستین که من هرگز می‌تونستم داشته باشم.

Proper tire grip provides a good level of handling which is a prerequisite for a vehicle's steering, braking, accelerating, and cornering. For material development for the acquisition of superior tire performance, full-scale tire testings are enormously sophisticated, time-consuming, and costly. It is sustainable and highly desirable to predict tire grip performance in a laboratory environment before manufacturing a full tire for the determination of rubber friction. However, considering all the influential factors in one laboratory apparatus is a complex task to accomplish, to predict tire tribological properties. In the present thesis, a new way of measurements in a laboratory environment is introduced which is in good agreement with actual tire testing.

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