

Prediction of Brake Pad Wear based on real Driving Profiles using a Pin-on-Disc Tribotester

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ABSTRACT: The possibility of predicting brake pad wear is increasing in relevance for various reasons. A reliable wear prediction allows brake pads to be characterized more precisely in terms of emission behavior and service life. On the other hand, pads can be sized in a more purposeful way in early development stages. The dimensioning of brake pads plays a particularly important role in electric vehicles, as these are much heavier than conventional combustion vehicles due to their battery, but at the same time have the possibility of recuperation. Accordingly, the number of brake applications and the load spectrum of the brake system are also undergoing changes.

Pin-on-Disc Tribotesters are especially suitable for the investigation of brake pad wear, as they represent an economical possibility for the specific investigation of mechanisms in tribological contacts. In addition, the Automated Universal Tribotester (AUT) of the Institute for Dynamics and Vibration (IDS) offers the opportunity for high-precision quasi in-situ measurements of the wear height.

In order to achieve a systematic understanding of the wear behaviour of brake pads, stationary test cycles were performed at the AUT. Due to the great importance of the friction history with regard to the performance of automotive brake systems (see e.g. [1, 2]), it is essential, however, that the same load cycles as in real vehicles are metrologically investigated. For this purpose, the software of the AUT has been expanded, whereby real driving profile data from motor vehicles can be reproduced on the test bench.

From the information obtained by the stationary and instationary measurements, a wear model can be derived, which enables the estimation of the service life of brake pads in electrified vehicles during the early development process.

KEY WORDS: wear prediction, pad wear characterization, driving profile measurements, tribotester, reservoir dynamics

1. Introduction

The friction brake is essential for a quick and reliable braking of the vehicle in safety-critical driving situations. However, since its performance can be significantly influenced by external factors, the automotive industry invests millions of Euros in research and engineering of brake systems each year - and the trend continues to rise [3]. The objectives in the brake design are to maximize the coefficient of friction, a high robustness with regard to NVH phenomena and a low wear rate [4].

Due to the increasing electrification of the motor vehicle (BEV), the load characteristic of the friction brake changes in contrast to the internal combustion engine vehicle (ICEV). On the one hand this can be explained by the higher weight of BEV and on the other hand by the concomitant advantage of recuperation. As a result of recuperation, most of the braking operations no longer have to be performed by the friction brake. This leads to less frequent use of the friction brake and thus to lower temperatures in the brake system [5]. Therefore, new questions arise especially with regard

to the dimensioning of the brake system and the associated wear of the brake pads.

The following investigations were carried out in cooperation with the AUDI AG based on the initial assumption that conventional disc brake systems would still retain their key function as a safety-relevant component in BEV, but could also be optimized in terms of weight reduction and resource conservation. The aim of this paper is to estimate the wear of brake pads in BEV. This requires the design and testing of a comprehensive test procedure. The tests were carried out on the "Automated Universal Tribotester" (AUT) of the Institute for Dynamics and Vibration (IDS). Advantage of the tribometer is the comparatively low-cost, fast and precise clarification of wear phenomena in comparison to road tests.

The results of the parameterization are to be used for the dimensioning of brake pads for use in BEV and thus contribute to the general optimization of BEV.

2. The Automated Universal Tribotester (AUT)

As mentioned in the introduction, the AUT of the IDS (described detailed in [6]) is very well suited for investigating the wear behavior of brake pads and is therefore predestinated for the aimed research objectives. The tribometer is based on a load unit that can be moved by linear stages. A brake pad sample (pin) is attached to the unit. Behind the sample a three-axial piezo sensor is placed, which records the forces in all three spatial directions. By the initial recording of a position-controlled force characteristic curve, the pin can be pressed against a rotating brake disc with a defined force (see Figure 1).

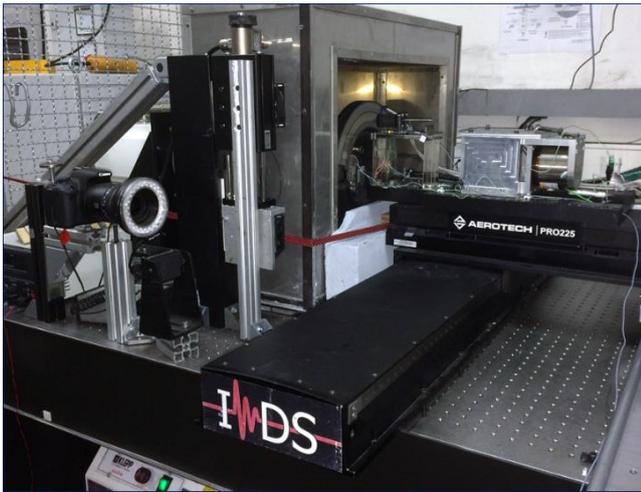


Figure 1 The Automated Universal Tribotester [6]

The load unit is designed to transmit the movement of the linear stages into a normal force [7]. A self-adjusting adaptronic system was also developed to compensate the tilting of the pin dynamically under varying loads [8, 9].

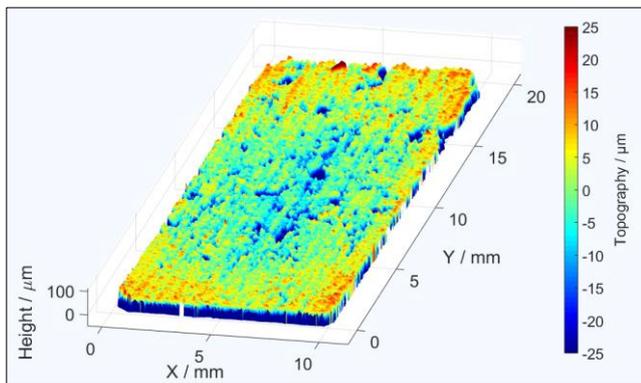


Figure 2 Surface topography recorded by the high precision surface measurement unit of the AUT [2]

A key element for wear investigation is the developed unit for automated recording of the brake pad surface topography, which can be used inbetween friction applications without disassembling the sample due to the linear stages [10]. The measurement is performed by an oscillating single-point laser triangulator with a height resolution of 0,15 μm. A total of one million data points is collected from the 20 x 10 mm² sample. After evaluation of the data point cluster, statements regarding the wear height and wear

volume can be made, and with a reliable density information also about the wear mass (see Figure 2).

In summary, a comprehensive and efficient wear investigation of brake pad samples can be carried out with the combination of the measuring equipment presented here. For this purpose, a suitable test procedure is developed in the following.

3. Measurements with Stationary Test Procedure

3.1. Development of a Test Procedure

Robust test procedures must be systematic and, if possible, reproducible under the same conditions, but they must also cover a parameter range that is close to the application or practice. The advantage of a systematic procedure is based on the comparability of results. To develop such a procedure, appropriate experience in measurement planning is required. Regarding the mentioned parameter range, the AUT provides a wide bandwidth of equivalent brake line pressures between 1 bar and 46 bar. The rotational speed ranges from 0 rpm to 1.500 rpm, which corresponds to a sliding speed of approx. 25 m/s and vehicle speeds of approx. 180 km/h. The duration of friction can be selected as required and various other functions are available, such as the possibility of force and speed ramps (incl. stop braking) or external heating of the brake disc and brake pad to up to 200°C.

Probably the most important interface between test procedure and test objective is the metrological recording of the relevant parameters for wear prediction. One of the most simple wear models known to date is that of J. F. Archard [11]. He describes the generated wear volume V_V as a function of normal force F_n and sliding distance s . By combining the measured wear volume and the specified system parameters, the so-called wear coefficient k can finally be determined, which reflects the wear rate of a material pairing. Since the wear coefficient of this simple model contains all information relevant for the wear prediction that is not captured by the system parameters F_n and s , various researchers have attempted to develop a more precise wear model. Most of them also use a wear factor k and a combination of the system parameters force, sliding velocity and friction duration [12-20].

S. K. Rhee, for example, has derived an empirical model which has been specially designed for brake applications [13, 14]. It defines the wear mass calculated by the normal force, the relative speed and the friction duration. However, all three system parameters have been given an exponent which allows the influence of the individual system parameters to be weighted:

$$V_m = k * F_n^a * v_{rel}^b * t^c \quad (1)$$

The wear is therefore parameterized by the three exponents a , b , c and the wear coefficient k . However, further investigations by S. K. Rhee in cooperation with T. Liu have shown that this model is no longer suitable for high temperatures above 230°C, since wear increases exponentially [15]. Such simple models therefore do not provide an appropriate solution for predicting wear in ICEV brake systems, where temperatures may exceed 800°C in critical situations. Dynamic friction models, such as that of G.-P.

Ostermeyer, for example, only consider such wear models by taking into account the existing temperatures [21].

In contrast to the ICEV, significantly lower temperatures are assumed in the brake contact of BEV due to the recuperation of the electric motor and the resulting smoother load conditions [5]. The influence of temperature is therefore neglected in the following. Initial measurements have furthermore shown that Archard's wear model provides good results at constant normal force and friction duration combined with a two-speed variation [22]. The procedure developed here claims to investigate a wider parameter space on the one hand and a higher number of parameter combinations on the other hand to investigate wear more comprehensively. For this purpose, Archard's wear model is slightly modified by considering all relevant parameters separately and relating them to the modal wear height of the entire brake pad sample (see Figure 2):

$$V_h = k * F_n * v_{rel} * t \tag{2}$$

The variation of all three system parameters with respect to a basic combination (also called base) provides a strict separation of the influences caused by normal force, velocity and friction duration [13]. Furthermore, it enables the evaluation of the obtained measurement data with various other wear models (see above).

The performance range of the AUT was used to define the parameter range (see Figure 3). Accordingly, the sliding speed is increased in 40 steps of 0,63 m/s between 0,67 m/s and 25,24 m/s. In the second block, the friction duration is increased by 1 s between 1 s and 40 s. The surface pressure is in the range of 0,05 N/mm² up to 2,0 N/mm² and is increased in the final block in steps of 0,05 N/mm². The base was set moderately at approx. 25 % of the machine limits, since experience shows that this system parameter combination leads to an average filling degree of the surface reservoirs [2]. Depending on the objective of the specific investigation, it may be useful to apply an increased base for certain system parameters, e.g. as described in [23].

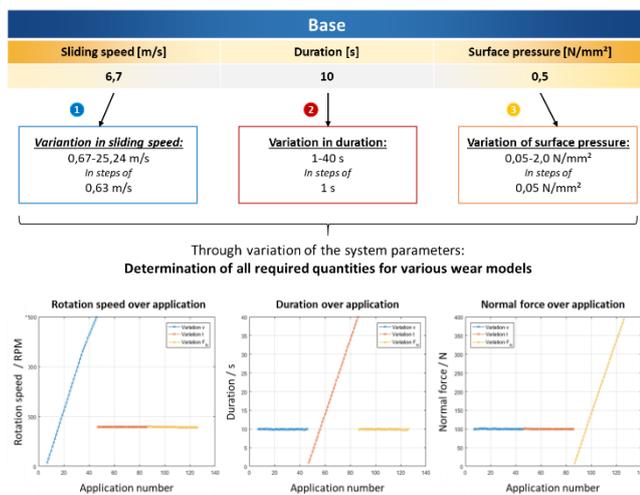


Figure 3 Stationary Test Procedure

3.2. Exemplary results

The advantage of the procedure shown in Figure 3 is not just the comparability of different tests. Due to the common base and the equivalent step sizes, all three sections of variation have the same Idle Work IW (3) in sum.

$$IW = F_n * v_{rel} * t \tag{3}$$

The wear coefficient k is therefore calculated across all sections as a function of the IW and the modal height wear V_h via a linear polynomial. Figure 4 shows exemplary measurement results with the stationary test procedure. A friction pairing was measured three times in a row to investigate the reproducibility of the results. The wear factors according to Archard (see Equation 2) are $k = -2,0292 \cdot 10^{-8}$ mm/Nm for the first measurement, $k = -2,3365 \cdot 10^{-8}$ mm/Nm for the second measurement and $k = -2,0408 \cdot 10^{-8}$ mm/Nm for the third measurement. The mean value correspondingly results in $k_{mean} = -2,1355 \cdot 10^{-8}$ mm/Nm, which means that the maximum deviation of the measurement results from the mean value is 9,4 %.

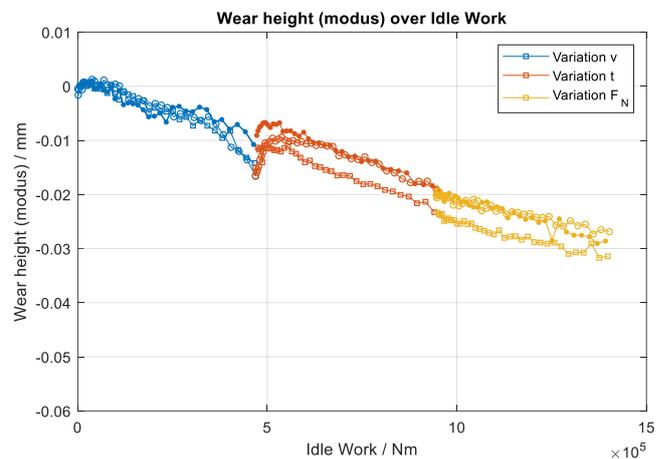


Figure 4 Wear results with Stationary Test Procedure

Further friction pairings also showed deviations of < 10,0 % and < 5,0 % after repeating the stationary procedure, which indicates a very good reproducibility of the wear measurements on the AUT. A particularly obvious characteristic is the almost linear wear over the idle work, so that Archard's model can be approximated with the stationary procedure. The wear height which decreases at higher speeds and increases abruptly at the beginning of the variation in friction duration offers a further indication of the Surface Reservoir Dynamics postulated by Ostermeyer [2].

Considering the corresponding coefficients of friction (Figure 5), a decreasing course can be seen in the first section, which increases continuously beginning with the 27th application (15,25 m/s sliding speed) until the start of the force variation.

A direct influence of the coefficient of friction upon the wear gradient can therefore not be identified here. Especially in the section of speed variation, however, the surface shows quite significant changes with respect to the topographic signature for most friction pairings. According to [2], this can be related to the development of the coefficient of friction.

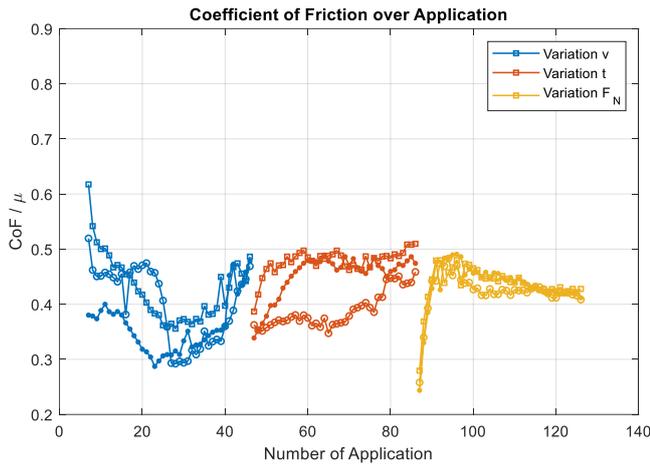


Figure 5 Friction results with Stationary Test Procedure

Table 1 summarizes the mean wear coefficients of a total of eight measured friction pairs (2nd column). The range of the coefficients is thus between $k = -1,5751 \cdot 10^{-8}$ mm/Nm and $k = -2,5711 \cdot 10^{-8}$ mm/Nm. In spite of the very good reproducibility, the parameters determined cannot straight away be used for a wear prediction. An important aspect that is not considered in the stationary procedure in comparison to the real driving scenario is the dynamics of braking. For this reason, the AUT is expanded in the following chapter to include the feature of driving profile measurement. Using real BEV driving profile data from AUDI AG, braking scenarios can accordingly be implemented on the AUT of the IDS and investigated with regard to wear.

4. Real Driving Profile Measurements

4.1. Driving Profile Measurement Feature

The implemented driving profile measurement feature of the AUT enables the import of real vehicle brake data from field tests, including brake line pressure, vehicle speed and friction duration. An advantage of this feature is that the dynamics of real braking and the associated influences on friction and wear are considered in the investigations. For this purpose, the raw profile data must first be converted into a format that contains all relevant braking data and can be read by the tribometer. The first step in filtering the data is the separation into individual brake applications. This is done using the status of the brake trigger. This means that if the trigger jumps from "1" (brake lever actuation) to "0" (no brake lever actuation), braking is complete and vice versa. Another filtering is the decoupling of the procedure from residual braking torques, which is why a minimum normal force of 10 N has been defined for the tribometer. Concerning the velocity, there is, as mentioned above, an upper limit of about 180 km/h. Accordingly, each procedure contains a mix of stop and drag brakings over a wide speed range. The transmission of a starting temperature for the brake disc is also possible, but negligible for BEV brake systems, since the driving profile data show brake disc temperatures below 70°C [5].

In the processed state each importable procedure contains the relevant force, velocity and braking duration data at a provision rate of 2 Hz (Figure 6).

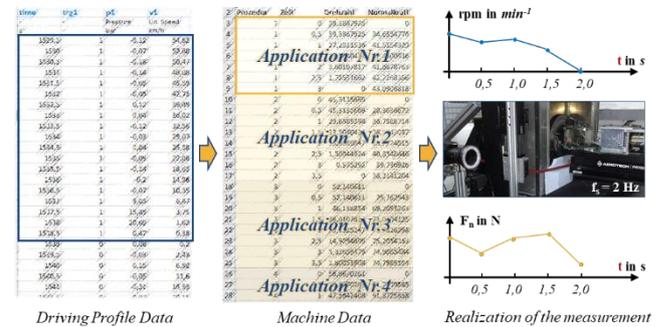


Figure 6 Transfer of Driving Profile braking applications to the AUT

Figure 7 illustrates a processed driving profile in various diagrams. For this purpose, all relevant braking actions were extracted from the raw data and then connected in sequence. At the top left and top right (red curve) the brake pressure and the equivalent normal force on the tribometer are plotted over time. The plots with the green curve show the vehicle speed over time on the left and the sliding speed in contact over time on the right. The two lower plots additionally display the idle work isolated and cumulated over time.

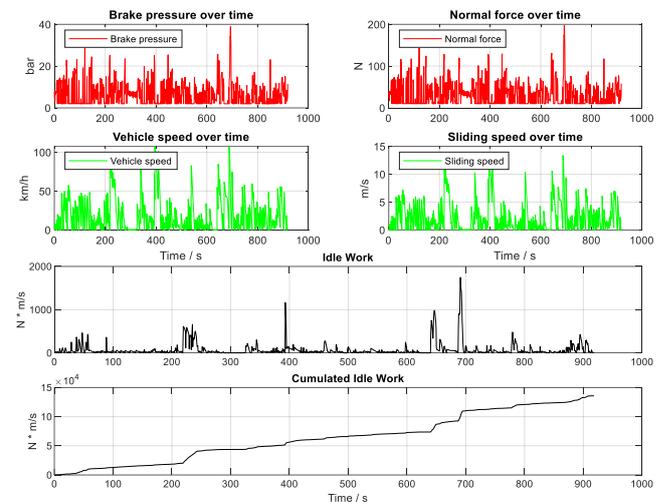


Figure 7 Driving Profile with relevant brake applications for the AUT

Figure 8 shows the velocity (left) and normal force curve (right) for a specific brake application of the created driving profile performed at the AUT. In red the real measurement on the tribometer is shown. The visual comparison already reflects the very good agreement of the actual values with the target values (blue).

The comparison with the actually realized idle work at the tribometer showed deviations of maximum 1,4 %. The driving profile measuring function of the AUT is therefore functional and satisfies the metrological requirements.

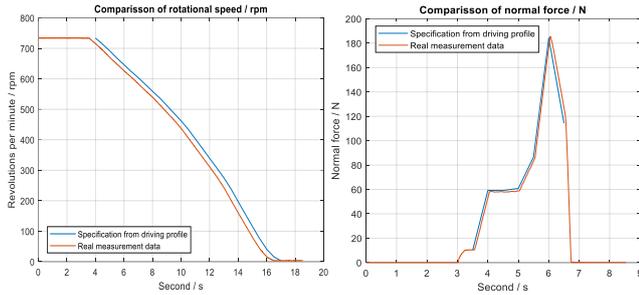


Figure 8 Comparison of rotational speed and normal force between Driving Profile Data and Measurement Data

With regard to the reproducibility of the results, initial test measurements already showed good agreement of the wear results. Figure 9 shows three successive driving profile measurements in which the wear coefficients were $k = -2,7462 \cdot 10^{-8}$ mm/Nm (black), $k = -2,9543 \cdot 10^{-8}$ mm/Nm (blue) and $k = -2,8904 \cdot 10^{-8}$ mm/Nm (yellow). The deviation from the average value is therefore less than 4,1 %.

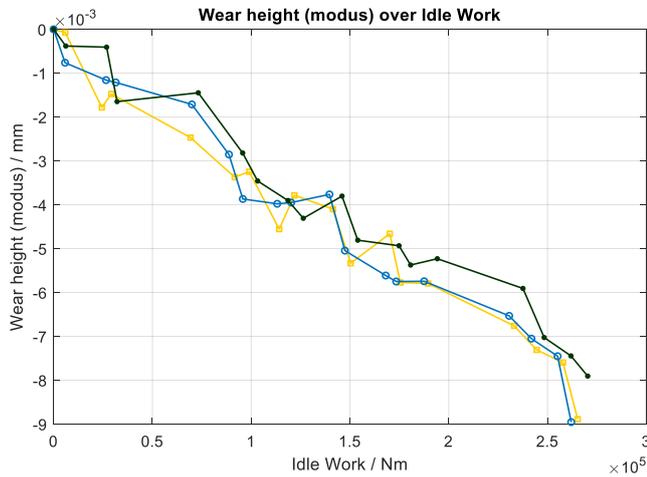


Figure 9 Test results using Driving Profile Measurement feature show a good reproducibility of the results

The results presented in the following chapter are based on multiple driving profiles which, with 1.515 applications, provide a total idle work of 632.000 Nm on the tribometer. This corresponds to a vehicle mileage of 1.470 km and a braking distance of 45 km.

4.2. Exemplary results

Figure 10 shows the result of the modal wear height with the above described driving profile measurement for the friction pairing H. For the total of 1.515 applications carried out, 19 pad topographies were recorded.

It is obvious that the measurement results fluctuate close to the determined wear gradient. Increases and decreases in wear height alternate accordingly, while the trend can be described very well with Archard's wear coefficient. Consequently, the wear coefficient in this example does not allow conclusions to be drawn about the short-term change in pad height, but rather enables a statement to be made about the mid-to-long-term wear evolution when measured on the tribometer.

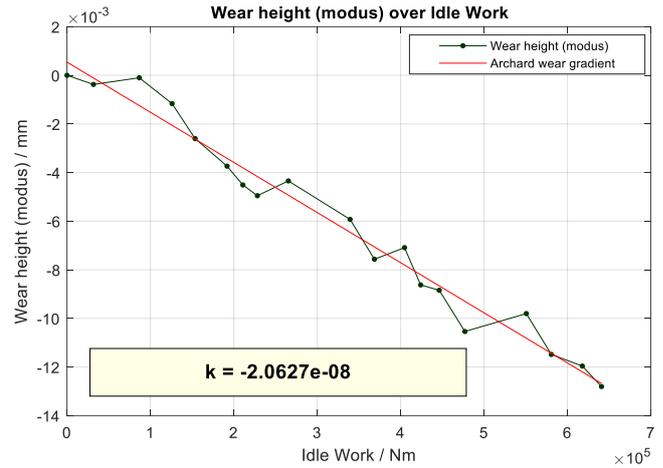


Figure 10 Exemplary wear results of a Driving Profile Measurement with Friction Pairing H

In contrast to the stationary measurement, the coefficients of friction are quite constant at approximately $\mu = 0,375$, see Figure 11.

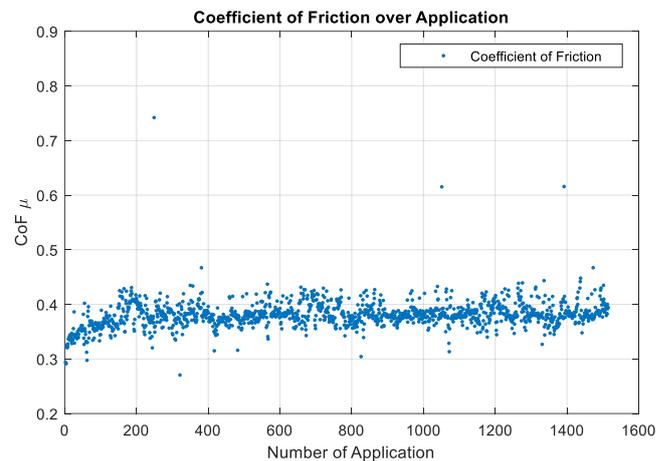


Figure 11 Exemplary friction results of a Driving Profile Measurement

Table 1 summarizes the wear results obtained by the driving profile measurements (3rd column). It is worth noting that both the minimum (Friction Pairing C) and the maximum (Friction Pairing A) wear gradient were recorded with the driving profile measurements.

5. Comparison of the wear results

Several findings can be derived from the comparison between the wear gradients of the stationary procedure and the driving profile measurement. Table 1 compares the results of the different friction pairings. In addition, the deviation of the respective values with respect to the mean value of the stationary and transient wear gradients according to Archard is given. In five out of eight cases this deviation is less than 5,0 %. Only Friction Pairing B shows a significant deviation of 16,2 %.

Table 1 Comparison of the wear results

Test pairing	Archards wear coefficient k determined by stationary test procedure [mm/Nm]	Archards wear coefficient k determined by driving profile measurement [mm/Nm]	Deviation from the average wear coefficient
Friction Pairing A	$-2,5711 \cdot 10^{-8}$	$-2,8213 \cdot 10^{-8}$	4,6 %
Friction Pairing B	$-2,1355 \cdot 10^{-8}$	$-2,41 \cdot 10^{-8}$	6,0 %
Friction Pairing C	$-1,6255 \cdot 10^{-8}$	$-1,1726 \cdot 10^{-8}$	16,2 %
Friction Pairing D	$-2,5437 \cdot 10^{-8}$	$-2,5782 \cdot 10^{-8}$	0,7 %
Friction Pairing E	$-1,5751 \cdot 10^{-8}$	$-1,9266 \cdot 10^{-8}$	10,0 %
Friction Pairing F	$-2,1203 \cdot 10^{-8}$	$-1,9948 \cdot 10^{-8}$	3,1 %
Friction Pairing G	$-1,733 \cdot 10^{-8}$	$-1,6776 \cdot 10^{-8}$	1,6 %
Friction Pairing H	$-2,186 \cdot 10^{-8}$	$-2,0627 \cdot 10^{-8}$	2,9 %

Accordingly, friction pairings seem to exist whose wear behaviour reacts very sensitively to the simultaneous variation of normal force and velocity, while other friction pairings remain unaffected (see e.g. Friction Pairing D). The temperature level of the pad and disc is always below 40°C, both in the stationary procedure and in the driving profile measurements. A temperature effect can therefore be excluded as far as possible. There could be a connection to the theory of Surface Reservoir Dynamics developed by G.-P. Ostermeyer [2], since the samples with a higher deviation also show more significant differences in the presence of dust on the pad surface during a measurement. Further reasons for this sensitivity could be the different material parameters of the friction materials. N. Viswanath and D. G. Bellow mentioned, for example, the specific heat capacity, the thermal conductivity or the modulus of elasticity as important parameters influencing the wear of polymers [19]. However, all these parameters are highly uncertain when determined for brake pads.

6. Conclusion and Outlook

Although recuperation challenges conventional vehicle brakes with modified, smoother load conditions, it does not free the brake from its key function as a safety-relevant component in critical driving situations. Nevertheless, there is potential for optimization, for example in the dimensioning of brake pads. In cooperation with AUDI AG, this paper could demonstrate that wear measurements on the AUT of the IDS generate reproducible results both with a

specially developed stationary wear procedure and with real driving profile data.

Seven of the eight friction pairings measured in total showed deviations from the mean value of $\leq 10\%$. Accordingly, one of the seven friction pairings still raises questions that will be examined in more detail elsewhere. Nevertheless, the measurements on the tribometer give a very reliable statement about the wear behavior of friction pairings, as far as temperature effects can be neglected.

The predictions based on the tribometer results already indicate a significantly increased service life of brake pads in BEV compared to brake pads in ICEV. However, in a further step the results must be verified by measurements on dynamometers and real vehicles. Based on the measurements, transfer functions are to be determined in a further step, which make the results obtained on the tribometer applicable in practice and thus enable an early brake pad dimensioning in BEV (see Figure 12).

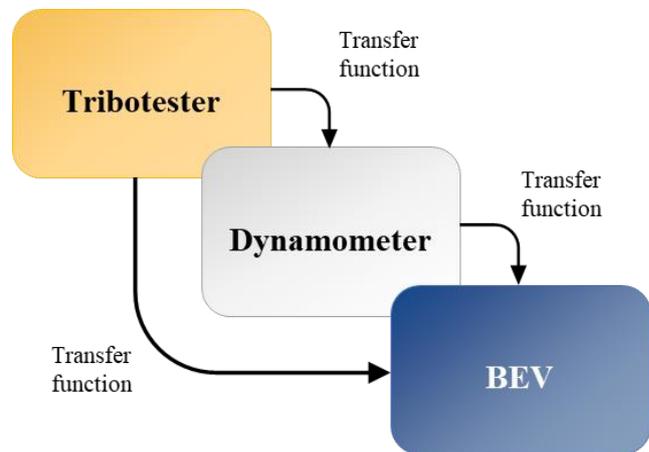


Figure 12 Determination of transfer functions between Tribotester, Dynamometer and Battery Electric Vehicle

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