

# On Implementing a Simulated Moment of Inertia in a Reduced Scale Brake Dynamometer

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**Abstract:** This paper describes the implementation of brake procedures with simulated inertia on a reduced scale brake dynamometer. For this purpose, the theoretical foundations are first explained, and test measurements are carried out. The results show good agreement between calculated and measured accelerations during brake applications. Furthermore, braking processes with various driver models were carried out using associated normal force functions, and examined with regard to their coefficients of friction. A clear difference between these models can be observed. This introduces the possibility for new measurement methods for the analysis of NVH-behavior in brake systems. This work is the first step towards identifying a transfer function between full- and reduced-scale brake dynamometers. In a second step, the temperature scaling must be taken into account. Based on these results, it is possible to isolate the influences of the braking system components from those of the dynamics of the friction coefficient in the boundary layer.

**Key words:** Brake, simulated inertia, friction, tribometer.

## Nomenclature

### Acronyms

AUT	Automated universal tribotester
CoF	Coefficient of friction
NVH	Noise vibration harshness

### Greek Symbols

AUT	Angular acceleration rad	$\text{rad} \cdot \text{s}^{-2}$
CoF	Moment of inertia	$\text{kg} \cdot \text{m}^2$

### Roman Symbols

A	Brake pad area	$\text{mm}^2$
F	Force	N
M	Torque	$\text{N} \cdot \text{m}$
R	Radius	mm

### Subscripts

AUT	Scaled parameters for measurements on the AUT
Fric	In frictional direction
Motor	AUT rotation drive
Real	Real characteristics of the AUT
Vehicle	Parameter of the simulated vehicle

## 1. Introduction

Investigations on high-load tribological contacts,

for example in automotive brake systems, present a formidable challenge for researchers. Many tribological phenomena are known, but there are still some effects, such as squeal, which are not fully understood [1]. To enable experimental research in this field, two types of tribometers are commonly implemented: full-scale and reduced-scale brake dynamometers. Each is specialized for specific investigations. Full-scale dynamometers are used to test the whole brake system, including, for example, the caliper and wheel suspension. A rotating body simulates a portion of the vehicle's mass. The brake system reduces the rotation speed to an equivalent target velocity. The measured braking torque is used to calculate the coefficient of friction. The amount of kinetic energy which is transformed to heat energy is equal to that of the real brake system. Such devices can therefore be used for performance tests [2].

To understand the tribological phenomena in the boundary layer, reduced-scale brake dynamometers are used. Using reduced-sized brake pad samples, these devices are capable of achieving surface pressures and sliding velocities equivalent to those implemented using full-scale dynamometers. This

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allows for investigations with higher precision, but reduces the heat generation in the system. The coefficient of friction is measured with a 3D force sensor directly behind the brake pad sample, close to the contact, reducing the influences of the brake system [3].

These two measurement methods can be understood as investigating two different systems: the reduced-scale tester offers insight into the tribological contact, the full-scale tester into the brake system [4, 5].

The transfer functions from the real brake system to the full- and reduced-scale brake dynamometers are known (Fig. 1a). A transfer function between the two dynamometers must also exist, but it is yet unknown. Once identified, this function could be used to transform the results from one test device to the other, or to directly compare results from different devices. Currently, results from full-scale brake dynamometers are useful for interpretation of the brake system’s behavior. Results from reduced-scale brake dynamometers are suited for interpretation of fundamental friction phenomena (Fig. 1b). The aim of the present work is to combine the benefits of both measuring methods, broadening the scope of interpretation for both devices. To this goal, the transfer function between the devices is indispensable. To attain this function, it is necessary to identify and isolate the contributions from the tribology and from the brake system towards the end results. In addition, the effects of the dynamics of brake components, such as the caliper, must be investigated.

In order to identify the influences of the full-scale dynamometer on the coefficient of friction, the wear- and NVH behavior, it is necessary to implement brake applications with a simulated moment of inertia and realistic temperature behavior on a reduced-scale brake dynamometer. This paper presents the implementation of the simulated moment of inertia in a reduced-scale brake dynamometer.

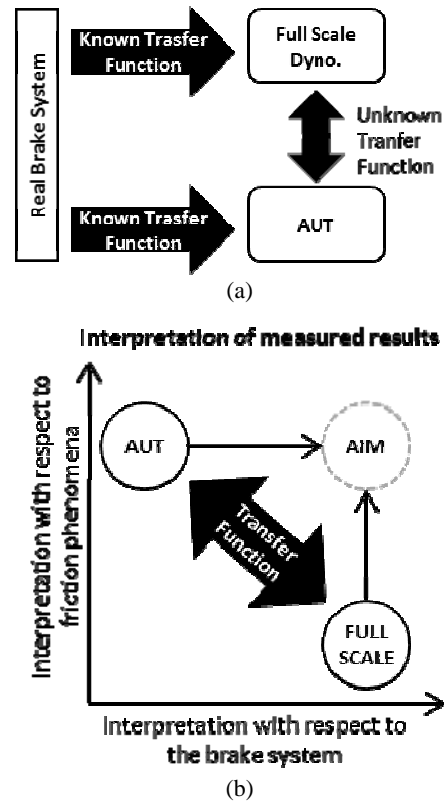


Fig. 1 Transfer function between full- and reduced-scale dynamometer.

## 2. Theory

The Institute of Dynamics and Vibrations has three reduced-scale brake dynamometers, optimized for various load requirements. For the following investigations, the AUT (automated universal tribotester) is used. The AUT is a highly-automated pin on disk test device. The maximum normal load is 500 N, the rotation speed can reach 1,500 rpm. A heating chamber can be used to heat the brake disk up to 200 °C. An optical measurement station is implemented for investigations of the brake pad’s surface. It can take digital photographs of the surface with ring and line lights and measure the topography with a resolution of about 1 μm. Further details on the components of the AUT, the measurement of pad and disk topographies, and the control software can be found in Refs. [6-8].

On such reduced-scale test devices, In-Stop brake applications are typically performed using uniform

speed ramps rather than a moment of inertia. Speed ramps on the AUT require a start velocity, an end velocity, and a set-point for the acceleration. For tests with a simulated moment of inertia, it is only necessary to adapt the acceleration at each time to step based on the current coefficient of friction. In Ref. [7], it shows how the friction force can be used on the AUT as a feedback parameter for realizing brake applications with constant friction power. Here we use a scaled friction force to calculate the acceleration for a simulated moment of inertia.

Based on the requirement that the sliding speed during a brake application on the AUT should replicate that on a full-scale dynamometer, the acceleration can be calculated as follows:

$$\ddot{\phi}_{AUT} = \ddot{\phi}_{Vehicle} \frac{r_{Fric,Vehicle}}{r_{Fric,AUT}} \quad (1)$$

Assuming that parasitic effects such as damping can be neglected, the acceleration for each test stand depending on the friction force, the friction radius and the moment of inertia can be computed as:

$$\begin{aligned} \ddot{\phi}_{AUT} &= \frac{F_{Fric,AUT} \cdot r_{Fric,AUT}}{\Theta_{Simulated}} \\ &= \frac{F_{Fric,Vehicle} \cdot r_{Fric,Vehicle}}{\Theta_{Vehicle}} \cdot \frac{r_{Fric,Vehicle}}{r_{Fric,AUT}} \end{aligned} \quad (2)$$

By expanding and rearranging this equation, the simulated moment of inertia is:

$$\Theta_{Simulated} = \frac{A_{AUT} \cdot r_{Fric,AUT}^2}{A_{Vehicle} \cdot r_{Fric,Vehicle}^2} \cdot \frac{\Theta_{Vehicle}}{2} \quad (3)$$

and can be used to calculate the acceleration for the AUT (see Eq. (2)). Additional assumptions are that the coefficients of friction in both systems are identical and the surface pressures are equal.

A rotation drive is used to generate the necessary torque for simulating the deceleration of a given moment of inertia. Lower simulated moments of inertia result in deceleration processes of shorter duration. In the extreme case, a zero-valued moment of inertia would need to decelerate instantaneously,

which is impossible to realize. The capability of the rotation drive determines the actual lower limit of the simulated moment of inertia, which is also influenced by the size of the brake pad, the friction radius of the simulated brake system, and the normal load. In this case, the torque of the engine is limited to 65 Nm (Eq. (4)).

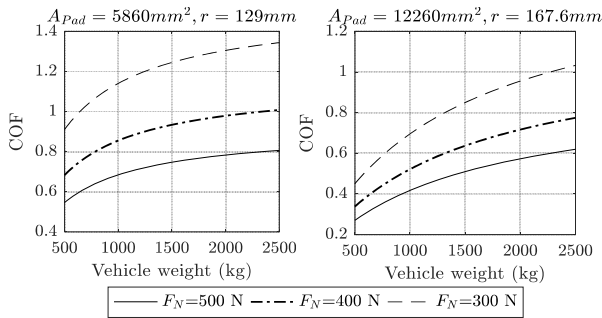
$$\begin{aligned} M_{Motor} &= M_{Fric} + \Theta_{real} \cdot \ddot{\phi} \\ &= M_{Fric} \left( \frac{\Theta_{real}}{\Theta_{AUT}} + 1 \right) \leq 65 \text{ Nm} \end{aligned} \quad (4)$$

Fig. 2 shows the theoretical maximum coefficient of friction in two brake systems. Smaller simulated brake systems require the simulation of lighter vehicle weight. When simulating the moment of inertia, a low vehicle weight and a large brake system can only be realized with low normal loads.

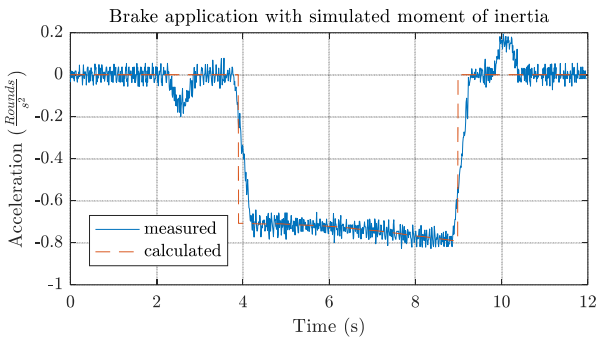
### 3. Results and Discussion

As a benchmark test for measurements with simulated moment of inertia, 270 brake applications with varying normal load, start velocities and target velocities were performed. An exemplary result for the acceleration is shown in Fig. 3. The blue line represents the measured acceleration, the red line represents the target acceleration. The discrepancies around  $t = 3.5$  s and  $t = 10$  s result respectively from the application and removal of the normal load, the actual simulation occurs between these two events. The AUT requires about 300 ms to reach the target acceleration. Once the target acceleration is reached, the deviation is low. It increases once again at the end of the brake process, when the acceleration is set to 0 m/s<sup>2</sup>. The discrepancies at the beginning and end of the brake application are correlated with the magnitude of the target acceleration and the friction force (Eq. (2)).

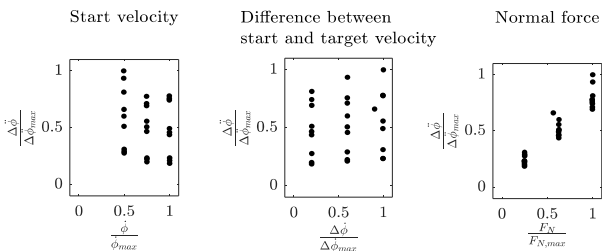
The results of the benchmark test in Fig. 4 show the normalized deviation between target and measured moments of inertia with respecting to the normal load, the start velocity and the difference between the start and target velocities. The AUT shows a strong sensitivity



**Fig. 2** Example for limitation of the simulated inertia on the AUT for two brake systems.



**Fig. 3** Target and actual acceleration during a brake application with simulated moment of inertia.



**Fig. 4** Results of the benchmark test.

to the normal load. Lower normal loads produce the lower deviation. The difference between target and start velocities shows the same tendency. The start velocity itself has the reverse effect. Increasing the velocity tends to decrease the deviation.

The implementation of this functionality opens up a wide range of possibilities for investigations of various friction phenomena. The transfer function between full- and reduced-scale dynamometers can be identified. Furthermore, it is possible to implement an additional controller in the AUT, for example, a virtual driver that interacts with the brake system. Studies can be conducted to investigate the response of the coefficient of friction and its history to a driver

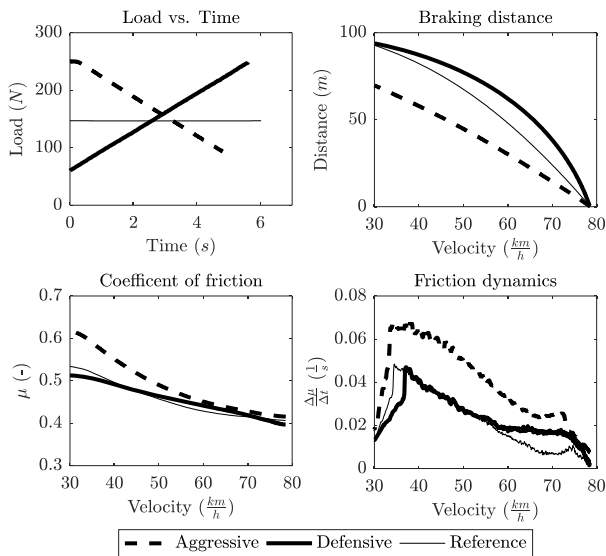
who brakes aggressively or defensively. Such a study has been carried out on the AUT, implementing a simple driver model using load ramps.

The following results show the coefficient of friction during a brake procedure from 80 to 30 km/h. The defensive driver increases the normal load from 180 N to 240 N in 3 seconds, the aggressive driver from 280 N to about 230 N. Because the coefficient of friction depends on several parameters (e.g. sliding velocity, normal force, temperature), the friction dynamics vary in these applications. Fig. 5 shows the results of two driver models. For each driver model, 150 brake applications were performed. Additionally, a reference brake with constant normal load was performed. It includes the average measurement data of the last 100 brake applications of each driver model. As expected, the braking duration of the aggressive driver is shorter than that of the defensive driver, the same applies for the braking distance. The coefficient of friction increases with decreasing velocity. The defensive driver model has friction dynamics that are very similar to the reference driver. The CoF gradients for the aggressive driver are higher.

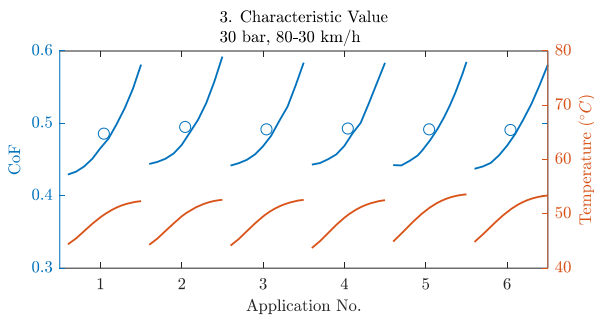
The results of the experiment show the influence of a simple driver model on the friction dynamics. It is known that the friction dynamics affect the squeal behavior of a brake system. Different driver models should therefore be expected to produce different squeal behaviors. The challenge of such measurements is identifying a brake disk and pad combination capable of exhibiting various degrees of squealing behavior [9].

#### 4. Conclusions and Outlook

This work focused on performing brake procedures on a reduced-scale brake dynamometer with a simulated moment of inertia. Based on the test device software, speeded ramps with varying accelerations are used. The acceleration is calculated using a scaled moment of inertia and the instantaneous measured friction force. This method has limitations based on the



**Fig. 5** Effect of different driver models on the friction dynamics.



**Fig. 6** SAE-J2522 test procedure: example test results with simulated moment of inertia on the AUT.

drive's torque, along with the normal load and the dimensions of the brake system. The simulation of vehicles with a weight greater than 1,500 kg and a braking force less than 400 N should be possible. This also depends, however, on the coefficient of friction.

The second part of this work involves measurements to verify the implemented algorithm. A benchmark test on the AUT shows that the controller works in a wide parameter range. The deviation between target and actual simulated moment of inertia seems to be high at low load and high speed. The difference between start and target velocities only has a minor influence and can be neglected. Along with the common In-Stop brake applications, as performed in the SAE-J2522, the AUT offers the possibility to implement various driver models. Here, an example

with two types of drivers was presented. The results show a variation in the dynamics of friction for the two drivers. These dynamics have a strong influence on NVH behavior. For further experiments, a brake system will be used which has a high occurrence of squeal.

This paper presented the first steps towards identifying a transfer function between full- and reduced-scale dynamometers. In-Stop brakes with a simulated moment of inertia were also performed on the AUT. Furthermore, the SAE-J2522 procedure has been carried out on the AUT, test results are shown in Fig. 6.

Because the temperature has a strong influence on the friction dynamics, it is necessary to consider and account for thermal effects. The reduced-scale generates less heat than the real brake system due to the relatively low friction power. Therefore, to create a thermally comparable system, additional heat energy must be introduced to the system. One solution is an external heating element. Currently, the AUT can heat the brake disk to a stationary temperature before each brake application. For a proper scaling between the real brake system and the reduced-scale brake system, both the absolute value and the dynamics of the temperature must be implemented. In fact, the effect of the friction power in both systems has to be the same. The implementation of such scaling properties will be carried out in the coming development steps of the AUT.

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