Measuring brake pad friction behavior using the TR3 test bench

Citation for published version (APA):

Document status and date:
Published: 01/01/2006

Document Version:
Publisher’s PDF, also known as Version of Record (includes final page, issue and volume numbers)

Please check the document version of this publication:
• A submitted manuscript is the version of the article upon submission and before peer-review. There can be important differences between the submitted version and the official published version of record. People interested in the research are advised to contact the author for the final version of the publication, or visit the DOI to the publisher’s website.
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Measuring brake pad friction behavior
using the TR3 test bench

DCT no. 2006.118

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Eindhoven, September, 2006
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Abstract

A computer-controlled test rig to perform tests on the friction behavior of different materials is developed. On this test rig brake pads in combination with an aluminum brake disk are tested. A controller to automate this measuring procedure is made. The brake pads are tested at various operating conditions with different disk surface temperature, brake pressure and vehicle speed. The tests show that the brake pads have an optimal operating temperature which lies around 220 °C. The resulting braking torque depends linearly on the brake pressure. Two sets of brake pads are tested. One set came standard with the brake caliper and the other set is produced by Galfer. It is found that the Galfer set in general delivers 20 to 30 Nm more braking torque than the standard set of brake pads.
Introduction

The Formula Student Racing Team Eindhoven is building a racing car which uses special aluminum brake disks. These disks are developed by the company Ceradure who donates these disks as a part of a sponsoring agreement. Since the disks are a new development, Ceradure is still trying to collect information on the best material for brake pads. Another part of the sponsor agreement states that the Formula Student Racing Team Eindhoven would research the friction behavior on different kinds of brake pads, delivered by Ceradure. In this project two sets of brake pads are compared to each other on a test rig developed at the TU/e.

These two sets are:

- the brake pads that came standard with the brake caliper that is used on the Formula Student car (which will from now be referred to as the “standard pads”)
- the Galfer FD107 G1003 brake pads (which will be referred to as the “Galfer pads”).

This paper will first discuss the development of the test rig used to perform the comparing friction tests in chapter 1. These tests are prescribed by a test program. In this test program operating conditions with varying brake pressure, surface temperature and vehicle speed are stated. This test program will be discussed in chapter 2. Finally the test results will be analyzed in chapter 3, after which some conclusions can be drawn regarding the friction behavior of the different sets of brake pads.
1. The test rig

1.1 Traditional brake testing

The first part of the project was to find a way to execute the desired comparison tests between the different materials of brake pads. For this, a test rig was needed. From Ahmed, Leung and Datta (2000) [1] and Pompon (1998) [2] is learned that a traditional way of testing disk brakes is to use an electromotor to speed the disk up to a desired amount of revolutions per minute, while the disk is on the same axle with a number of flywheels. These flywheels should have the same moment of inertia as the vehicle that would normally carry the tested brake disks. After speeding up the disk the electromotor would be shut off and the brake would be applied. At this point the torque on the axle is measured by a sensor. From this measured braking torque, together with the braking pressure applied to the hydraulically circuit and the dimensions of the brake disk and pads, the friction coefficient between the disk and the pads can be calculated. However, at the TU/e such heavy flywheels were not available for this project, nor was a strong electromotor.

1.2 The TR3 test rig

Instead of using the traditional way as described in above, the decision was made to use the TR3 test rig which is located in the engine cells of the TU/e. The TR3 is a computer-controlled (Dspace) test rig with a Nissan Primera 2.0 Elegance Hypertronic M6 internal combustion engine and a Nissan CK2 CVT. Furthermore the TR3 already had a hydraulic brake circuit with master cylinder and an actuator for this master brake cylinder.

The brake disk could be mounted on the axle coming out of the CVT. By setting the cruise control of the TR3 to a desired speed and then applying the brake the engine would deliver more torque to overcome the resistance caused by the brake and maintain the desired speed. The difference between the measured torque delivered by the engine before braking and the torque measured while the brake is being applied equals the braking torque applied by the disk brake. From there on the friction coefficient can be calculated.

1.3 Sensors and actuators

To control and measure the various parameters in this brake test some additional sensors and actuators were installed on the TR3 test rig. The TR3 already has a torque sensor and a sensor that measures the amount of revolutions per minute on the axle coming out of the CVT. On top of that an infrared pyrometer is installed to measure the surface temperature of the brake disk as the test program would contain measurements at different surface temperatures. Furthermore a pressure sensor is installed to measure the applied braking pressure. This sensor is installed as far away from the caliper as possible to avoid that the brake fluid near the sensor would become to hot. In this way a time delay in the measured pressure and the actual...
pressure applied to the brakes exists though, but this is not relevant because according to the test program the measurements are done only at constant pressure. The brake cylinder was applied by a Linak actuator and a current sensor was installed here for protection. This way a check can be made on the amount of current that is actually going to the actuator. Also a tube which blows pressurized air is directed at the brake disk. This was necessary to keep the surface temperature of the disk under control, as there was no driving wind to cool the disk, unlike when it is mounted on a real car. Without the cooling the brake fluid would become too hot. This would cause expansion of the fluid, so there would always remain some pressure in the brake line, even though the brake cylinder was not applied. The pressurized air also helps the disk to cool down faster between the tests. More detailed information on the sensors and actuators can be found in appendix B.

1.4 The brake pressure controller

A brake pressure controller had to be designed in order to be able to apply the desired amount of pressure with use of the actuator. This was done by putting noise on the Linak actuator and then measure the pressure sensor’s output. In this way a bode-plot of the sensitivity function was created (figure 1.2). The sensitivity function is the transfer function between the input of the system (as shown in figure 1.1) and the error. In this case the input is the voltage sent to the Linak actuator. The error is the difference between the measured pressure and the reference pressure calculated by the test loop model (described in paragraph 2.5).

![Figure 1.1 schematic representation of the controlled system](image_url)

The absolute value of the sensitivity function has to remain below 6 dB. Using the sensitivity function bode-plot, a controller was designed which ultimately consisted of a gain (1.1), a lead-filter (1.2), a lag-filter (1.3) and a low-pass filter (1.4). The transfer-functions of the individual filters and the total controller are as follows:

\[
\text{Gain} = 0.015
\]

\[
\text{Leadfilter} = \frac{1}{2 \cdot \pi \cdot s + 1}
\]

\[
\text{Lagfilter} = \frac{1}{0.1 \cdot \pi \cdot s + 1}
\]
Figures 1.2 and 1.3 on the next page show the bode-plots of the sensitivity functions without and respectively with the controller. It can be seen that the absolute part of the sensitivity function remains under the 6 dB after applying the controller.

\[
Lowpassfilter = \frac{1}{(40 \cdot \pi)^2 \cdot s^2 + \frac{2 \cdot 0.7}{40 \cdot \pi} \cdot s + 1}
\]  
(1.3)

\[
Controller = \frac{0.007599 \cdot s^2 + 0.09549 \cdot s + 0.3}{3.208 \cdot 10^{-6} \cdot s^4 + 0.000767 \cdot s^3 + 0.08636 \cdot s^2 + 3.21 \cdot s + 1}
\]  
(1.4)

(1.5)
Figure 1.2 Bode-plot of the measured sensitivity function without controller

Figure 1.3 Bode-plot of the sensitivity function with controller
1.5 Visual of the test rig

Figure 1.3 shows a schematic representation of the test rig.

![Schematic Representation of the Test Rig](image)

**Figure 1.3 schematic representation of the test rig**

1: Nissan combustion engine and CK2 CVT.
2: Axle coming out of the CVT with torque- and revolutions sensors.
3: Brake caliper
4: Brake disk with pyrometer and pressurized air tube.
5: Brake line with pressure sensor.
6: Brake cylinder with actuator.
7: Axle connected to the CVT-axle, This axle goes to a differential, a flywheel and an eddy current brake

Some photos of the test rig can be found in appendix C.
2. The testing program

2.1 Operating conditions

The goal of this project is to compare brake pads of different materials under various circumstances. For this a testing program with the desired measuring points is needed. Following Ahmed [1] the following operating conditions are chosen:

- Load (Brake pressure from 10 to 40 bar)
- Speed (rotational speed of the disk, directly related to the car speed, 60 and 120 km/h)
- Temperature (surface temperature of the disk, from 95 to 275 °C)

2.2 Brake pressure

The brake pressure is based on the research of Ahmed [1] and almost reaches the theoretically needed brake pressure to produce the desired deceleration $a = 1.2 \, \text{g}$ of the Formula Student car. With assumptions made for the mass of the car, $m = 300 \, \text{kg}$ and the friction coefficient between the materials ($\mu = 0.5$, also based on the research of Ahmed [1]), the braking torque needed for such a deceleration at 60 km/h ($v = 16.7 \, \text{m/s}$) follows from (2.3):

$$v \, a = m \cdot a \cdot v = T \cdot \omega$$  \hspace{1cm} (2.1)

$v$ and $\omega$ are related to each other by

$$\frac{v}{\omega} = r$$  \hspace{1cm} (2.2)

Where $r = 0.28 \, \text{[m]}$ is the radius of a front wheel of the formula student car, so:

$$T = m \cdot a \cdot r = 300 \cdot 9.81 \cdot 1.2 \cdot 0.28 = 989 \, \text{[Nm]}$$  \hspace{1cm} (2.3)

With the total required braking torque of 989 [Nm] and 3 disk brakes on the car, depending on the brake balance, each brake has to deliver approximately 329.7 [Nm] of braking torque. With an average radius of the brake pad’s rubbing path $R_{av} = 0.0925 \, \text{[m]}$ and the surface area of the brake pads $A_{pad} = 17 \, \text{e}^{-4} \, \text{[m^2]}$ the delivered braking moment at $p = 40 \, \text{[bar]}$ brake pressure can be calculated as followed, according to Ahmed [1]:

$$\mu = \frac{F_{friction}}{F_{normal}}$$  \hspace{1cm} (2.4)

---

1 According to the information on http://www.formulastudent.tue.nl
2.3 Speed

The rotational speed of the disk is, as said before, directly related to the car speed. Since the cruise control on the TR3 allows us only to state a desired speed in km/h the choice is made to measure at 60 and 120 km/h, two speeds which are very reasonable for the Formula Student car. However, during testing the brake disks were damaged at severe operating conditions; the hard top-layer of the aluminum disk had come off at some points (figure 2.1). Due to this damage the 120 km/h measurements turned out to be unreliable.

![Figure 3.3 Damage on the brake disk](image)

2.4 Temperature

The range of the disk surface-temperature at which the measurements would take place was limited by the range of the available pyrometer, since it has an upper limit of 320 °C. However a discussion with Mr. Coumans, the director of Ceradure, the company which produces the tested brake disks learned that the aluminum brake disks have a very low optimal temperature range compared to, for example steel disks. Going much higher than 300 °C would even damage the disk. So the choice was made to start measuring with a maximum temperature of 275 °C, knowing the temperature of the disk would rise while the brake is being applied. From this maximum temperature 3 lower temperatures were chosen. So measurements are executed at

\[
F_{\text{friction}} = \frac{M_{\text{brake}}}{R_{av}} \tag{2.5}
\]

\[
F_{\text{normal}} = p \cdot A_{\text{pad}} \tag{2.6}
\]

\[
M_{\text{brake}} = R_{av} \cdot \mu \cdot p \cdot A_{\text{pad}} = 0.0925 \cdot 0.5 \cdot 40 \cdot 10^3 \cdot 17 \cdot 10^{-4} = 314.5 \ [Nm] \tag{2.7}
\]
95 °C, 155 °C, 215°C and 275 °C. Going much lower then 95 °C would not be reasonable as the surface temperature of the disks would never get so low when the car would actually be driving.

2.5 The test loop model

In order to do automate the test procedure a test loop is created which automatically repeats the measuring process at a given operating condition. A simulink-model is made to calculate the reference value of the pressure that should be delivered during the measuring process. The input of the model is the measured brake disk surface-temperature. The disk surface should be warmed up to 20 °C above the desired measuring temperature. If the measured temperature is 15 °C below the desired measuring temperature, which is checked in part 1 of figure 2.2, the output of the model would be a warm-up pressure which can be entered in Dspace (part 2 in figure 2.2). This warm-up pressure is in general 5 bar and makes sure that the disk warms up while it is spinning at the desired speed.

When the measured temperature has reached the warm up temperature, the model puts out a reference pressure of 0 and the actuator stops braking. While the disk keeps spinning the surface temperature starts to drop. As soon as the desired measuring temperature is reached the model sends out a reference value for the desired measuring pressure for a certain time. Both the desired measuring pressure and the braking time can be altered in Dspace. When the braking time has passed the output value of the model becomes 0 again. The actuator stops braking and the while the disk keeps spinning the disk surface will cool down. When the measured disk surface-temperature reaches the desired measuring temperature the model will again put out the desired reference pressure for the desired braking time. This loop is used to get 8 measurements of each possible operating condition in the test program. From those 8 measurements the average braking torque is calculated. The loop is controlled in part 3 in figure 2.2.

The idea was to apply the brake until the cruise control made sure the disk was rotating at constant speed, so there will be an equilibrium between the braking torque and the torque delivered by the engine. This would take to much time however and the disk would run to hot. Therefore the choice is made to apply the brake for only 3 seconds and look at the highest braking torque measured during those 3 seconds. The braking time is set back to 2 seconds at operating conditions of 275 °C surface temperature and 40 bar brake pressure to avoid the disk getting to hot. In order to be able to compare the measured braking torque at these operating conditions with other the braking torque at other operating conditions the measurements at 275 °C surface temperature and 30 bar where done with a braking time of both 2 and 3 seconds. The simulink-model is loaded into Dspace. In this Dspace environment values for the desired measuring temperature, the warm-up pressure, the total braking time and the desired measuring pressure can be entered. The warm-up pressure was occasionally set to 7 or 10 bar when the disk had to warm up to maximum temperature. Furthermore the brake pressure could at any time be controlled manually in Dspace through a switch in the model. The model also contained a safety switch which would set all reference values to 0 when a given critical disk surface temperature was passed. Figure 2.2 shows the simulink-model where the variables that could be altered in Dspace are highlighted.
Figure 2.2 Simulink-model for the test loop
Figure 2.3 shows the subsystem of the simulink-model depicted in figure 2.2

![Simulink Model](image)

**Figure 2.3 Subsystem of the simulink-model for the test loop**

### 2.6 The test program

Table 2.1 shows the test program with all the operating conditions one set of brake pads is subjected to.

<table>
<thead>
<tr>
<th>Starting temperature</th>
<th>P = 10 bar</th>
<th>P = 20 bar</th>
<th>P = 30 bar</th>
<th>P = 40 bar</th>
</tr>
</thead>
<tbody>
<tr>
<td>T = 95 °C</td>
<td>60 km/h, 3 sec</td>
<td>60 km/h, 3 sec</td>
<td>60 km/h, 3 sec</td>
<td>60 km/h, 3 sec</td>
</tr>
<tr>
<td>T = 155 °C</td>
<td>60 km/h, 3 sec</td>
<td>60 km/h, 3 sec</td>
<td>60 km/h, 3 sec</td>
<td>60 km/h, 3 sec</td>
</tr>
<tr>
<td>T = 215 °C</td>
<td>60 km/h, 3 sec</td>
<td>60 km/h, 3 sec</td>
<td>60 km/h, 3 sec</td>
<td>60 km/h, 3 sec</td>
</tr>
<tr>
<td>T = 275 °C</td>
<td>60 km/h, 3 sec</td>
<td>60 km/h, 3 sec</td>
<td>60 km/h, 2 and 3 sec</td>
<td>60 km/h, 2 sec</td>
</tr>
</tbody>
</table>

*Table 2.1 Test scheme for a set of brake pads*
3. Results and analysis

3.1 Example of one operating condition measurement

To get an idea of what is exactly happening during a measurement, the measurement at one certain operating condition is discussed here. This measurement is done on the set of Galfer pads with a measurement temperature of 155 °C, a brake pressure of 20 bar and a vehicle speed of 60 km/h. Figure 3.1 shows the data that is collected from this measurement. The data is plotted after using a Butter-filter to suppress the noise on the signal.

In the first few seconds the test loop model is not switched on yet. The measured torque and disk rotational speed can be seen oscillating around a constant value. This is due to the fact that the cruise control is not a very powerful controller in this case. This constant torque value is the torque that is needed to keep the brake disk at the desired rotational speed. This value has to be subtracted from the measured torque signal to get the value for the engine torque.

After approximately 10 seconds the model is switched on. At that moment the disk surface temperature is about 75 °C, well below the desired measurement temperature. The model sends out a reference value of 5 bar, the warm-up pressure. It can be seen that the measured pressure follows the reference value very good. The temperature starts to increase when the brake is applied. The disk rotational speed drops at first, but the cruise control makes the engine deliver extra torque so that the rotational speed is set at the desired value again.

As soon as the disk surface temperature reaches the value of 175 °C the brake pressure drops to zero. The temperature then starts to drop until 155 °C is reached. At that point the test loop model sends out a reference value of 20 bar for 3 seconds. During these 3 seconds the measured torque rises and the disk rotational speed drops. When after 3 seconds the brake pressure is brought back to 0 again the temperature starts to drop and the rotational speed goes back up again. When the temperature reaches 155 °C the model sends out a reference value of 20 bar for 3 seconds again. This is repeated several times so the values of the maximum measured torque can be averaged.

During the whole measurement the effect of the weak cruise control can be seen. When the brake pressure rises or drops, the torque signal reacts a few seconds later. This can be seen very well at the point of 40 seconds, when the brake pressure is already at its maximum level and the torque is only halfway its maximum. The effect of the cruise control can also be seen in the rotational speed signal which has an overshoot with respect to the desired speed when the brake pressure becomes 0 and the cruise control is trying to get the speed at the desired value again.

In figure 3.2, in which the torque is plotted against the brake pressure, it can be seen that the pressure rises first and then the torque starts to rise. Then the pressure falls back to zero after which the torque starts to decrease. With a fast-enough cruise-control the line should be much more linear. The loops in this figure have a length of 3 seconds, except for the warming-up loop.

Finally the data shows that the cruise control can establish a constant torque during the warm-up stage of the measurement. This is because this stage starts at a very low disk surface temperature and the temperature does not rise very quickly because of the low brake pressure. In situations with a higher starting temperature and a higher brake
pressure the disk surface temperature will reach a critical value before a constant rotational speed is acquired. As a result of not being able to measure an average torque while applying the brake under these operating conditions, it was decided to average the maximum values measured in 8 measurement loops. By doing this, the real friction coefficient between the brake pads and the disk cannot be calculated. After all it is unknown what will ultimately be the constant braking torque produced by the brake pads. A comparison between the standard and the Galfer pads can be made however by looking just at the maximum braking torque delivered in the 3 seconds.

Remarkable in figure 3.1 is the fact that the reference pressure decreases gradual after it reached its top. It should, however, drop to 0 bar at once when the 3 seconds of braking time are over. It is not known where this effect comes from and should be researched in the future.
3.2 Braking torque as a function of temperature

From the results of the tests at constant pressure the behavior of the delivered brake torque as a function of the temperature can be seen (figures 3.3 to 3.6). The last figure only has three measurement points because the 2 second measurements made at a brake pressure of 40 bar can not be compared with the rest of the data in these graphics.

It is clear that the braking torque increases until the surface temperature reaches values around 220 °C, after which it starts to decrease. Especially the standard brake pads show this behavior very clearly. The Galfer pads produce less torque in the first three measurements at the 10 bar operating condition. A reason for this could be that these were the first three tests done with the Galfer pads; they might not have been warmed up enough or have not worn enough. Another thing that attracts the attention is the last point in the 30 bar graphic of the Galfer pads, where the brake torque suddenly is much less then expected. This could be the effect of the earlier mentioned damage on the disk, but that is not certain. The fact is that the data collected after this operating condition, the torque of the Galfer pads at constant brake pressure of 40 bar, showed some large variations in the 8 times the torque was measured. For this reason the data on the Galfer pads in figure 3.6 is less reliable. However this data at constant pressure has shown a very clear temperature-dependent behavior at which the brake torque decreases at values higher then ± 220 °C. It also shows that the Galfer pads in general produce a higher braking torque, sometimes more than 30 Nm higher than the standard pads.
Figure 3.3 Braking torque at constant pressure of 10 bar

Figure 3.4 Braking torque at constant pressure of 20 bar
Figure 3.5 Braking torque at constant pressure of 30 bar

Figure 3.6 Braking torque at constant pressure of 40 bar
3.3 **Braking torque as a function of pressure**

By plotting the torque of the operating conditions which had the same temperature at the start of the measurement against the brake pressure, the pressure-dependent behavior of the sets of brake pads can be seen (figure 3.7 to 3.10). In this case the data from the operating points at 275 °C and 40 bar braking pressure can be compared to data collected at 275 °C with 30 bar pressure where measurements have been made with both 2 and 3 seconds braking time. These measurement points are registered in the graphics by a * for the standard pads and a + for the Galfer pads. Here we see an almost linear relation between the brake pressure and the braking torque, which could be expected looking at equation (2.7). This linear behavior between 10 and 40 bar shows a rate of approximately 3 Nm/bar. If this would also be the case below the value of 10 bar, then the braking torque will not be 0 at a brake pressure of 0 bar. So it is likely that the graph has a steeper slope below the value of 10 bar. Again the standard pads show this behavior very clearly, while the Galfer pads deliver less torque than expected at the 10 bar measurements, probably due to the reasons mentioned before. Apart of that it can again be seen that the Galfer pads in general deliver 20 to 30 Nm braking torque more then the standard pads and also show linear behavior. Figure 3.10 shows the deviation in the Galfer pads data where also the 2-second braking time measurements are nearly equal to the standard pads at 30 bar and even lower then those pads at 40 bar. The standard pads also show an increasing line when comparing the 2-second data.

![Graph showing braking torque at constant temperature of 95 degrees Celsius](image)

*Figure 3.7 Braking torque at varying brake pressure with start-temperature of 95 °C*
Figure 3.8 Braking torque at varying brake pressure with start-temperature of 155 °C

Figure 3.9 Braking torque at varying brake pressure with start-temperature of 215 °C
3.4 Friction coefficients during the tests

By rewriting formula (2.7) to

\[ \mu = \frac{M_{\text{brake}}}{R_{av} \cdot p \cdot A_{\text{pad}}} \]  \hspace{1cm} (3.1)

the friction coefficient between the brake pads and the brake disk can be calculated. In figures 3.11 and 3.12 the friction coefficients of the brake pads are plotted for various operating conditions. This is done to give an indication of the values the friction coefficients reached during the measurements. The values reached under the various operating conditions are not the maximum friction coefficients that can be reached however. The figures merely show the maximum friction coefficient reached within 3 seconds. Formula (3.1) shows that \( \mu \) is dependent on \( M_{\text{brake}} \), which has not reached its maximum value after 3 seconds of braking during the tests. We can not determine what the maximum friction coefficients would be under constant torque conditions. It can be seen that \( \mu \) decreases when the brake pressure rises. A comparison between the standard and the Galfer blocks can be made. It can be seen that the friction coefficients reached within 3 seconds are in general higher for the Galfer blocks. In figure 3.12 it can again be seen that the Galfer pads had a lower performance during the first 3 tests, at operating conditions of 10 bar and 95, 155 and 215 °C respectively.
Figure 3.11 Friction coefficients reached during the standard pads measurements

Figure 3.12 Friction coefficients reached during the Galfer pads measurements
4. Conclusion and recommendations

This project has shown that comparative tests on friction behavior of brake pads of different materials in combination with a brake disk can be done at the TU/e. The testing method differs from the traditional way found in literature. There has been developed a computer-controlled test rig with controllers and models to perform such tests. A test program has been made for testing the friction behavior on different operating conditions, with brake pressure and temperature as possible variables.

From the test data is learned something about the temperature-dependent behavior of the friction between the brake disk and the pads. The delivered braking torque increases with the surface temperature of the disk until it has reached around 220 °C, after which the braking torque starts to decrease. So the optimal operating condition regarding the surface temperature lies around this 220 °C, which is much lower then what the case is with for example steel disks.

The brake pressure-dependent behavior is almost linear as could be expected regarding the formulas on braking torque.

From the comparison between the two different sets of brake pads is shown that the Galfer set in general produces 20 to 30 Nm more braking torque, which equals approximately 15 %. The data of the beginning and the end of the measurements done on this seems less reliable however.

Finally the tests revealed that the disk gets damaged when performing tests at operating conditions where there is a combination of high temperature and high brake pressure.

In the future the test program could be extended with a greater range for the used variables, although a bigger temperature range and higher brake pressure would require a pyrometer with a bigger range. Also the torque sensor should be located closer to the brake disk and the pyrometer should measure the disk surface temperature closer to the brake pads. A better solution for the cooling of the disk should be found. The pressurized air is not ideal and could be replaced by a fan for example.

Furthermore it can be tested what the exact effect of the cruise-control is and whether it is useful in these brake tests. Other test programs could be developed which would make the measurements more accurate and would allow a better calculation of the friction coefficient and the actual friction behavior between the materials.

As a next step the model which controls the input brake pressure could be extended to create different test loops, for example a simulation of the brake pressure applied when the car rides on a (test) circuit. Also the fact of the reference pressure decreasing gradual, instead of dropping to 0 at once should be researched.

Finally more different brake disks and brake pads can be tested in the future.
References


# Appendix A: List of symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{\text{brake}}$</td>
<td>Braking power</td>
<td>[W] Watt</td>
</tr>
<tr>
<td>$F_{\text{brake}}$</td>
<td>Braking force</td>
<td>[N] Newton</td>
</tr>
<tr>
<td>$v$</td>
<td>Vehicle velocity</td>
<td>[m/s] meters per second</td>
</tr>
<tr>
<td>$m$</td>
<td>Vehicle mass</td>
<td>[kg] kilogram</td>
</tr>
<tr>
<td>$a$</td>
<td>Deceleration</td>
<td>[m/s$^2$] meters per second squared</td>
</tr>
<tr>
<td>$T_{\text{brake}}$</td>
<td>Braking torque</td>
<td>[Nm] Newtonmeter</td>
</tr>
<tr>
<td>$\omega$</td>
<td>Angular speed</td>
<td>[rad/s] radians per second</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Friction coefficient</td>
<td>[-]</td>
</tr>
<tr>
<td>$F_{\text{friction}}$</td>
<td>Friction force</td>
<td>[N] Newton</td>
</tr>
<tr>
<td>$F_{\text{normal}}$</td>
<td>Normal force</td>
<td>[N] Newton</td>
</tr>
<tr>
<td>$R_{\text{av}}$</td>
<td>Average radius of</td>
<td>[m] meters</td>
</tr>
<tr>
<td></td>
<td>brake pad rubbing path</td>
<td></td>
</tr>
<tr>
<td>$p$</td>
<td>Brake pressure</td>
<td>[Pa] Pascal</td>
</tr>
<tr>
<td>$A_{\text{pad}}$</td>
<td>Brake pad contact surface</td>
<td>[m$^2$] squared meters</td>
</tr>
</tbody>
</table>
## Appendix B: Detailed sensor and actuator information

<table>
<thead>
<tr>
<th>Sensors/Actuators</th>
<th>Type</th>
<th>Serial number</th>
<th>Range [measured variable]</th>
<th>Signal output/input</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Temperature</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>CHINO IR-BT3</td>
<td>BT98YT922</td>
<td>0 tot 300 °C</td>
<td>0 to 20 mA</td>
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<tr>
<td><strong>current sensor</strong></td>
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<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>ELMO VIO25-60</td>
<td>VIO5345588</td>
<td>0 tot 25 A cont.</td>
<td>-3.9 to 3.9 V DC</td>
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<tr>
<td><strong>Actuator brake cylinder</strong></td>
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<td></td>
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<tr>
<td></td>
<td>LA30.1S</td>
<td>30.1SP-50-24DC</td>
<td>-3500 N tot 3500 N</td>
<td>-24 to 24 V DC</td>
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<td><strong>Control LA30.1S</strong></td>
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<tr>
<td></td>
<td>ELMO VIO25-60</td>
<td>VIO5345588</td>
<td>0 tot 25 A cont.</td>
<td>-10 to 10 V DC</td>
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<tr>
<td><strong>Conditioning</strong></td>
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<td></td>
<td>AI141 V1</td>
<td>1002563</td>
<td>500 Hz filtered standard</td>
<td>-</td>
</tr>
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</table>
Appendix C: Detailed test rig pictures

Figure C.1 Nissan combustion engine and CK2 CVT

Figure C.2 Axle coming out of the CVT with torque- and revolutions sensors
Figure C.3 Brake caliper

Figure C.4 Brake disk with pyrometer and pressurized air tube
Figure C.5 Brake line with pressure sensor

Figure C.6 Brake cylinder with actuator