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Testing the Mechatronic Wedge Brake

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ABSTRACT

The eBrake is a novel self-reinforcing electromechanical wedge brake, which operates around the point of maximum self-reinforcement in order to minimise actuation forces. Beyond this point, the system would be unstable without an electronic controller. It is therefore important to demonstrate that this controller is robust to the range of parameter variations likely to be encountered in practice. The first stage of this process can be conducted on a dynamometer under laboratory conditions, to ensure that problems are addressed before proceeding to vehicle tests.

This paper reports testing of the prototype brake on a such a dynamometer. The prototype brake itself is first briefly described, including the main instrumentation used during the tests. This followed by a short section detailing the capabilities of the dynamometer. The main body of the text is devoted to the tests themselves: the rationale behind them; a description of how they were conducted; presentation of the results; and, where relevant, a discussion of how they compare with theory. Of particular interest is the performance of the brake under the influence of thermal effects, and the high dynamics and low power consumption demonstrated. It is shown that the control system and hardware can successfully manage the range of test conditions experienced.

To conclude, a review is presented of the programme status, including the developments underway which will lead to vehicle tests.

INTRODUCTION

In the automotive industry today, there is a strong trend towards ‘power-by-wire’ technologies, aimed at replacing hydraulic or pneumatic systems with equivalent electrically powered ones. While many electrically actuated systems are already in series production, the brake system has always represented a major challenge, both because of the high powers involved and the demands of failure management. In both respects the existing hydraulic systems are good, because the power and energy density are high, and a well-defined failure management strategy already exists. Accordingly, hybrid electro-hydraulic systems have been introduced which meet all existing requirements, and improve the controllability and performance of the brakes. However these systems are relatively complex and expensive, and still rely on hydraulic fluid. Because of the potential benefits of ‘dry’ systems, most manufacturers have also been researching electro-mechanical brakes.

In floating calliper brakes, a large clamping force must be produced between two brake pads to create a frictional torque on the rotating assembly. For electro-mechanical actuators, the clamping force is typically generated by coupling a motor through a gearbox onto a ball- or roller-screw. For the actuator to produce a force, a current is required, resulting in a power drain. A compromise has therefore to be found between the high gear ratio needed to minimise the current for continuous braking and the low gear ratio which minimises the current for dynamics. This requires optimisation of the motor, the gearbox ratio, and the ball-screw lead, subject to the available space and the performance requirements.

It would clearly be beneficial if the steady actuation force could be reduced, since this would make this compromise between static and dynamic performance much easier to reach. The eBrake® solves this problem elegantly by using a wedge to generate the clamping forces. This exploits self-reinforcement of the braking forces by the rotating brake disc to minimise the actuation forces. At the ideal operating point, where the coefficient of friction is equal to the tangent of the wedge angle, the steady-state actuation force required to generate any braking torque is zero. Mathematically, the characteristic brake factor for a floating calliper brake actuated by this method is given by[1]

$$C^* = \frac{\text{Pad Braking Force}}{\text{Brake Actuation Force}} = \frac{2\mu_B}{\tan\alpha - \mu_B}$$

For low coefficients of friction, C^* is positive, so a steady pushing force is required to maintain the braking force. When the coefficient of friction exceeds the tangent of the wedge angle, a negative force is required from the actuator to stop the wedge being pulled further in.

Optimum performance is obtained when operating around the point at which the characteristic brake factor is infinite, since this minimises the actuation forces. From a control standpoint, this is a point of neutral stability, since any small perturbation in the wedge position will result in it remaining in the new position. When the coefficient of friction increases, the wedge position becomes unstable and needs to be controlled to stop the wheel jamming. As a result of this instability, it is important to develop an adequate mathematical model of the brake system prior to testing the hardware[2].

The results published to-date have shown the performance of the brake for relatively short braking periods. Both the coefficient of friction and the disc temperature remained relatively constant during the individual brake applications. Further tests have now been conducted on larger dynamometers to examine the performance of the brake when these parameters are subject to a wider variation. This is more representative of the conditions which will be encountered during normal operation.

The objective of this paper is to present the results from these tests. It is split into subsections describing the prototype hardware; the test equipment used; the tests performed; the programme status; and conclusions. Further details of the underlying concept are provided in References 1 and 2.

SYSTEM DESCRIPTION

Although the principle of a controlled wedge brake is relatively simple, the mechanical implementation is critical to its success. Major factors that need to be considered are:

- Elimination of free-play within the drive-train, regardless of component wear;
- Minimisation of friction in the direction of the wedge travel;
- Operation in both directions.

The prototype is best explained by means of a diagram. A section through the system is presented in Figure 1. The design is based around a modular concept suitable for laboratory testing, rather than being optimised for minimum volume and weight, and uses off-the-shelf commercial components wherever possible.

The brake is driven by two brushless D.C. motors[3], mounted at either end of the assembly. Commutation and current control are performed using commercial motor drives with an incremental encoder on each motor shaft. For controlled braking, a moment sensor provides the feedback to the moment controller. Alternatively, the encoder can be used to provide motor position control. Motor rotation is converted to axial motion by means of roller-screws, which are mounted within the rotors on preloaded angular thrust ball bearings.

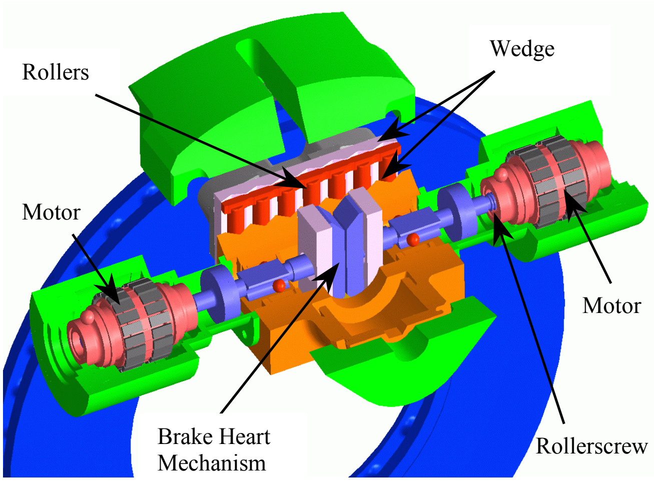

Figure 1. Cross Section through Prototype

The roller-screws drive the so-called brake heart, which contains the wedge mechanism. Within this component, forces are only transmitted by compression between neighbouring surfaces. This allows the motors either to work together or to preload the system and so remove free-play[4]. Backlash is inevitable, both as a result of construction tolerances and due to wear, particularly in the bearing surface which allows the wedge to slide outwards from the motor axis. If they are working together, then one roller-screw pulls the wedge in the appropriate direction while the other one pushes against the first roller-screw. This reduces the motor loads when the coefficient of friction is not near the optimum value. For a preload to be introduced, both roller-screws pull against their respective sides of the wedge.

The wedge is actually composed of two ground 'W' surfaces. The inner one relative to the motors is static, while the outer one moves both axially and in translation. This construction spreads the loads and can generate self-reinforcement in both directions of travel. Between these surfaces, there is a series of rollers, which minimise the sliding friction from the high calliper forces. The outer part of the wedge, to which the brake pad is attached, is held against the static one by a preloaded spring. It is axially actuated via a bearing surface, which allows it to move laterally away from the motor centreline.

TEST EQUIPMENT

The tests described in this paper were performed on a two axis dynamometer at DaimlerChrysler Commercial Vehicle Division in Untertürkheim, Germany (Figure 2). The machine is designed to simulate commercial vehicles and has a rotational inertia of up to 3840 kgm^2, and a maximum permissible braking moment at each station of 30,000 Nm. Its maximum speed is 1200 rpm, and the electrical power rating is 370 kW. For the tests described in this paper, the inertia was set to 75 kgm^2 and the full operating speed range was used.

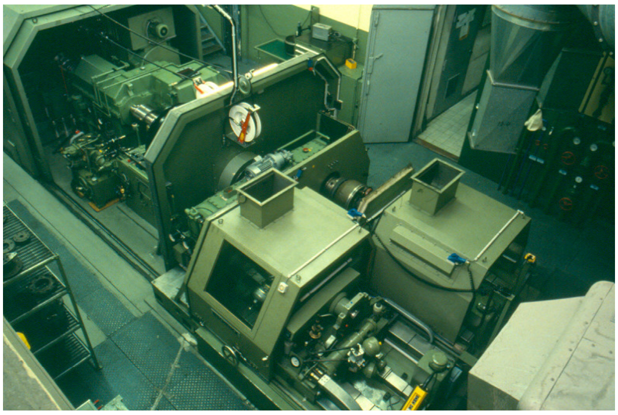

Figure 2. 370kW Dynamometer

Additional testing has been performed in-house at estop on a 125 kW machine, capable of simulating a passenger car up to a weight of approximately 1600 kg and operating up to a rotational speed of 2400 rpm. This corresponds to a speed of approximately 270 kmh or 170 mph. It is illustrated in Figure 3.

Figure 3. 125kW Dynamometer

In addition to the parameters needed to control the brake, there are many additional variables required for analysis of the results. These include the motion of the wedge and calliper, loads within the calliper, and temperatures throughout the system. The controller and data logging system is all run from the xPC Target rapid prototyping environment, produced by the Mathworks. This offers an adequate interface, a broad choice of hardware, and relatively straightforward extensibility for the system. Currently the hardware consists of:

- Host P.C.;
- Target P.C. including:
    - one PCI-QUAD04 Encoder Card;
    - two PCI-DAS1602/16 I/O Boards;
    - one PCI-DAS1200/JR I/O Board.
- conditioning electronics for the sensor signals.

The dynamometer at eStop can be run from within xPC Target, but that in Untertürkheim had to be controlled separately.

TEST RESULTS

A variety of tests have been conducted. The overall aims were to investigate the performance of the brake under the following conditions:

- continuous braking;
- extreme friction coefficient variations;
- large temperature variations.

Friction and temperature variations occur naturally during the use of a brake, particularly when enough energy is introduced to the tribological system. The original test dynamometer (see Reference 2) could not introduce sufficient energy to induce these effects.

Results will be presented for the following test cases:

- Braking to Stand-still;
- Sine wave responses;
- Simulated ABS cycle;
- High temperature test.

For each of these cases, a description will be given of how the test was conducted, followed by presentation of the results and a brief discussion of their significance.

The friction coefficient presented in these results is calculated from the braking moment and the normal force across the calliper.

$$M_B = 2\,\mu_B\, r_B\, F_N \Rightarrow \mu_B = \frac{M_B}{2\, r_B\, F_N}$$

It is thus an average for the two brake pad to disc interfaces.

The brake pads and disc are compatible standard components, but can only be considered a first prototype pairing for this brake. In particular, the testing to date has focussed on the brake in isolation and has not attempted to represent realistic vehicle or certification duty cycles.

STOP BRAKING

In this case, the dynamometer was set to run at a prescribed speed, then switched off and stopped using a constant moment from the brake.

The results from this test are shown in Figure 4. It can be seen that the brake rapidly applies the deceleration torque required and holds it accurately until the system has reached a stop. The highest input power is required to accelerate the motors to cross the air-gap, and once the brake is in contact, the self-reinforcement reduces the power requirements to a very low level. The increase at the end is because the controller was programmed to hold a position in proportion to the moment demand. Once the self-reinforcement is removed, this requires considerably more power. In reality, it is only necessary for the brake to exercise enough moment to prevent the vehicle, and trailer if present, from rolling on a gradient.

Several other features are evident from the plots. The general trend of the wedge motion is to be pulled back during the course of the braking as the coefficient of friction increases. There is also some higher frequency motion of the wedge as the controller attempts to smooth out irregularities in the braking torque.

At the beginning of the input, the coefficient of friction was noticeably lower than the optimal (the initial peak is not a real effect). As a result, both motors work together to spread the load of driving the wedge. This can be seen in the difference in motor positions: motor 1 had to drive an additional 0.8 mm to be able to contribute to the generation of moment. However, during the course of the braking, the friction coefficient increased and the motors then started working preloaded against each other as it was around the optimum value. This prevents uncontrolled steps in the moment due to a change in the sign of the characteristic brake factor.

SINE WAVE RESPONSES

For these tests, the speed of the dynamometer was held constant while the braking moment was controlled. An approximately two second long pulse was set as the moment demand, which was summed with a sine wave of a prescribed frequency and amplitude. The objective was to check how well the brake followed the sine wave demand while under a constant load.

Figure 5 shows the response to a 2 Hz sine wave. It can be seen that the response follows the demand extremely well, showing little evidence of any friction in the drive mechanism despite the constant load on the calliper. The small errors that are apparent appear to result from a once per revolution drop in the coefficient of friction by approximately 0.03 (in 0.38) over a sector of the disc. The controller attempts to compensate this but only has a finite reaction time. As a result, it over-compensates as the coefficient rises again, resulting in some peaks below and some above the ideal line.

In Figure 6 a 10 Hz sine wave has been used for the tests. A phase shift is now evident in the response, but the signal is still followed relatively well. The power required to drive this motion can also be seen to be very small. Finally, a 20 Hz case is illustrated in Figure 7, where phase lag of the order of 90° is visible. Again the power requirements are moderate.

These tests demonstrate that there is potential to use the brake to compensate for dynamic thickness variation in the disc up to quite high speeds. However, the additional phase lag of the response needs to be compensated as the speed increases.

SIMULATED ABS RESPONSE

Exercising the ABS must be simulated on the dynamometer because of the lack of a tyre/road interface. Ideally, a (quarter) vehicle model should be included in the simulation to provide simulated dynamic variation of the tyre/road coefficient of friction and the normal load on the wheel. This would permit realistic brake action based on the response of the simulated vehicle. This has not yet been implemented.

As a first step, the control software has been modified such that when the braking moment exceeds a given threshold, the command is given to reduce it until it is below another threshold. Both thresholds are unknown to the controller, so no shaping of the responses can be used to optimise the test results. This logic is designed to be similar to the wheel acceleration-based pressure cycles described in [5]. For initial tests, the lower threshold is set to be 80% of the higher one.

A sample result in Figure 8 shows the initial overshoot on the braking moment as it exceeds the first threshold, before being pulled rapidly back to below the lower one. After this point the moment climbs again back to the higher threshold, but at a more gentle rate. This results in less overshoot in the next cycle. The power consumption during the cycle is still relatively low despite the fact that the coefficient of friction is greater than the tangent of the wedge angle. This means that the wedge is being pulled in, so the motors are having to work against the wedge forces when reducing the moment, which is the fast part of the cycle. This represents a worst case for the power consumption.

HIGH TEMPERATURE TEST

The main objectives of the high temperature test were to check the robustness of the mechanical design and to investigate the maximum braking moment generated with an unfavourable coefficient of friction. The reduction in self-reinforcement under these conditions means that the wedge brake is more sensitive to this factor than conventional brakes.

Figure 9 illustrates a good example of fading. For the test, the initial ambient temperature in the test chamber was around 28°C and there was no cooling air provided.

During the test, the temperature on the backing plate of the passive pad exceeded 600°C and the measured coefficient of friction dropped to approximately 0.14. Instead of the demanded 1300 Nm, the brake only produced around 700 Nm, with both motors working co-operatively on the current limit programmed in the controller. This response would be unacceptable in a series application. The main cause of this reduction in moment was that the wedge reached its travel limit, preventing the brake from generating any further normal force. This is a problem for the current prototype, which does not have an automatic wear adjustment mechanism to compensate for removal of material from the brake pads. The next generation prototype will correct this anomaly, allowing more realistic test results to be produced.

The steady state power requirements (ca. 125 W) are purely determined by the software current limit, since the motors are stationary. The test was repeated several times without damaging the hardware. Figure 10 shows the appearance of the brake during a different high temperature test.

Importantly, it can be seen that the motor temperatures are both around 50°C, representing a rise of only 22°C. This demonstrates that there is scope for increasing the current limit without causing the motor to over-heat, and hence to increase the braking moment at least for a limited period of time. However, the amount by which this can be done will depend on three major factors. Firstly, a rapid increase in temperature will result in a more uneven distribution through the brake. Thus a prolonged heat soak followed by a fading test (or vice versa) will probably represent a greater challenge. Secondly, the motors are the parts of the structure which are furthest away from the heat source. When a more packaged version of the brake is produced, the motors will be closer to the brake heart and so will lose some of this advantage. Thirdly, the power drain on the supply must not be too high.

While there is still clearly further study required, particularly in representative vehicle installations, these results are an important step on the way to demonstrating the robustness of the design. The next major challenge in this respect is demonstrating the robustness with local electronics.

PAD WEAR

The tests conducted at Daimler did not include enough braking time to give representative results as regards brake wear. However, the wear on the brake pads has been monitored periodically during subsequent longer-term testing on the dynamometer in Seefeld.

Concern has been expressed by several parties that the pads will wear unevenly, particularly tangentially to the disc. The measurements made so far indicate that this is not the case. Radial thickness differences are larger than the tangential ones, even on worn brake pads.

This issue will continue to be monitored, particularly once endurance testing begins. Nevertheless, first indications are that the bearings in the brake heart do distribute the pressure evenly across the pad.

PROGRAMME STATUS

The results presented here are based on testing of the second prototype. The functionality of the brake has now been demonstrated for longer brake applications, for extreme friction coefficient variations, and for large temperature variations.

The next stage is the construction of a third prototype, which is scheduled for the summer of 2004. This prototype will include features which are absent from the existing one and which will be required for a practical installation. The most important of these are the adjustment for pad wear, which will include an emergency release mechanism, and more robust motor position sensors. This system will be tested first on the dynamometer, with the objective of starting vehicle testing by the end of the year.

Parallel developments are aimed at taking the electronic hardware closer to a system that would be suitable for series production, and implementing a detailed failure management strategy.

CONCLUSION

This paper describes testing conducted on the prototype wedge brake. The results presented here demonstrate that the system can handle a range of realistic parameter variations and still exhibit the fast dynamics and low power consumption predicted.

This is another important step in demonstrating that the concept is not only of theoretical interest, but that it can be translated into a robust and practical braking system.

ACKNOWLEDGMENTS

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

C*	Characteristic brake factor
DLR	Deutsches Zentrum für Luft- und Raumfahrt e.V.
F_N	Normal force across calliper
M_B	Braking moment
r_B	Effective radius of brake pad
α	Wedge angle
μ_B	Brake pad to disc coefficient of friction

APPENDIX

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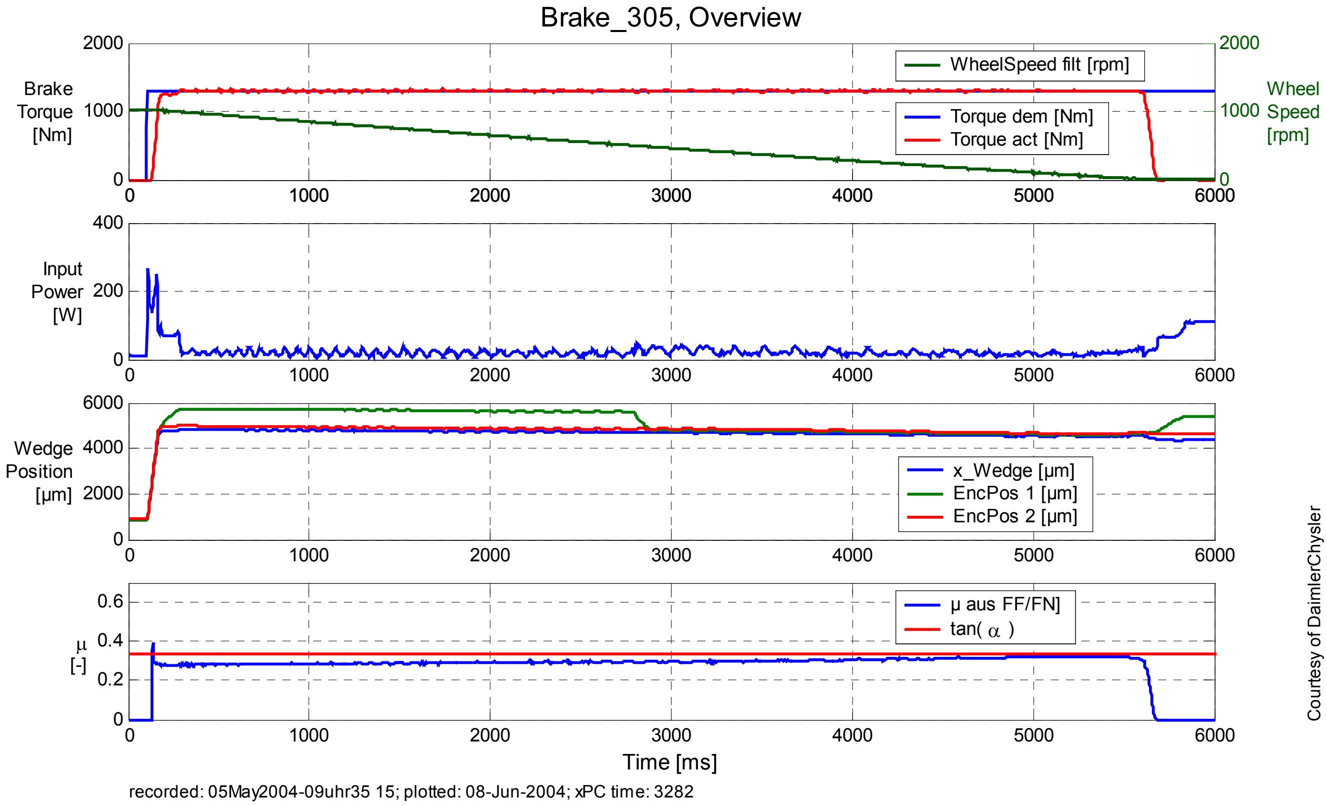

Figure 4: Stop Braking

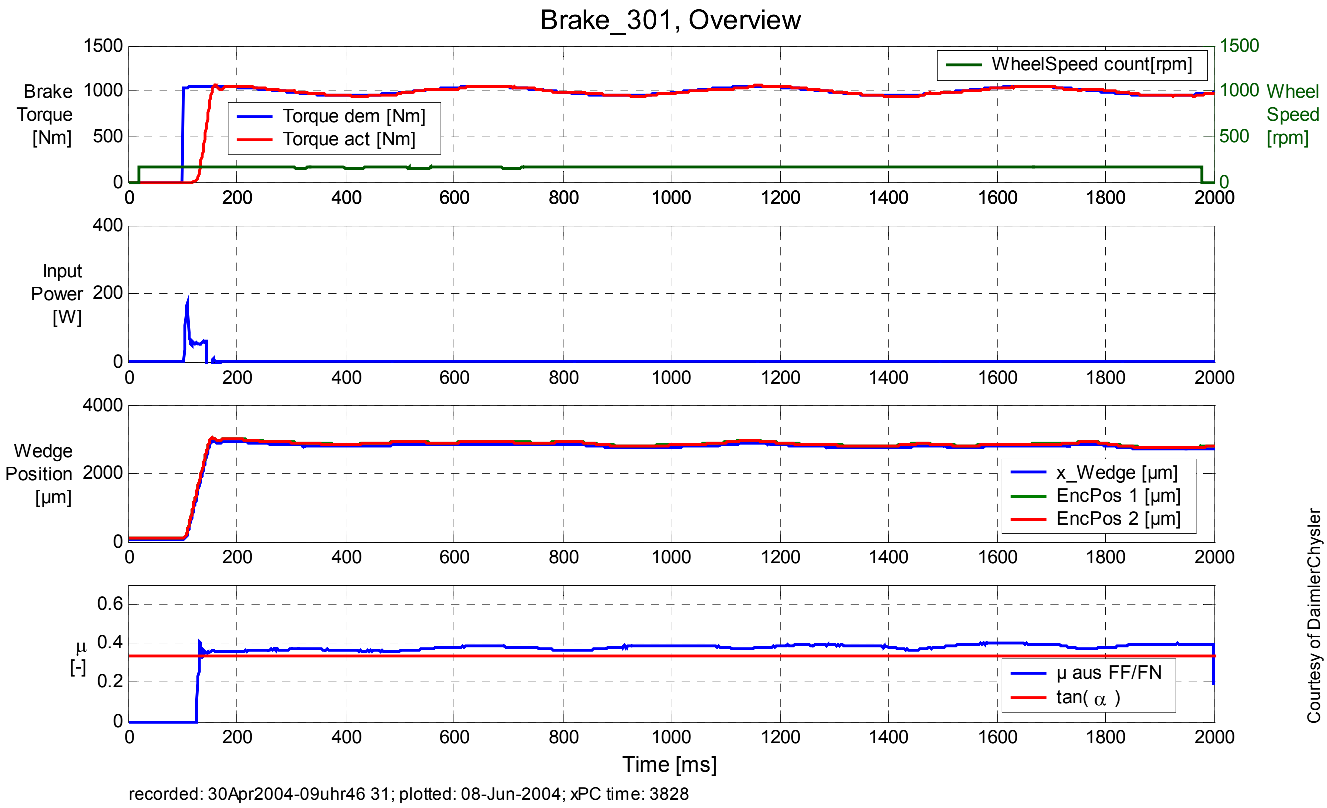

Figure 5: Sine Wave Demand: 2 Hz

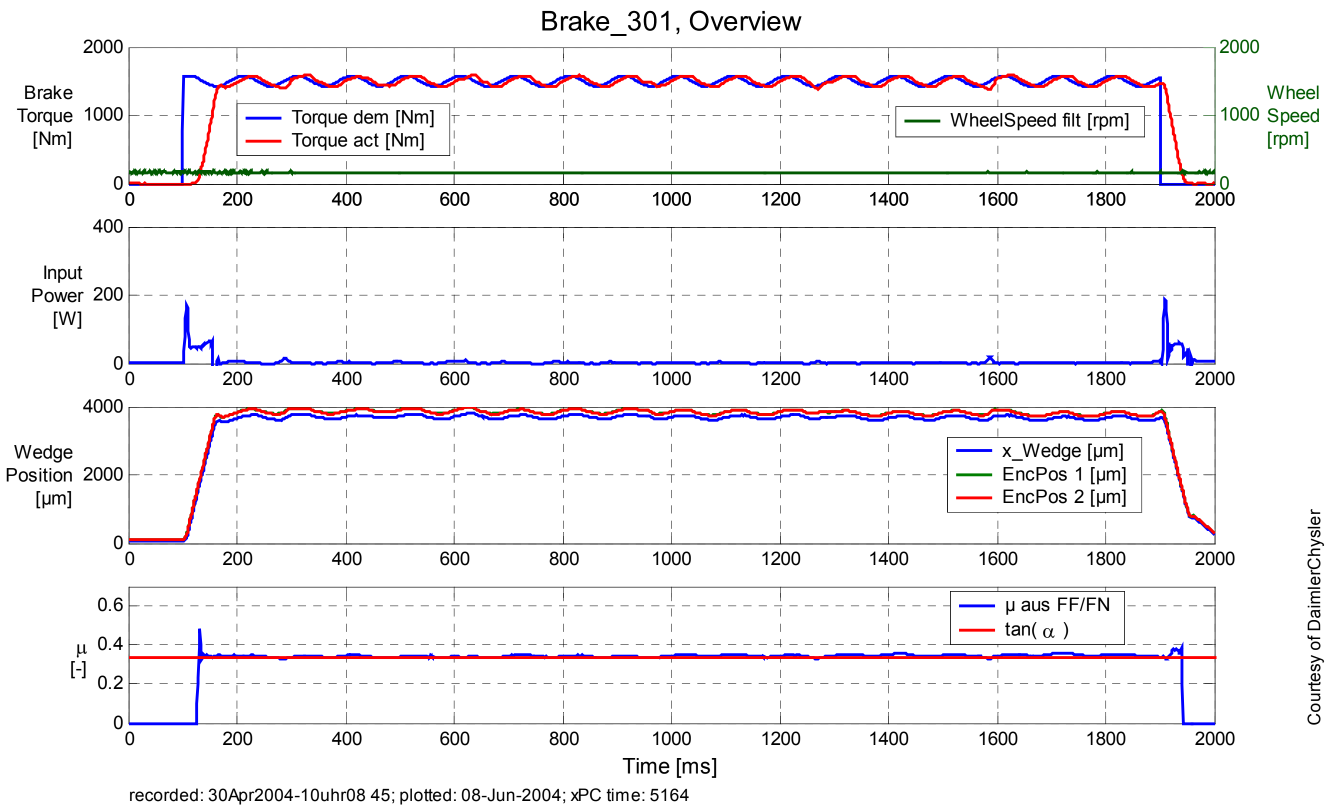

Figure 6: Sine Wave Demand: 10 Hz

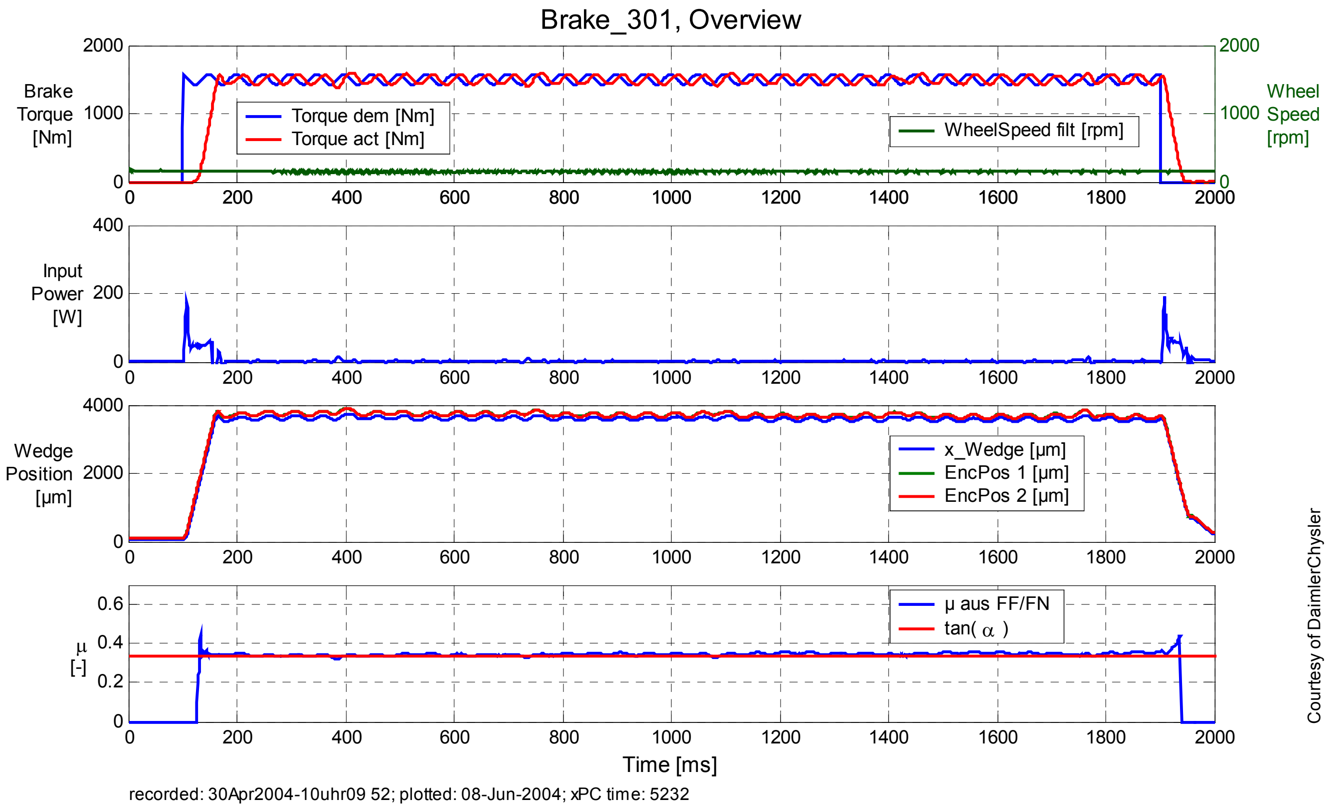

Figure 7: Sine Wave Demand: 20 Hz

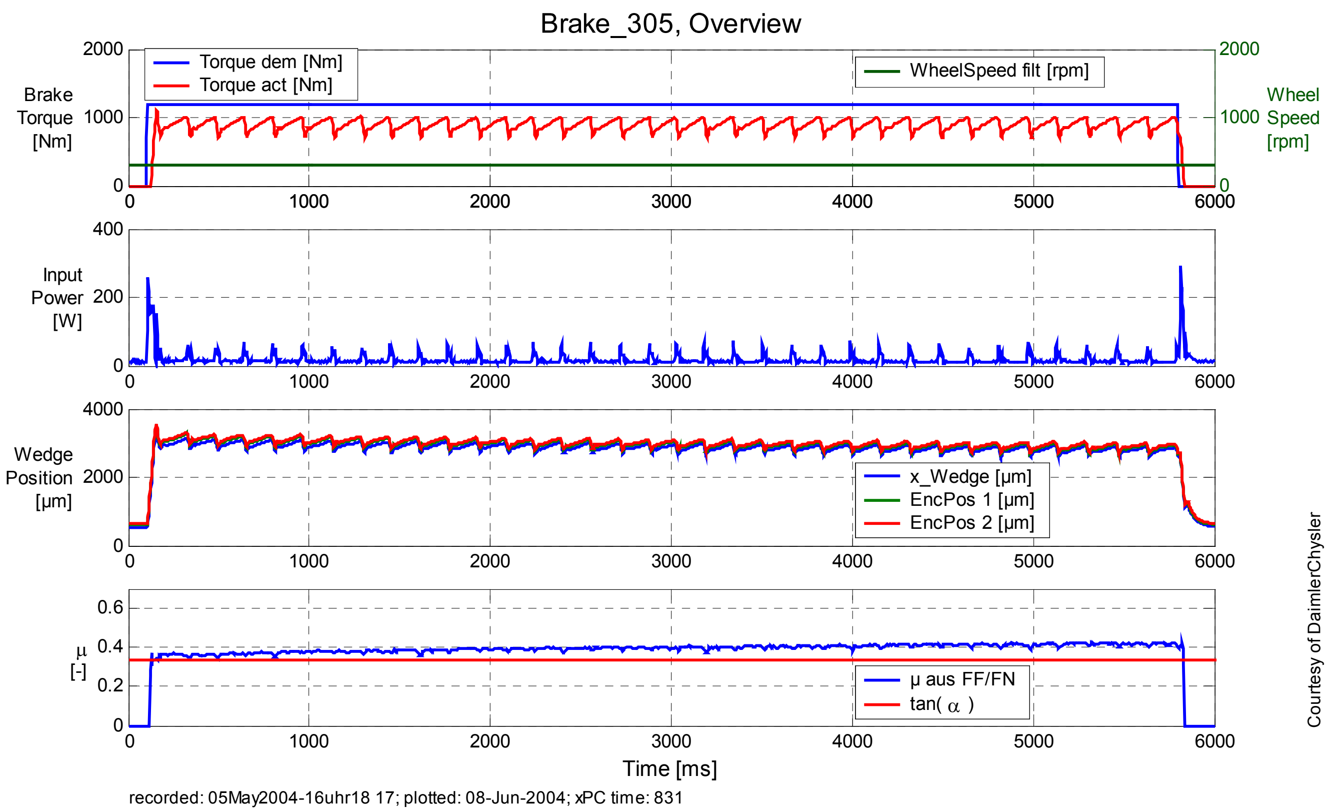

Figure 8: Simulated ABS Response

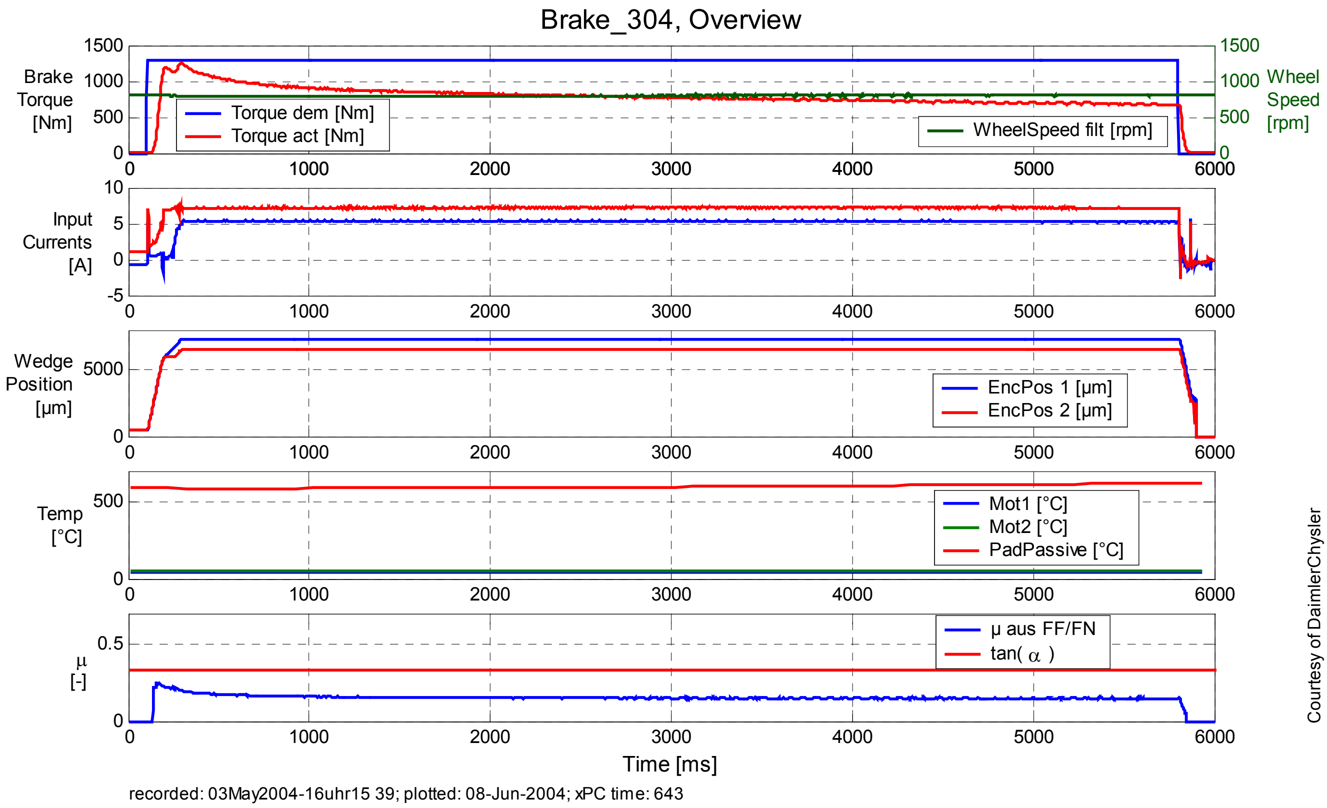

Figure 9: High Temperature Test

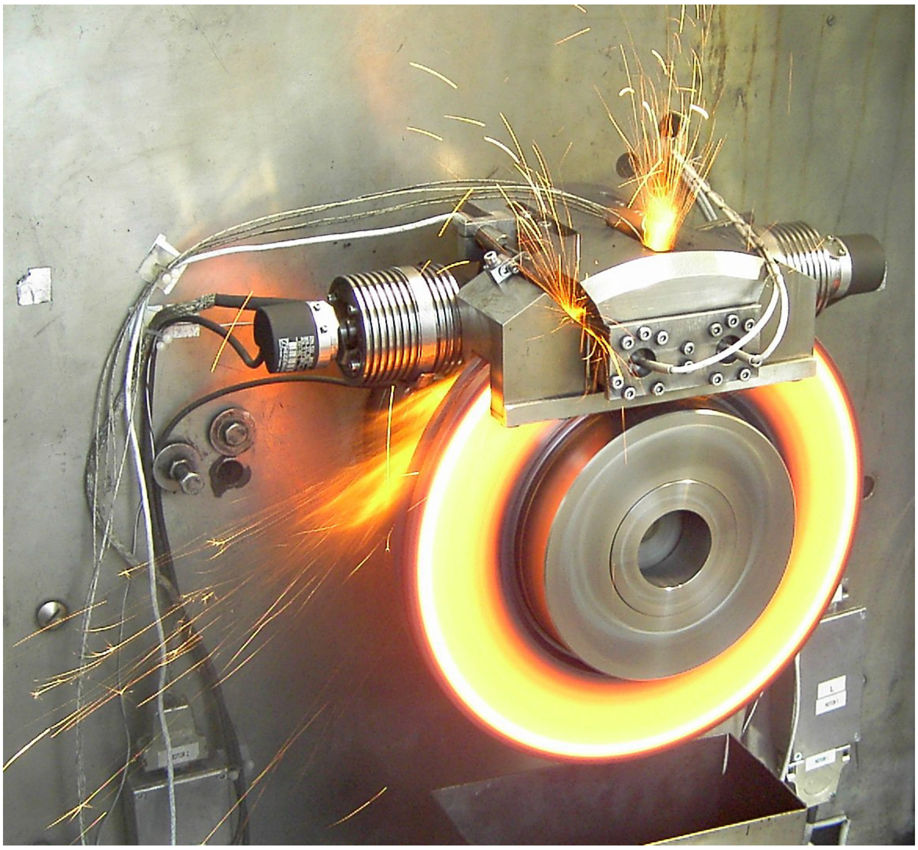

Figure 10: High Temperature Test