

Modeling and Control of a Single Motor Electronic Wedge Brake

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ABSTRACT

Siemens VDO is currently developing a brake-by-wire solution called the Electronic Wedge Brake (EWB). Earlier prototypes used a two-motor concept inside the brake actuator to drive the wedge. In this paper, a new prototype generation is presented where only one motor is needed. This is more efficient in terms of cost, and actuator weight, and also reduces the complexity of the control strategy. A state-space model is derived for the new actuator and two controller structures are explained. Simulations and experimental results from a roller test bench are shown.

INTRODUCTION

In the past few years, much effort has been spent at Siemens VDO to develop a brake-by-wire solution called the Electronic Wedge Brake. Because the brake uses the self-reinforcing wedge principle, the actuation forces and therefore electrical power requirements are much lower than in conventional electromechanical brake systems. Tests on dynamometers and in prototype vehicles have already proven that the EWB can outperform hydraulic systems with regard to braking distance and driver comfort, particularly on low friction surfaces.

Previous publications have described a realization where two motors are used to drive the wedge. By this means, it is possible to create a preload in the drive system such that no backlash can occur. In this paper, a new single motor concept is presented, which satisfies the low backlash requirement mechanically. The actuator differs from the tandem motor implementation in both its drive-train concept and geometry, and therefore a new state-space physical model is developed. The nonlinearities within this system, most notably the effect of braking direction, and their effects on the accuracy of the model are discussed.

The wedge principle which provides the self-reinforcement can also lead to an unstable open-loop system. Therefore, a control algorithm has to be carefully designed to guarantee the closed-loop stability under all circumstances. It must be robust to the variation of

parameters within the new design, particularly to the coefficient of friction between pad and disk, but also to brake pad wear and thermal effects. Two different control approaches are highlighted here: one is based on a cascaded controller which was originally developed for the tandem-motor prototype. The second concept uses full or partial state-feedback and is able to generate almost identical step responses for a broad range of parameter variations.

The performance of the second controller is demonstrated in simulation results. For the cascaded controller, experimental results are included, which show the potential of the new prototype to fulfill the requirements of a modern brake system.

HISTORY OF THE ELECTRONIC WEDGE BRAKE

The electronic wedge brake (EWB) is a self-reinforcing, electro-mechanical brake system. The self-reinforcing operating principle allows reduction in the actuation force, such that system is able to realize the required braking forces using existing 12V vehicle electrical systems.

The initial investigation of the wedge brake concept is described in [1]. The concept was then further developed by eStop, which introduced the alpha-prototype, with the ability to realize the full braking force expected from a modern disc brake system [2,3]. At the center of the alpha prototype design is the brake heart, which realized the wedge braking principle, and the dual motor concept, which removed backlash in the transmission (Figure 1).

In 2003, this was followed by the beta prototype (Figure 2), which was a more robust development while retaining the same basic mechanical concept. The beta prototype also contained other advanced functionalities, including brake pad wear adjustment, and a fail-safe mechanism which reduces braking force to zero when electric power is lost. The fail safe mechanism is required to prevent uncontrolled braking due to the self reinforcement of the EWB. The beta prototype was subsequently installed onto dynamometers, test trailers and test vehicles, where it was validated in real-world conditions, including

demanding winter tests. Its development and the testing results can be found in papers by Siemens AG [4,5].

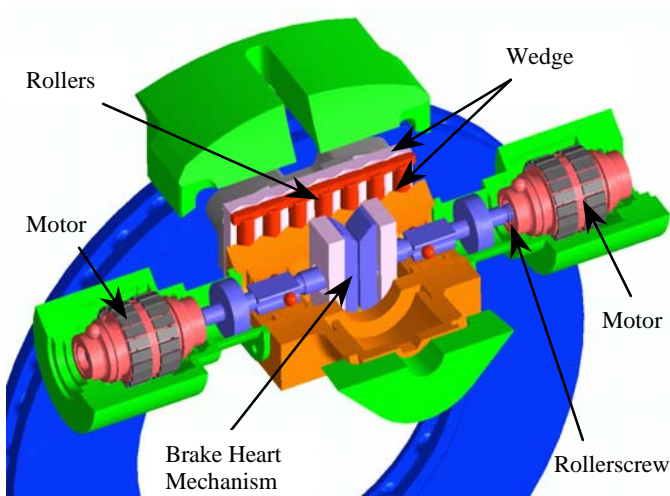


Figure 1: Dual Motor Solution as Used in the Alpha Prototype. The Brake-Heart Mechanism is shown.

In previous publications, it was shown that the stability of the wedge brake varies with the coefficient of friction between brake pad and disk. At low coefficient of friction, the net force on the wedge acts to push it back out of the caliper, while at high friction coefficient, it pulls it in. A change in this parameter can therefore result in the wedge jumping across the backlash in the drive mechanism, resulting in a step change in braking force. To solve this problem, the alpha and beta prototypes both used a tandem motor design, such that the two motors can be used to preload the drive train.



Figure 2: Beta Prototype

Although the tandem motor solution has proven to be a reliable and functional method to remove backlash, there is the obvious disadvantage of increased cost and

complexity over a single motor solution. As a result, the next generation, more production-oriented prototype, denoted in this paper as the prototype 1 (PT1, Figure 3), was designed with the constraint that only one motor will be used to control the wedge mechanism .

THE NEW PROTOTYPE 1 CONCEPT

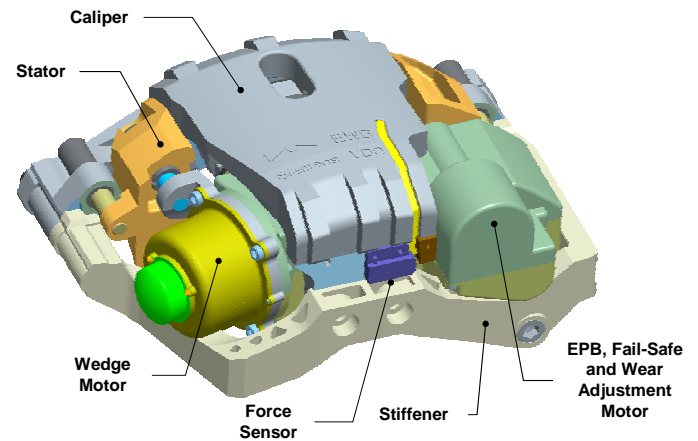


Figure 3: PT1 Prototype (90kN / 4500 Nm)

The PT1 wheel unit concept uses two electrical motors in total, each with its own position sensor. While the EWB based service brake is actuated by the first motor, parking brake, wear adjustment and fail-safe function are realized using the second motor, compare Figure 3.

Position sensors on each motor and additional clamping force and fail-safe sensors allow a precise control and enhanced self diagnostics including plausibility checks of motors and sensors.

An integrated Wheel Control Unit (WCU) allows the minimization of cable harness efforts and electro-magnetic interferences.

SINGLE MOTOR ACTUATION FOR THE SERVICE BRAKE

The service brake is based on wedge brake technology which is combined with a floating caliper, compare Figure 4. In summary, the changes of the service brake wedge drive include:

1. angled drive shaft to increase the spindle durability of braking in the "forward" direction. This also results in a more complex geometry when the wedge is actuated in "backward" direction.
2. new, low backlash power screw concept
3. increased torque class (4500 Nm)

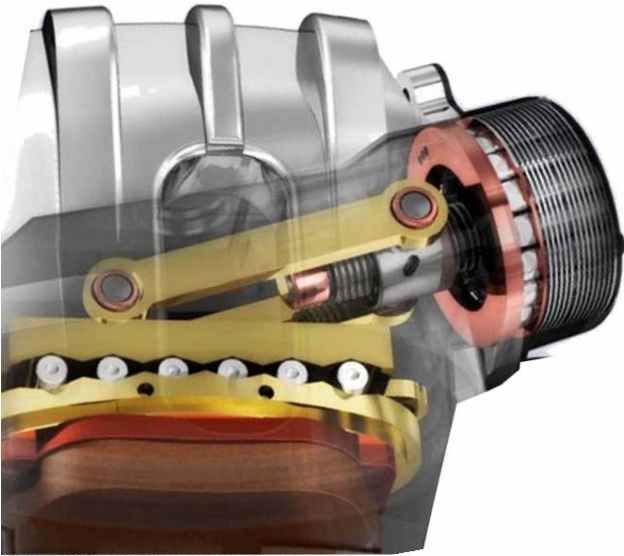


Figure 4: Single motor actuation for the service brake

INNOVATIVE FORCE SENSOR CONCEPT

Classical force sensors for brake by wire systems are typically based on quite expensive pressure sensors derivatives and need a very sophisticated mechanical integration due to a high sensitivity to side forces. As a result, noise to signal ratio is quite poor related to the efforts.

The force sensor principle used for the EWB system measures the caliper deformation. The sample shown in Figure 5 uses the principle to provide a non loaded reference arm which is outside the force flow of the caliper. This arm is created designing a slot inside the caliper. The typical relative movement of the loaded part to the reference surface is approx. 1 mm for the maximum clamping force.

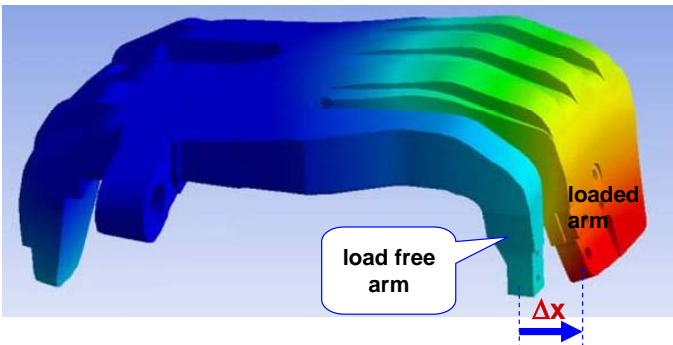


Figure 5: FEA simulation for the deformation of a slotted caliper loaded with a high clamping force

Figure 6 shows the sensor integration in the caliper. The base sensor used is a linear Hall sensor (10 bit) which is combined with a NdFeB magnet.

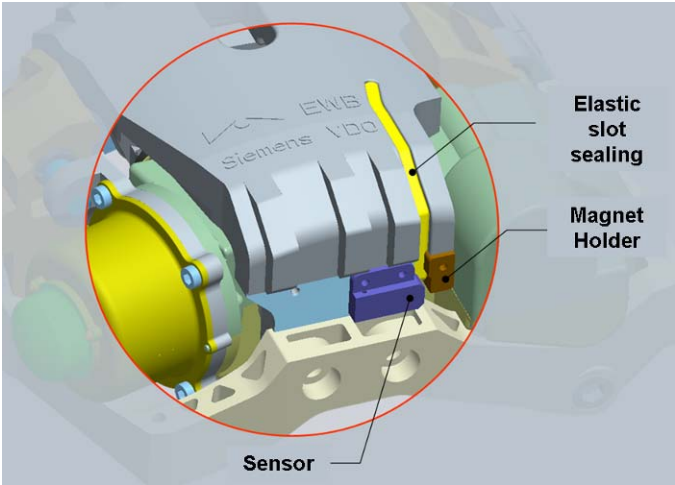


Figure 6: PT1 force sensor integration

The achieved signal quality of the new force sensor concept is significantly better compared with the former prototypes. While the noise of the new force sensor is limited to 0.1% (Figure 7), the noise of the classical sensor is approx. 0.4% (Figure 8). The main reason of the better signal to noise ratio of the new force sensor is based on the low sensibility related to side forces parallel to the disc. While the classical sensor shows a low robustness related to side forces, the PT1 sensor is not being affected by side forces mainly for 2 reasons:

1. the high stiffness of the loaded arm in side direction
2. the hall sensor does not measure the side movement of the loaded arm, but the movement in clamping direction.

Another reason for the high signal quality is the absense of stick-slip-effects inside the sensor system.

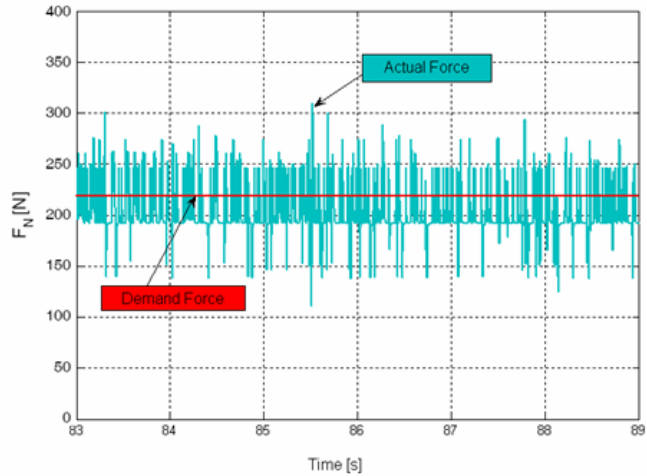


Figure 7: Signal of a classical force sensor in Prototype 1 (max. 90 kN). The noise level is approx. 100 N (0.1% of max. range)

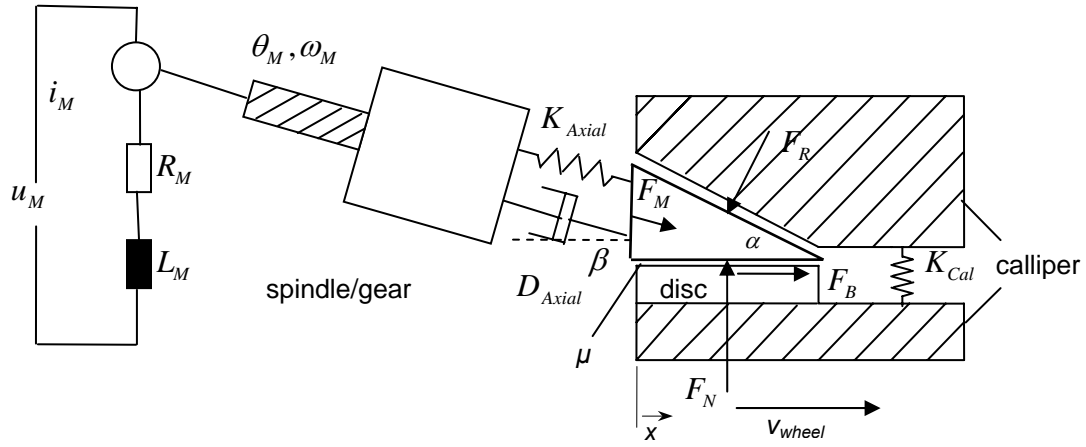


Fig 9: Simplified Model of the Electronic Wedge Brake

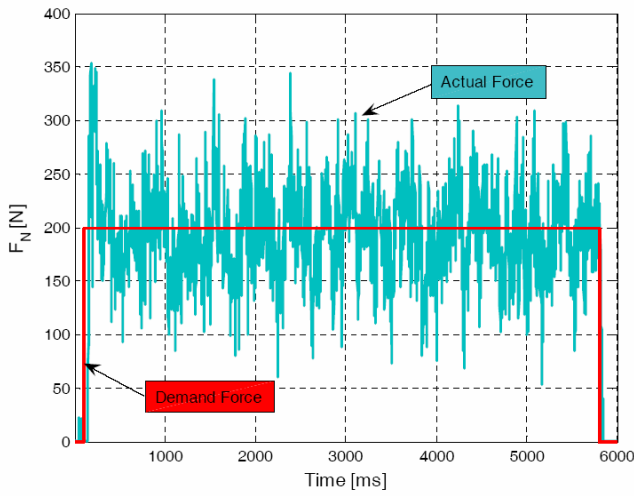


Figure 8: Signal of a classical force sensor in a BETA prototype (max. 40 kN). The noise level is approx. 150 N (~0.4% of max. range)

The following chapters will concentrate on the wedge drive train of the service brake.

STATE-SPACE MODEL OF THE SINGLE MOTOR BRAKE

BASIC LINEAR EQUATIONS

In order to derive a mathematical model for the PT1, consider the simple model of the brake actuator from Figure 9. The wedge has an angle α . Four forces act on it: a motor force F_M , a normal force F_N between the brake disc and the brake pads that are mounted on the wedge, a braking force F_B that results from the relative movement of pads and disc and the coefficient of friction μ between them, and a reaction force F_R which is normal to the wedge edge because the friction on this edge can be neglected. The angle under which the motor force acts on the wedge is called β . If $\beta = \alpha$, the relation between motor force and braking force is optimal, and only this case is considered throughout this paper.

The coordinate x_W is defined to be parallel to the brake disc. In this direction, the balance of forces then reads:

$$\cos \alpha F_M + \mu F_N - \sin \alpha F_R = m_W \dot{v}_W$$

m_W is the mass of the wedge, v_W its velocity in x_W direction. In the direction of the normal force, the force balance reads:

$$\sin \beta F_M - F_N + F_R \cos \alpha = m_W \tan \alpha \dot{v}_W$$

Both equations can be combined to

$$\dot{v}_W = \frac{1}{m_W (1 + \tan^2 \alpha)} \left(\frac{1}{\cos \alpha} F_M + (\mu - \tan \alpha) \tan \alpha K_{Cal} x \right).$$

To calculate the motor force F_M , the system is modeled in axial direction by a stiffness K_{Axial} and a viscous damping D_{Axial} . The roller screw has a lead L and thus transforms the motor angle θ_M , the motor rate ω_M , and the motor torque M_M to the wedge position, velocity, and motor force. Assuming a simple model of the roller screw where one assigns a constant torque/force efficiency $0 < \eta < 1$ to it, one can write

$$M_M = -\frac{L}{2\pi} \left(K_{Axial} \left(x_W - \theta \frac{L}{2\pi \cos \alpha} \right) + D_{Axial} (v_W - \omega_M \frac{L}{2\pi \cos \alpha}) \right) \frac{1}{\eta}.$$

However, this model is inaccurate for many conventional roller screws. They usually show a nonlinear behavior such that the above equation could also be replaced by a general nonlinear term:

$$M_M = M_M(F_M, v_W)$$

In the EWB, the nonlinearity of the roller screw can be approximated by a lookup-table for the simulation models. However, the controllers that are developed in the following section are robust with respect to the magnitude of the roller screw's nonlinearity.

$$A = \begin{pmatrix} \frac{(\mu - \tan \alpha) \tan \alpha}{m_w(1 + \tan^2 \alpha)} K_{Cal} - \frac{K_{Axial}(\cos \alpha + \tan \alpha \sin \alpha)}{m_w(1 + \tan^2 \alpha)} & \frac{1}{m_w(1 + \tan^2 \alpha)} & \frac{LK_{Axial}(\cos \alpha + \tan \alpha \sin \alpha)}{2\pi m_w(1 + \tan^2 \alpha) \cos \alpha} & \frac{LD_{Axial}(\cos \alpha + \tan \alpha \sin \alpha)}{2\pi m_w(1 + \tan^2 \alpha) \cos \alpha} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ \frac{LK_{Axial}}{2\pi J_A \eta} & \frac{LD_{Axial}}{2\pi J_A \eta} & -\frac{L^2 K_{Axial}}{4\pi^2 J_A \cos \alpha \eta} & -\frac{L^2 D_{Axial}}{4\pi^2 J_A \cos \alpha \eta} - \frac{d_M}{J_A} & \frac{k_M}{J_A} & -\frac{R_M}{L_M} \\ 0 & 0 & 0 & -\frac{k_M}{L_M} & \frac{R_M}{L_M} & \frac{L_M}{L_M} \end{pmatrix}$$

$$B = \left(0, 0, 0, 0, \frac{1}{L_M} \right)^T \quad C = \begin{pmatrix} K_{Cal} \tan \alpha & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \end{pmatrix}$$

Fig 10: State-Space Matrices of the PT1 Model

The motor that is used to drive the wedge is a high-performance brushless DC motor. For the purposes of controller design, it can be modeled as a single-phase DC motor with resistance R_M , inductance L_M , a friction (modeled as a viscous damping) D_M , and a torque constant k_M . The motor angle θ_M is measured, its angular velocity ω_M is obtained from an observer. The motor is driven by a voltage u_M . Using these variables, an equation for the motor current i_M can be written:

$$\dot{i} = -\frac{R_M}{L_M} i - \frac{k_M}{L_M} \omega_M + \frac{1}{L_M} u_M$$

STATE-SPACE MODEL

The equations described in the previous section represent a fifth order system which can be written in state-space form:

$$\dot{x} = Ax + Bu$$

$$y = Cx$$

Here, $x = (x_w, v_w, \theta_M, \omega_M, i_M)^T$ is the state vector and $y = (F_N, \theta_M, i_M)^T$ the vector of outputs. The matrices A , B , and C are then given in Figure 10.

The coefficient of friction, μ , is a parameter that will change heavily during the brake's operation. While in a normal operation point, μ would be around 0.4, the control must be able to handle very high friction (it is designed to handle even $\mu = 1$ situations) and zero friction (during standstill, under no load, no braking force is acting and therefore $\mu = 0$). When looking at the system matrix A , one notes that μ can be found in only one matrix element. One can also see that if the axial stiffness is large compared to the caliper stiffness, the influence of variations in μ is reduced. Therefore, a high axial stiffness is advantageous.

KNOWN NONLINEARITIES

It has already been mentioned that the roller screw can be a major source of nonlinearity. This is due to the fact

that its efficiency depends on whether the rotational part is driving the translational part or vice versa. In the simulation models of the EWB, these effects have been handled by a lookup table which is based on direct measurements of the roller screw. Another effect to be modeled is stick friction in the motor and the roller screw.

Another mechanical nonlinearity comes into play when one looks closer at the wedge motion if the direction of the brake disc changes. Since the basic brake mechanism of Figure 9 shows self-reinforcement only for the depicted direction of the wheel, the real brake has a 'double-wedge' construction. Its principle can be seen in Figure 11. If $\alpha = \beta$, then the spindle is parallel to the wedge flank that is used during forward braking (the case described above). If however, the wheel moves backwards, the wedge has to be driven into negative x_w direction. There is an angle of 2α between the spindle axis and the velocity vector of the wedge. Therefore, a rod with two joints has to be attached between the spindle and the wedge. The nonlinear kinematics of this mechanism make the state-space equations more complicated, but result in a similar matrix structure. The new kinematics are a major difference to the Beta prototype where the spindle axis was parallel to the x_w axis.

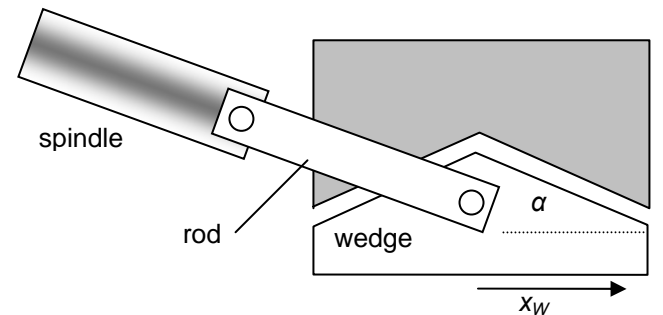


Fig 11: Forward and backward braking

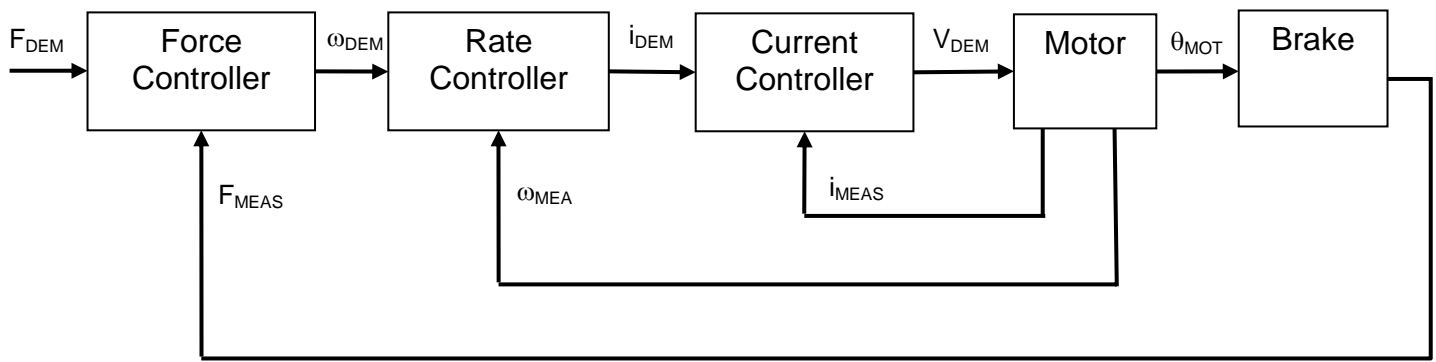


Figure 12: Cascaded Controller Structure

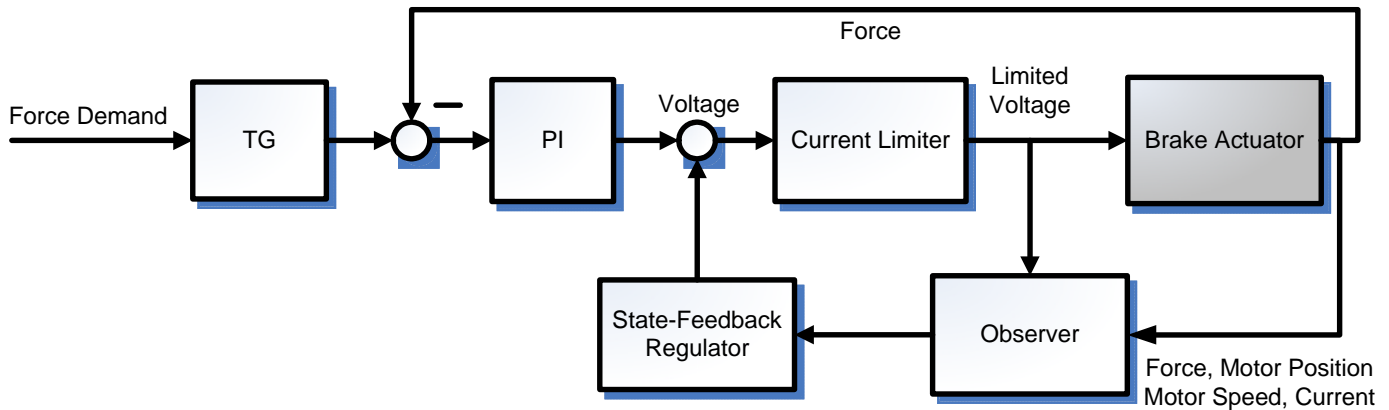


Figure 13: PI State Feedback Controller

CONTROLLER CONCEPT

CASCADED CONTROLLER

For initial commissioning of the brake, a standard cascaded controller has been used. A simplified representation of the concept is provided in Figure 12. This structure has the advantage that each loop can be consecutively built up, tested and validated. The clear relationship between the loops and the physical parameters within the brake also simplifies handling the nonlinearities in the system. Additionally, the loops can run at different sample times and therefore allow the computational effort to be minimized.

The inner current loop is in fact a field-orientated motor torque control. This generates holding and acceleration torques at the motor and is fundamental to the operation of the brake. The rate control loop is tuned to guarantee the linear stability of the brake controller for all values of friction coefficient. The advantage of this approach is that complete flexibility is retained for the outer loop, since there are no longer restrictions on its closed-loop bandwidth. The sensor from which the motor rate is measured is also less affected by mechanical disturbances and noise than the force sensor.

The force loop is responsible for implementing the brake demand required by the driver. One notable difference between control of a wedge brake and a conventional electro-mechanical brake is that it requires motion in

both directions to build up force. The direction chosen depends on the direction of rotation of the wheel. This problem is solved by allocating signs within the controller based on the demanded direction.

The individual controllers within this structure are built up from P, I and D elements, with some additional filtering to shape the frequency response, and varying degrees of feed-forward. The majority of the work involved in developing the controller is associated with producing good basic brake feel for comfort braking on the one hand and high dynamics for ABS on the other.

The cascaded controller concept for the single motor solution is structurally simpler than the control for the Beta prototype which had two motors. In the Beta solution, which was used in the references [2] and [3], a preload control had to be added to prevent backlash effects disturbing the control quality. Since the new prototype is designed such that the freeplay can be neglected over the lifetime of the brake, a preload control is no longer required and the controller has a pure cascaded structure, as depicted in Fig. 12.

PI STATE FEEDBACK CONTROL

A new controller concept which is currently under development for the PT1 is depicted in Figure 13. The core of the controller is a full state feedback which has been designed as an output Riccati controller in discrete time. This approach minimizes a weighted integral of the

output signals. Since not all system states can be measured, namely the wedge position and the wedge velocity which are decoupled from the motor's position and velocity because of the finite axial stiffness, these states must be observed or estimated. Steady-state Kalman filter gains have been used for tuning the observer. For this part of the controller, alternative designs can also be used, for example a partial state feedback regulators or output feedback. In the latter case, no observer would be needed anymore. Indeed, recent research on this topic has shown that output feedback is sufficient to give a good control performance.

If one uses a Riccati regulator, the controlled system is guaranteed to be stable for the parameters used during the design. However, as stated above, at least the coefficient of friction μ must be regarded as highly varying. Also K_{Cal} will vary due to brake pad wear and other environmental influences. Therefore, a robustness analysis of the designed controller has to be carried out. Here, the influence of μ will be examined more closely.

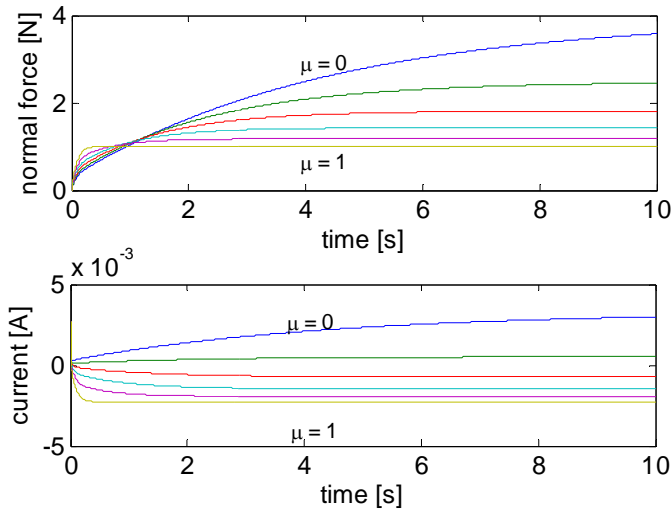


Figure 14: Step Responses of the PI State-Feedback Controller for Different μ (no outer PI Loop)

As can be seen in Fig. 14, the step response in force of the controlled system shows for $\mu=1$ a damped system with no overshoot. For lower μ , the steady-state value is higher and the system shows increasingly less dynamics, but for μ ranging from 0 to 1, the system is always stable. This is due to the fact that the controller has been designed for $\mu=1$, i.e. for the highest admitted friction. The higher μ is, the more the wedge is pulled in and the more control effort is required – the system can be stabilized more easily for lower μ . If however, the controller had been designed for the same step response at lower μ , it would have been unstable for $\mu = 1$.

The step response for $\mu = 1$ shows a rise time of 0.1 s. For this performance, the motor power consumption has

a moderate value. If higher dynamics are needed (for example in braking scenarios where ABS is applied), different controller parameters can be used to generate faster responses.

As can be seen in Fig. 14, if the parameters of the system are not known, state-space controllers do not have zero steady-state error. Therefore, an outer PI loop is added which controls the error between the measured normal force and the force demand.

The tuning of the Riccati regulator is very intuitive because with the weighting coefficients, you can choose whether you want to put more controlling effort to influencing the outputs, state, or inputs. The outer PI loop is also very easy to tune because it doesn't regulate the whole system but only the errors of the inner state feedback controller.

Two blocks in Fig. 13 need further explanation. Just before the actuator, the controller's output can be modified by the current limiter block. This is done to allow only for a limited motor current. In the cascaded controller, this can be done by simply limiting the input to the inner current loop to a certain value. Since however the new approach does not have an inner current loop, the current limiter is slightly more complicated. Here, a model based approach is used which also has the flexibility to limit other values like the motor power consumption.

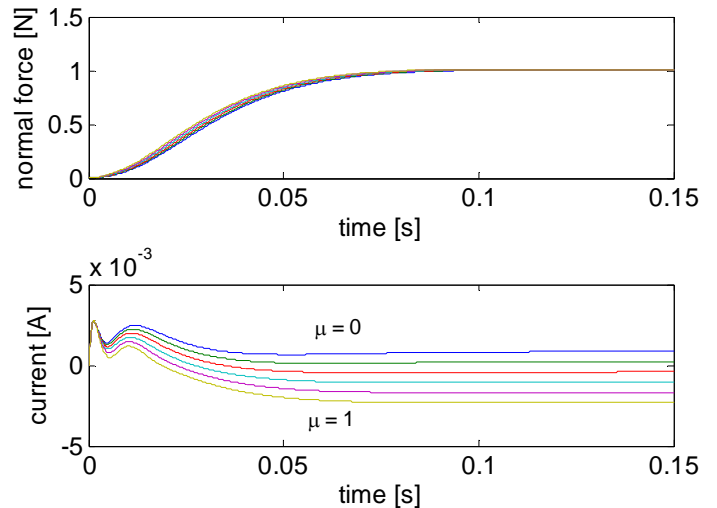


Figure 15: Step Responses of the PI State-Feedback Controller for Different μ (including outer PI Loop)

The block called TG is a trajectory generator. It is inserted to shape the force demand that comes from outside the actuator. The aim of the trajectory generator is to ensure that only demands that can be handled physically by the actuator are passed to the control algorithm. The trajectory generator can also modify the demand in different braking scenarios and thereby slow down the controller, for example during comfort braking,

or speed it up when ESP, ABS, or emergency braking are in action. This has the following advantages:

- the regulator reactions can be designed without touching the control loop itself – therefore, stability is not influenced by the trajectory generator
- energy saving algorithms can be designed if high brake performance is not needed
- the generated trajectory can be used for diagnosis: since it is designed such that it outputs only physically possible trajectories. Its outputs can be compared to the measured force and, if the difference between them is too large, a malfunction of the brake can be detected

In Fig. 15 the step response of the complete controller structure is shown. Because of the outer PI loop, the differences between $\mu = 0$ and $\mu = 1$ have been reduced to a very small value. The controller is stable for all μ and shows a step response without overshoot. The step response in current however is different for different μ . This is due to the varying amount of self-reinforcement.

Another way to realize a controller for the EWB would be to combine the cascaded controller with the PI state feedback. The inner current loop of the cascade is then the same as in the original approach, but the rate and

the force loop would be replaced by a PI state feedback. By doing this, one combines the flexibility and robustness of the new approach with the simple implementation of the current limits. Additionally, one can process the state feedback control loop at a lower sample rate than in the full state feedback approach. Only the inner loop has to be processed at the high rate. Thus, one reduces the computational effort. These issues are under investigation. The new controller concept has only been tested in simulation so far. But since the design step is partially automated, it can be easily adjusted to mechanical design variants of the actuator. An efficient implementation of the controller is ongoing while the tests described below are carried out with the original cascaded controller.

EXPERIMENTAL RESULTS

ROLLER TEST BENCH RESULTS

Initial testing of the new brake was conducted on a roller test bench. This consists of a driven drum upon which the test wheel is mounted. The test bench is capable of producing a torque of 5000 Nm and a longitudinal velocity of 250 Km/h. The speed of the drum is normally controlled but it is also possible to brake against the rotating inertia of the test rig. The vertical force on the

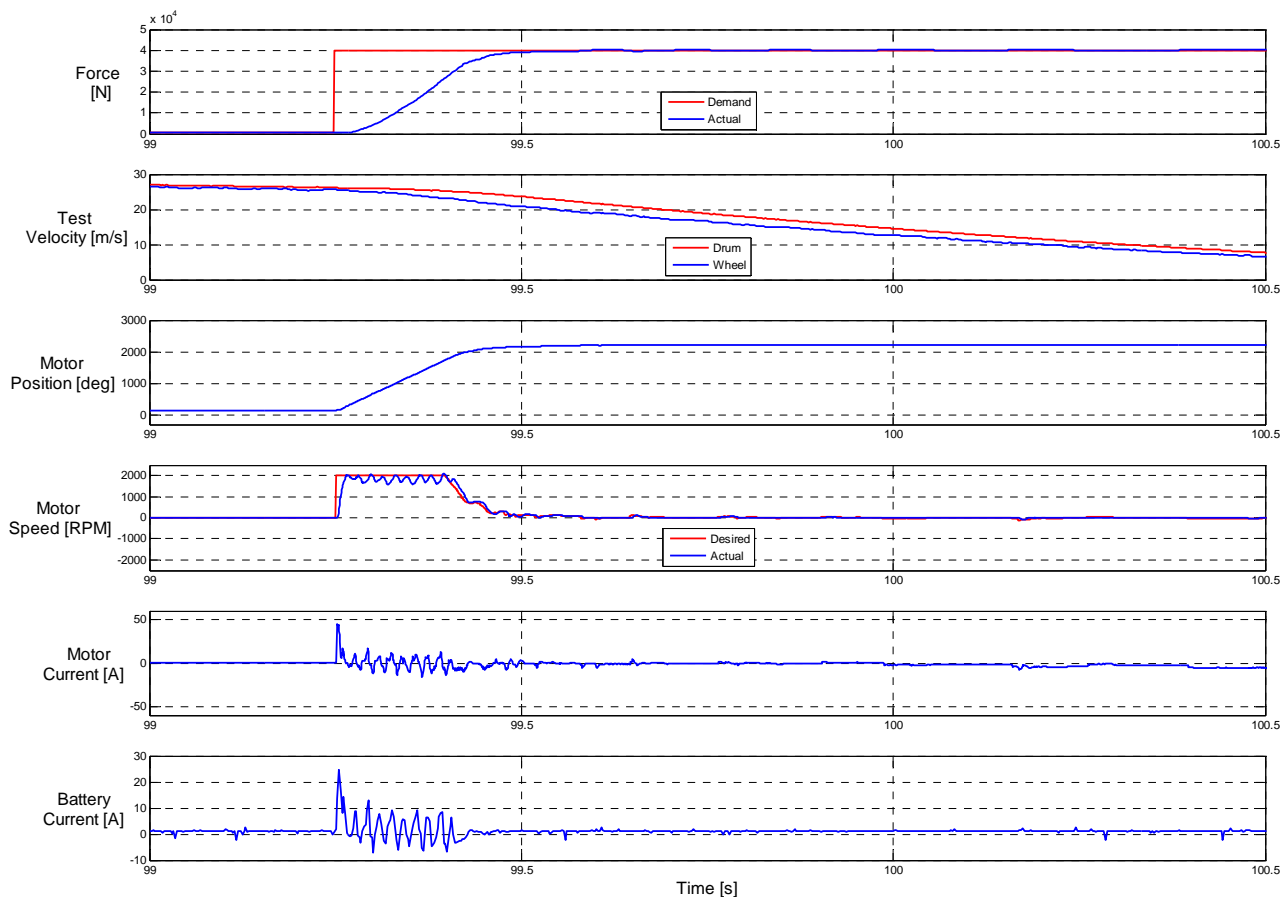


Figure 16: 40kN Step Response

tire can be varied to limit the maximum braking moment.

The cascaded controller described in the previous section is embedded on a microcontroller, and the data communication flow between the EWB electronics and the GUI is done via CAN.

The results presented in this paper were generated using an initial version of the controller, prior to optimization of the mechatronic system. Consequently they represent relatively slow dynamics compared both to earlier results from the tandem motor concept [3] and to the current state of development of this actuator.



Figure 17: Roller Test Bench

For these tests, the actuator was driven directly from a 12V vehicle battery, so the measurements of battery current taken correspond exactly to what would be seen on the road. Battery current is positive when power is being taken out of the battery, and negative when regeneration occurs.

The first example is a 40kN step demand in brake normal force, illustrated in Figure 16. The battery current peaks during the initial acceleration of the motor at about 25 A and afterwards oscillates somewhat in response to the demands of the rate controller. Once the brake demand is reached however, it almost settles back to its quiescent level and the power draw is negligible (of the order of 15 W). This is typical of the results seen with all versions of the EWB to date.

The motor rate control is a little uneven due to the motor running close to the voltage limit, resulting in some oscillation in both the motor and battery currents. This is an aspect of the control which is currently being worked on.

For comparison, a smaller step response is shown from 1kN normal force to 5kN and back (Figure 18). A similar pattern is evident in the battery current in this case too, with both steps leading to sharp peaks in the power as the motor is accelerated. The much smaller demands are both followed accurately by the controller, so comfort braking with this actuator will be satisfactory.

As a final example, a slow ramp in force is shown up to

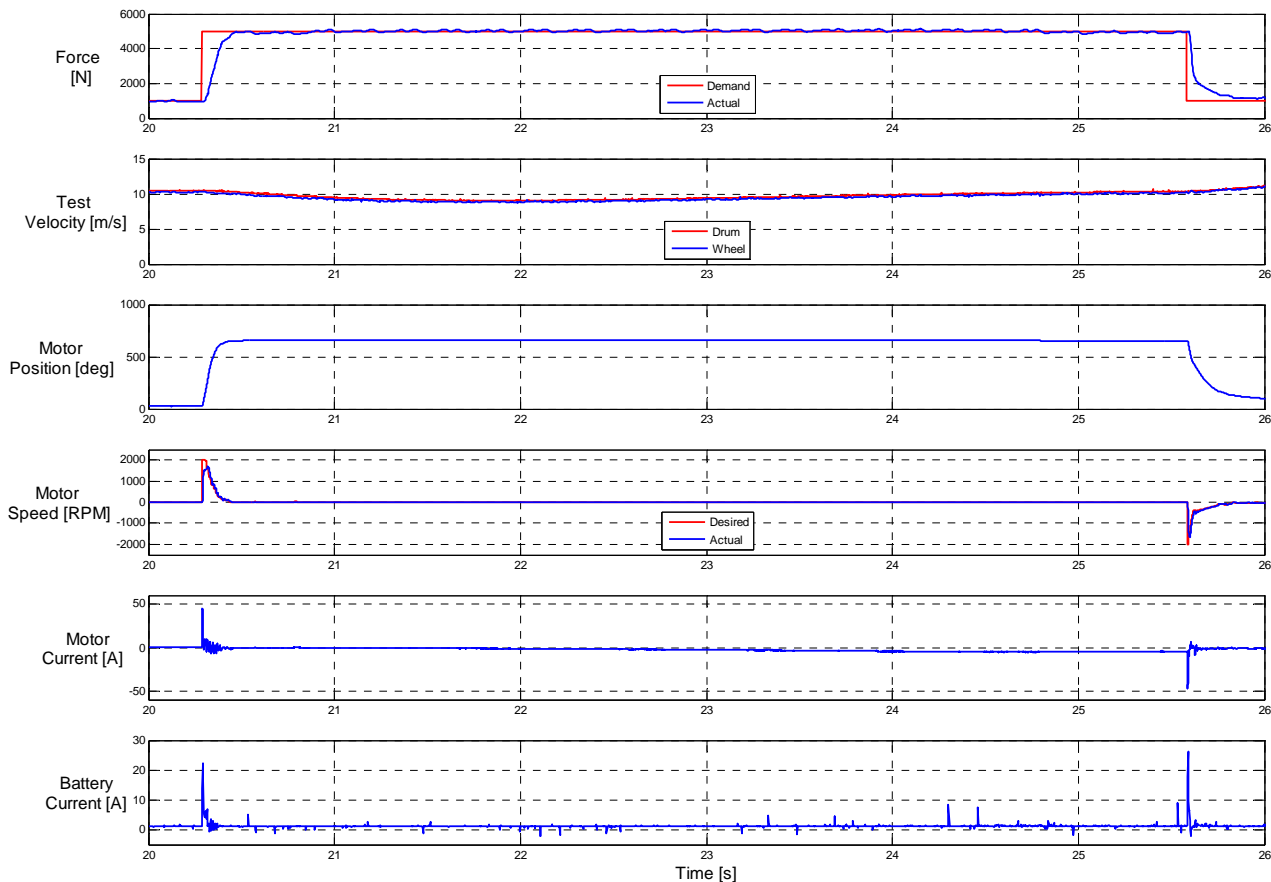


Figure 18: 5kN Step Response

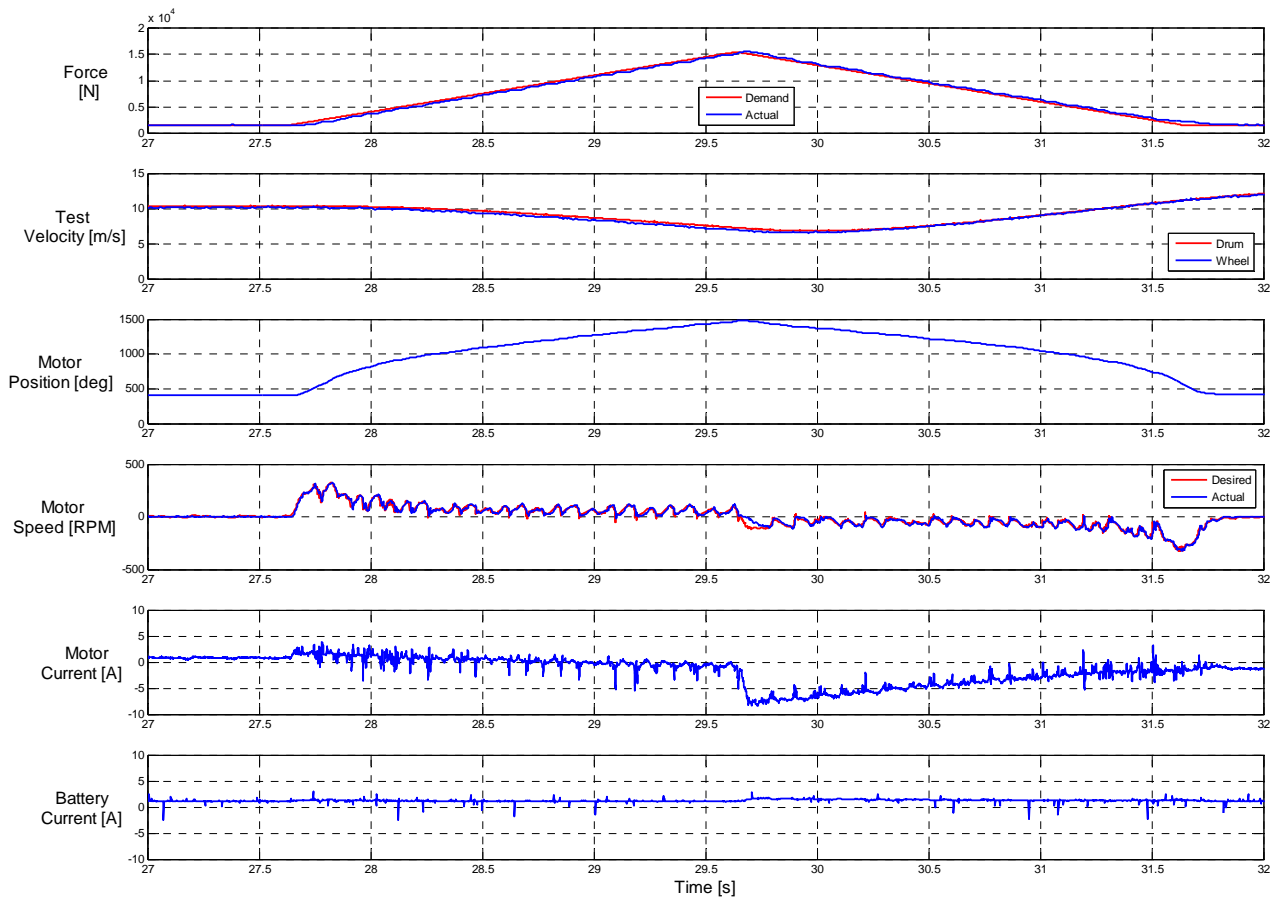


Figure 19: 15kN Ramp Response

15 kN and back. This is illustrated in Figure 19. It can be seen that subjectively the ramp is followed quite well, although there is some unevenness in the rate demand.

Because of the low wedge angle, the wedge is being pulled in throughout all the experiments described here. At the start of the response, a positive motor current can be seen which indicates that the motor has to overcome losses in the drive mechanism and push the wedge in. As the force increases however, it has to prevent the wedge being drawn further into the brake. Because of the different driving and back-driving efficiencies of the roller-screw, the motor current required to 'allow' the wedge to move into the brake is considerably smaller than that required to pull it back out again. The battery current, however, is hardly different from its 'no load' value, demonstrating the low power requirements. Fading situations with a very low friction coefficient have also been tested and it has been shown that the controller handles these with the same performance.

One final aspect which can be seen on this plot is that the brake caliper has a nonlinear stiffness characteristic. This is clear when one compares the triangular form of the force response with the distorted triangle of the motor position. At low forces, the motor has to move relatively much further to generate the same change in force, indicating a much lower value of the caliper stiffness.

CONCLUSIONS AND OUTLOOK

In this paper, an overview has been given of the current status of the controller development of the EWB project. While the design and the performance of the new prototypes are being optimized, other aspects of the brake development are focused. The vehicle dynamics group is developing advanced control strategies for safety and driver comfort, while reliability, robustness with respect to environment, functional safety, and system development are some of the major topics that are being investigated. Vehicle testing is also being performed on high and low friction surfaces to examine the advantages of the new brake system.

The control performance on the brake actuator itself is the backbone for all systems on top of it. The high dynamics of the electrically driven system allow for faster control reactions during, for example, ABS braking. This results in shorter stopping distances and this benefit has already been shown in test vehicles. Thus, driver safety will be improved dramatically in cars equipped with the EWB.

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