Development of A **Self-energising Electro-Hydraulic Brake (SEHB)** for **Rail Vehicles**

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**Abstract**

This paper presents the innovative concept of a **Self-energising Electro-Hydraulic Brake (SEHB)** which is developed at IFAS for a railway application. Its advantages over conventional air brakes are high dynamics, the possibility to control the actual retarding torque, more compact design, higher braking forces and higher efficiency compared to conventionally used air brakes. Due to the concept of self-energisation only low electric power is required for brake actuation and no central hydraulic power supply is needed, thus significantly reducing design interfaces to the bogie. After introducing the working principle of the innovative SEHB system, some safety features are presented that allow load adaptive braking and braking for long periods. The non-linear system simulation gives insight into potential brake performance and supports the development process of the first brake prototype on the basis of an automotive brake calliper. The paper closes with an outlook on the further development of SEHB, including a full-size prototype for tests on a brake test rig for heavy rail vehicles.

**KEYWORDS:** Hydraulics, Self-reinforcement, Self-energisation, Brake, Brake coefficient, Simulation, Pressure control, Friction, Friction coefficient, Trains

**1 Introduction**

Today, pneumatic brake systems clearly dominate in the market for heavy rail vehicle brakes. This is mainly due to the easy handling of “air”, the robust safety concept which can be implemented by an end-to-end pneumatic brake line, and the simplicity of pneumatic couplings between waggons. Advantages of hydraulic brakes are short reaction times, much more compact design and higher braking forces. In commuter rail services, where reduced installation space has superior priority, hydraulic brake systems are well established. In fact, low-floor car concepts would not have been possible without switching from pneumatic to hydraulic brakes. Commonly in these trains the hydraulic brakes are connected to a power unit installed in each bogie. Since this carries the danger of loosing all brakes of a bogie in case of hydraulic power supply failure, the direction of actual developments is towards independent actuators with individual power supply, [2].
This paper presents the concept of a new brake called the Self-energising Electro-Hydraulic Brake (SEHB), which is being developed at the Institute for Fluid Power and Controls (IFAS, RWTH Aachen University) within a research project funded by the DFG (German Research Foundation). The SEHB concept was first published in [1]. It offers the advantages of hydraulic brake actuation without the mentioned disadvantages of a centralized bogie power supply. This is possible by the principle of self-energisation. The wheelset’s inertia momentum is used by each calliper as the source of power to supply hydraulic pressure for braking. Only low electric power for the operation of a small hydraulic valve, pressure sensors and controller electronics is required to operate the brake as explained in Section 2. Section 3 shows that one of the major advantages of this concept is the possibility to control the actual braking torque, unlike conventional brakes that can only control the perpendicular force due to uncertainty of friction coefficient and brake radius. Inherent safety is very important for brake systems in heavy rail trains. With safety features like hydraulic-mechanical fall-back solutions presented in section 4 it is likely to be possible to meet the high safety standards of today’s pneumatic brake systems. SEHB has been successfully studied in simulation with a proportional feedback control. Section 5 discusses the braking dynamics on the basis of an example design. The main aim of the project currently is the verification of the brake principle on a small scale test rig, which will be described in Section 6.

2 Working principle of SEHB

Unlike conventional brakes, where the brake calliper is mounted firmly to the bogie, in the SEHB concept it is movable tangential to the friction contact. A hydraulic supporting cylinder connects the calliper to the bogie structure, thus fixing it between two columns of oil. In case of friction contact one of the oil columns is charged with a certain pressure according to the brake force. This pressurised oil can in turn be used as a hydraulic power source for brake actuation via a control valve, see Fig. 1.

For understanding the SEHB working principle it may be helpful to define the difference between a self-reinforcing and a self-energising brake. For any brake principle the brake coefficient \( C^* \), also known as the shoe factor [3], is a measure for the self-reinforcement. It is defined as the quotient of the total circumferential force on the brake area \( F_{\text{Brake}} \) divided by the tightening force of the brake shoe \( F_{\text{Clamp}} \).

\[
C^* = \frac{F_{\text{Brake}}}{F_{\text{Clamp}}} \tag{1}
\]

For drum brakes the brake coefficient is between 1.5 to 20, [4]. A self-energising brake may now be defined as a brake where the brake coefficient \( C^* \) becomes infinite or even turns negative. This means that no tightening force is needed to achieve any desired circumferential force, and even a negative tightening force (lifting force) results in positive friction. The self-reinforcing brake amplifies an external tightening force with a specific ratio, defined by \( C^* \). In these terms it is always a question of design whether a brake is self-reinforcing or self-energising. However, in most cases a self-energisation is not desirable because it leads to an unstable process and self-locking, if not controlled.

Fig. 2 compares two popular self-reinforcing principles, the drum brake and the wedge brake. The braking coefficients \( C^* \) depend on the design of the angle \( \alpha \) and on the friction
Figure 1: Principle of self-energising electro-hydraulic brake, (SEHB)

Figure 2: Comparison of the self-reinforcing principles for drum and wedge brake

Coefficient $\mu$:

$$C_{\text{Drum}}^* = \frac{\mu \tan \alpha}{\tan \alpha - \mu}, \quad C_{\text{Wedge}}^* = \frac{\mu}{\tan \alpha - \mu}$$  \hspace{1cm} (2)

In both cases, designing the brake such that $\tan \alpha$ becomes smaller than $\mu$, self-energisation is reached and the brake will be self-locking, if not controlled. The idea of the electronic wedge brake, developed by Siemens VDO, is to control the position of the wedge pulled into the self-energising friction contact with electric spindle nut drives. The required power of a tested prototype for automobiles is 100W in average during a brake actuation which is fairly low compared to concepts of direct actuation, [5], [6].

Similarly the self-energising electro-hydraulic brake has design parameters that determine whether the brake is self-reinforcing or self-energising. The following considerations apply in the case where the right flow scheme of the proportional brake valve in Fig. 1 is active. Given an actual brake force, the smaller the $A_{\text{Sup}}$, the higher the pressure in the supporting cylinder. Consequently, the bigger the $A_{\text{BA}}$, the higher the perpendicular force
is achieved. The brake coefficient $C^*$ of the self-energising hydraulic brake thus depends on the ratio between the pressurised piston area in the supporting cylinder $A_{\text{Sup}}$ and the brake piston area $A_{BA}$.

$$C_{\text{SEHB}}^* = \frac{i_L \mu}{2 A_{BA} - \mu}$$

(3)

If the supporting cylinder is connected to the calliper via a joint lever, an additional gear transmission ratio $i_L$ has to be accounted for in Eq. 3. By the factor 2 the fact is considered that the perpendicular force is applied from both sides of the brake disc. For the derivation of the SEHB brake coefficient, $F_{\text{clamp}}$ was assumed to be an additional force on the supporting cylinder. The precondition for self-energisation follows from Eq. 3 and yields:

$$\frac{i_L A_{\text{Sup}}}{2 A_{BA}} \leq \mu$$

(4)

The special feature of the SEHB concept is that by activating the left flow scheme of the proportional brake valve in Fig. 1, self-energisation is turned into self-extinction. As simulation shows in section 5, all is needed is a small solenoid to actuate the valve of the size like those used in automotive antilock brake system.

Fig. 1 contains some features that have not been explained yet. Obviously, without further actuation, the brake is not energised during a stand-still or when the brake is not engaged yet. A spring between actuator and brake pad bracket presses the actuator against the brake disk. When opening the proportional brake valve, the spring initiates the braking operation. The energy to lift the brake actuator away from the brake disk and to set a defined clearance must be provided by a high-pressure accumulator that is charged during regular operation. The low-pressure accumulator depicted in Fig. 1 is necessary as an expansion tank because the brake actuator is a differential cylinder. A differential cylinder takes in more volume when it is extended. Without an expansion tank, cavitation would occur. High-pressure and low-pressure sides are separated by check valves that act as a hydraulic rectifier and allow bidirectional braking. The springs in the chambers of the supporting cylinder allow the retraction of the supporting cylinder to its initial mid position when the brake is lifted and the 2-way switching valve is opened.

### 3 Control strategy of SEHB

The SEHB concept allows the direct control of the actual friction force, independently of friction coefficient changes. Since $\mu$ is influenced by speed, brake pressure and temperature, for conventional brakes it is an unknown parameter. Uncertainty about $\mu$ consequently means uncertainty about the actual friction force $F_{\text{brake}}$ and the retardation torque respectively. The retardation torque, however, is the control variable for vehicle dynamics control systems like the Electronic Stability Programs (ESP). The dynamics of self-reinforcement of SEHB depends on the friction coefficient $\mu$, similar to the drum and wedge brake as can be seen in Eq. 3. But SEHB offers a simple way to measure the actual friction force $F_{\text{brake}}$ via pressure transducers connected to the chambers of the supporting cylinder. Therefor the load pressure in the supporting cylinder can be used as control variable for SEHB in a closed loop control.

Proportional-action controllers offer a suitable solution for conventional valve-controlled closed-loop pressure control systems without disturbances in the form of volume flow
values, [7]. The proportional controller was therefore used in simulative examinations of SEHB. It proportionally controls the valve input corresponding to the deviation of the actual friction force from a given set value, see control scheme in Fig. 5. If the deviation is positive (actual brake force is smaller than required brake force) the control valve is actuated to apply high pressure to the piston face (right flow scheme). If the deviation is negative, it connects the piston face to low pressure side. The valve closes completely when the setpoint is achieved. Further investigations are being carried out at IFAS to adapt the friction force control to the varying operating conditions.

4 Safety features of SEHB

A new brake actuator for railways must comply to or exceed the high safety performance of state-of-the-art air brakes. According to the basic safety policy a brake has to be fail-safe, [8]. In any case of failure of control systems or components a brake system must fall back into a fail-safe mode, to stop the train within a predefined maximum distance. Safety requirements for railway brake actuators also include that the power for braking has to be inexhaustible in all possible scenarios. A passive adaptation of brake force respective wagon load as well as wheel slide protection systems (WSP) are essential to keep braking distances within the limits defined by the railway operator. The maximum braking distance should be as small as possible since it has direct influence on the travel interval between successive trains. SEHB depends on closed loop control, therefore special attention has to be given to fail-safe features that work without electric energy. Fig. 3 shows which safety properties (inexhaustibility, load adaptation, and wheel slide control) have to operate in different braking scenarios. Safety is a property of a brake system as well as of each brake actuator. This section focuses on solutions for a fail safe brake actuator. The safety properties

- inexhaustibility and
- load adaptation

shall be discussed and a solution for SEHB will be presented. Since wheel slide control can be achieved by adapting the friction force set value for the brake, it is not further discussed here.

Inexhaustibility Air brakes have air reservoirs mounted in the wagons supplying power, enough that they are assumed as inexhaustible, considering that air is also delivered from the main brake line. SEHB is inexhaustible by principle since the brake power comes from the train’s motion. There are a few exceptions, however, which need a closer look.

1. **Supporting cylinder in end position.** The brake controls must prevent the critical situation, that the supporting cylinder would reach its end position.

2. **Repeated braking and venting during stop.** Activating the brake during stand still (e.g. for test purposes) releases pressure off the high-pressure accumulator to accelerate the brake initiation.
3. **Leakage.** Long periods of standing on a slope require the cylinders to be absolutely leak proof especially the supporting cylinder that carries the load. Several design requirements result from these critical points. For example the supporting cylinder stroke should be enough to deliver power for any operating condition. Blending between brakes in one bogie is a method to permit the retraction of one actuator while the others are in service. Through proper accumulator design it has to be ensured that there is always enough pressure left to lift the brake pads from the disk. Leakage has to be minimized in the control valve and prevented, especially at the piston sealing in the supporting cylinder and all external sealings. The parking brake has to maintain its braking pressure for long periods of time up to several years. A separate or separable pressure circuit might be helpful as a proof of safety for the parking brake. Seat valves are widely used as safety relevant components in applications where heavy loads have to be hydraulically held e.g. in jacks, cranes, and elevators. A seat valve like the parking valve shown in Fig. 4 significantly reduces the number of safety relevant sealings.

For parking, a specified friction force has to be ensured. According to [9] the minimal brake force of the parking brake in a commuter train has to ensure stand still of the empty train in a slope of up to 4%. In case every brake of a train has a parking brake mode, this results in a relatively small braking force for each brake. A pressure reduction valve can be used to load a parking accumulator with a corresponding parking pressure during regular service. This parking pressure is applied on the actuator for parking.

A differential cylinder produces the clamping force

\[ F_{\text{clamp}} = p_A A_{\text{area of rod}} + (p_A - p_B) A_{\text{area of ring}} \]  

(5)

With both chambers pressurised it still produces a braking force with the advantage that no leakage can occur between the piston and the cylinder boring. The parking valve depicted in Fig. 4 connects the high-pressure line with both actuator chambers. Thus, the number of safety-critical parts is reduced to the external piston rod sealing and the check valves to high and low-pressure feed lines.
Load adaptation  Since the kinetic energy \( E_{\text{kin}} = \frac{1}{2}mv_0^2 \) grows proportional to the translatory and rotary inertia \( m \) of the train, the retardation torque must be load dependent to satisfy required stopping distances. Air brakes match the brake demand signal with the pressure in the air suspension to achieve the required load adaptation. The SEHB load adaptation could be done by the brake controller in terms of signal processing using pressure transducers. Special care would have to be given on the reliability of the sensors and their energy supply.

The concept of SEHB is capable of incorporating a subsidiary hydraulic–mechanic fall-back solution which also allows a secure load adaptation without sensors. Therefore it provides a fail-safe mode in case of total loss of electric power. As can be seen in Fig. 4, the valve spool is balanced between two springs of which the right one can be offset by an actuator. By creating a leftward offset of the right spring, in case of electricity dropout the valve would give way for braking. On the opposite side of the valve spool, high pressure from the supporting cylinder is applied, resulting in closing the valve spool when a specified pressure is reached. The offset of the spring could be designed to depend on the air pressure of the air suspension to achieve the load dependency.

5 Simulation of SEHB

A DSHplus simulation model served the verification of the SEHB concept. Fig. 5 illustrates the layout of the model. It is comprised of a hydraulic section with cylinders, accumulators and valves, and a signal section including the state dependent friction coefficient and the closed loop control.

Friction force calculation  The parameters of the components represent the current development status of the first prototype. The brake force calculation corresponds to a...
Figure 5: Layout of the brake simulation model

fictive train car design that has been agreed on within the research project “EABM” of the German Research Foundation (DFG).

- Maximum speed: \( v_0 = 120 \text{ km/h} \)
- Maximum waggon load: \( m = 13.6 \text{ t} \)
- Two pairs of individual wheels, four disc brakes
- Diameter of wheel (new / old): \( d_{\text{wheel}} = 920 \text{ mm} / 840 \text{ mm} \)

According to [9], taking a maximum stopping distance of 500 m at maximum velocity and a response time of 0.8 s into account, it makes sense to calculate the brake parameters for a maximum deceleration of \( d = 1.2 \frac{\text{m}}{\text{s}^2} \). The maximum retardation force \( F_d \) is then calculated by multiplying the mass inertia per disc brake times deceleration plus a constant force resulting from slope of \( s = 4 \% \) and acceleration of gravity \( g \). The rotary inertia of wheels and drives is included with a factor \( k_r = 1.1 \) in the translatory inertia.

\[
F_d = \frac{m}{4} (k_r d + sg) = 5822 \text{ N} \tag{6}
\]

The maximum friction force \( F_{\text{brake}} \) acting on a friction radius of \( r_l = 245 \text{ mm} \) then yields:

\[
F_{\text{brake}} = F_d \frac{d_{\text{wheel,new}}}{2 \cdot r_l} = 10931 \text{ N} \tag{7}
\]

**Brake design parameters** The supporting cylinder translates the friction force into pressure via a potential joint lever with gear transmission ratio \( i_L = 1.8 \). The supporting cylinder has a piston diameter of \( d_{1\text{Sup}} = 32 \text{ mm} \) and a piston rod diameter of \( d_{2\text{Sup}} = 20 \text{ mm} \). At maximum brake force it produces a pressure of \( p_{\text{max}} = 222 \text{ bar} \). Friction of the supporting cylinder is parametrised with 50 N breakaway force. The size of the actuator follows from the precondition of self-energisation, Eq. 4. The minimum friction coefficient is appointed \( \mu > 0.1 \). From manufacturer catalogue data an appropriate differential thruster with piston diameter of \( d_{1\text{BA}} = 80 \text{ mm} \) and piston rod diameter
of $d_{2BA} = 60 \text{ mm}$ with $40 \text{ mm}$ stroke is chosen. Friction is parametrised with 200 N breakaway force. The spring in the actuator initiates the braking, and compensates for all losses due to friction and pressure difference in the system. Its task is twofold. Firstly it overrides friction $F_{fBA}$ and pressure force $F_{\Delta p} = \frac{\pi}{4}(p_{1BA}d_{1BA}^2 - p_{2BA}(d_{1BA}^2 - d_{2BA}^2))$ in the brake actuator. This is to pull out the actuator piston until clearance is zero. The second task is to compensate friction $F_{f\text{Sup}}$ and spring force $F_{\text{Springsup}}$ in the supporting cylinder which has a suppressive effect on the self-energisation, once the friction contact is achieved. From this requirement follows Eq. 8 to calculate the spring force $F_{\text{Spring}}$.

$$F_{\text{SpringBA}} = F_{\Delta p} + F_{fBA} + \frac{F_{f\text{Sup}} + F_{\text{Springsup}}}{2\mu^2}$$  \hspace{1cm} (8)

For the presented simulation the spring in the actuator applies 1225 N and has a stiffness of $35 \frac{N}{\text{mm}}$. The balancing springs in the actuator have a cumulative stiffness of $4 \frac{N}{\text{mm}^2}$. The brake valve is parametrised as a zero-overlapped 4/3-way control valve with $2 \frac{l}{\text{min}}$ nominal flow at 35 bar and 30 Hz natural frequency. The low value is intended to accentuate the fact that robust, reasonably priced components can be used. The high-pressure accumulator has a storage capacity of 8 ml, enough for retracting the actuator for more than 3.5 mm. The expansion tank has a storage capacity of 141 ml. Fully charged it generates a system pressure of around 5 bar on the low-pressure side. The accumulator is fully charged when the brake piston is completely retracted.

The parameters of the fluid simulate the behaviour of HLP 46 hydraulic fluid. Pressure dependency of the bulk modulus and the influence of contained air is accounted for. The bulk modulus has a significant influence on the initiation performance of the brake, as proved by simulation. The simulation results shown below were yielded for an undisolved air content of 0.1%.

The mechanical stiffness of the brake calliper, brake linings and brake disc is estimated to be $25 \frac{kN}{\text{mm}}$. At the beginning of a simulation a clearance of 0.5 mm is parametrised between brake pads and disc.

**Friction coefficient model** The frictional force is calculated using a characteristic diagram. It was derived in the context of this research project from test data supplied by a manufacturer of brake linings. Based on the conclusion found in various friction related publications [10], that temperature below a critical value of approximately $150 ^\circ\text{C}$ does not have a distinct influence, Fig. 6 shows the friction coefficient trajectory in relation to velocity and pressure used for the simulation.

The friction model facilitates a more realistic simulation. The friction coefficient rises while the vehicle is decelerating. Therefore the brake controller will act to minimize the resulting brake force deviation. Also for the initiation of the brake it gives valuable insight. For very low braking pressures, as they occur in the initiation phase, the friction coefficient and the self-energisation respectively is lower than for higher pressure. This leads to more realistic evaluation of the rise time of the brake.

**Simulation results** The simulation results provide evidence of the dynamic efficiency of the brake. The dynamic performance essentially depends on whether the high-pressure accumulator has been charged by a previous braking operation or not. It also gives insight
into the system’s dead time. This can be defined as the period between brake demand and 10 % achievement of the set value.

The response of the brake can be demonstrated particularly well with reference to a sudden change in the reference input variable. This is not intended to be the simulation of a typical rail vehicle braking operation, which, of course, is not sudden for reasons related to passenger comfort and safety. The achievable brake dynamics plays an important role for wheel slide protection performance and constitutes one of the main advantages of hydraulic systems over pneumatic brakes. Three simulations with sudden changes in the reference input variable give information about the expected dynamic performance of the brake:

1. Initiating braking with maximum braking force $F_{\text{brake}} = 10931 \text{ N}$ with high-pressure accumulator completely discharged, Fig. 7

2. Venting the brake and setting the air clearance, Fig. 8

3. Initiating braking with maximum force with preloaded high-pressure accumulator, Fig. 9

Under worst-case conditions, the high-pressure accumulator is completely empty and cannot make any contribution towards overcoming the air clearance. Fig. 7 shows the result of the above specified simulation for the case when the brake is initiating without pressure in the high-pressure accumulator.

Because of the relatively strong spring in the actuator the clearance is overridden in only 150 ms. After that for a time of about a second nothing much seems to happen. The only significant change takes place in the movement of the supporting cylinder. It moves
4.4 mm while the braking actuator only moves 8 µm without any changes in the high-pressure accumulator. The reason why the self-energisation starts so slowly is because of the compressibility of the fluid which, at low pressures with small contents of unsolved air, is comparatively low. The orifices and friction forces in the system slow down the initiation process. Therefore, in the development of SEHB, special care has to be given on friction and compressibility. A previous plan to use silicon oil as alternative braking fluid was cancelled for the sake of its lower bulk modulus compared to mineral oil. The rise time for 10% of the maximum force of 10931 N is 1.41 s, the rise time for 90% is 1.65 s, which is too slow for a future implementation. Reducing the dead time is very important for reducing stopping distance and has to be further studied.

If the braking force set value suddenly changes to 0 kN the servo-valve opens in the other direction and relaxes the compressed fluid of the brake piston to the expansion tank. The brake actuator releases, as shown in Fig. 8.

During venting the brake the high-pressure line is connected to the surface of the brake actuator piston ring, increasing the relaxing effect. The surface of the brake actuator piston ring is small compared with the surface of the piston face, which means that considerably less volume flow is required for the return stroke. The fluid stored in the high-pressure accumulator is sufficient to lift the brake, as is shown by the fact that the supporting piston does not give way any further. It even moves backward slightly as the previously highly compressed fluid relaxes. The proportional-action controller is not capable of cancelling out the brake force completely. The spring power in the brake actuator acts as a disturbance and, where a proportional-action controller is used, the principle is such that a permanent system deviation remains. The final static value is reached after around 0.5 s with 125 N supporting force. At this moment \( t = 5.3 \) s, a control pulse opens the servo-valve negatively again for controlled lifting of the actuator away from the brake disk and to set a predetermined air clearance.

**Figure 7:** Simulation of braking initiating with maximum braking force \( F_{\text{brake}} = 10931 \) N with high-pressure accumulator completely discharged
Figure 8: Simulation of venting the brake and setting the air clearance

Under normal operating conditions, it may be assumed that the high-pressure accumulator is still preloaded from a previous braking operation. Fig. 9 shows the way in which the response time improves when the high-pressure accumulator is preloaded.

Figure 9: Simulation of braking initiation with maximum force with preloaded high-pressure accumulator

The application spring in the brake actuator is now assisted by the high-pressure accumulator, so that the brake linings are already being pressed against the brake disk after 84 ms.
Compared with the characteristic shown in Fig. 7, the self-energizing process ignites with a much steeper initial gradient. It has reached 10% of the target value of 10.931 kN already after 165 ms and 90% after 336 ms. The total period, including the dead time required to overcome the air clearance, amounts to 420 ms. This means that the time for reaching 10% of the target value has been reduced by 88% and for 90% target value by 74%.

6 Prototype design

Current efforts focus on the implementation of the self-energising electro-hydraulic brake. After having analysed the principle and its dynamics in simulation it is a vital matter to verify the theoretic results by laboratory tests. For this purpose a down-scaled prototype is being designed on the basis of a automotive brake disc driven by a hydrostatic velocity controlled drive in connection with a flywheel. The goal is to prove the hydraulic design which is independent of the size of the brake disc. With the experience gained from experiments with the first prototype which are scheduled for April 2007, a second prototype will be designed for testing on a railway roller dynamometer. Fig. 10 shows the assembly of the first SEHB prototype.

![Figure 10: Design of the first testing prototype of self-energising electro-hydraulic brake](image)

The brake and the brake disk are mounted on two separate aligning shafts. The supporting cylinder is connected to the brake calliper via a lever with a slotted hole that allows adjustment of the transmission ratio. The brake calliper is from an original car brake since the SEHB principle does not necessarily need a double acting differential thruster as brake actuator. Fig. 11 shows a sectional view cut through the calliper.

A special feature of this arrangement is that the shaft which the brake is mounted on
experiences twice the braking force, because both brake calliper and supporting cylinder, conduct the same force into the lever if the transmission ratio $i_L$ is 1. This setup was chosen because it causes only few flexural stresses and yet is very compact. Efforts to put the supporting cylinder in tangential alignment to the friction radius result in a bigger design. It is also important to note that to connect the supporting cylinder on a radius larger than the friction radius ($i_L > 1$) necessitates a smaller supporting cylinder or a larger brake actuator respectively, as can be seen from Eq. 4. Since the piston area of the automotive brake could not be arbitrarily changed, it was important in this case to reach $i_L = 1$.

7 Conclusion and Outlook

The principle of a Self-energising Electro-Hydraulic Brake for a railway application was introduced and discussed concerning some specific railway related safety issues. A more comprehensive safety concept will be addressed in future work. The control concept has been explained. Currently, only a proportional-acting controller has been applied in simulative studies. Further studies are being done to analyse the dynamic interaction in theory and enhance the closed-loop performance using an adaptive controller. We will also pursue a control using switching valves in analogy to ESP and ABS systems. The dynamic performance has been shown in simulation. In the actual simulation phase the brake shows good performance during braking, anticipating good braking performance in applications using wheel slide control, but also shows shortcomings in reaction times for the case when the brake has to override clearance. A better choice of spring stiffness for the springs in the cylinders, friction force minimization, and optimization of accumulators will be further investigated. The assembly of the first prototype design has been presented. The results from measurements scheduled April 2007 will be used to improve the simulation model and thus serve for a deeper understanding of the system. On this basis a second prototype will be designed for testing on a railway roller dynamometer. The authors want to express their thank for the support by the German Research foundation.
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