Self-energizing hydraulic brakes for rail vehicles

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1 Introduction

A self-energizing electrohydraulic brake system is being developed at the Institute of Fluid Power Drives and Controls (IFAS) at RWTH Aachen University within the framework of a DFG (German Research Committee) research project. The Institute of Automatic Control (IRT), the Institute of Rail Vehicles and Materials Handling Technology (IFS) and the Institute of Power Electrics and Electrical Devices (ISEA) are also involved in this research project, which aims to develop a compact traction and braking module for an individual wheel on a rail vehicle. It has therefore been given the name "intelligent, integrated, independent wheel traction/braking module" or EABM (German: Einzelrad-Antriebs-Brems-Modul).

The distribution of traction and braking systems in a vehicle is such that their integration has posed a major challenge in the design of a rail vehicle up to now. The components required by the pneumatic braking systems include pressure reservoirs and switchboards, which frequently have to be fitted in the wagon because of the lack of space available in the undercarriage /Gra99/. The electronic traction power system is also separated from the motor, which is connected to the wheel set directly or via a transmission according to the traction concept. The number of interfaces to be taken into consideration includes the lines that carry power to the motor, as well as the pneumatic supply lines for the brakes. This poses a problem as far as the
design of the undercarriage is concerned, as these components are usually developed by different departments. The purpose of this project is therefore to develop an integrated traction and braking module that requires as few mechanical, data and electrical power interfaces as possible. Figure 1 illustrates the subject of the research project in the form of a sketch. The independent wheel module outlined in the lower part of the sketch is fitted in a closed-loop control circuit that realises wheel slip and non-skid braking, as well as lateral guidance.

Figure 1: Intelligent, integrated, independent wheel traction/braking module (EABM)

Unlike the wheel sets that are usually used today, comprising two wheels securely joined together by a shaft, the developed module is intended to drive and brake just one wheel. This objective gives rise to requirements which cannot be met by a conventional pneumatic solution purely by virtue of the limited space available. Although hydraulic systems offer the significant advantage of higher force density/Kip95/, this is offset by the disadvantage of an environmentally harmful fluid medium. A central pressure supply to the undercarriage is out of the question for safety reasons and because of new problems that would arise with respect to interfaces to the undercarriage. The IFAS is therefore endeavouring to develop a completely new hydraulic brake concept that only requires an electric control interface to the outside with a low power level, whereby the energy needed to apply the brakes should be generated by the braking process itself. The principle of this brake and the current development status is being published for the first time in this article.

Utilising self-energizing effects in brake systems
Efforts to achieve greater braking comfort in spite of increasing vehicle weight have given rise to many approaches to servo-assisted or self-energizing braking concepts over the last hundred years. The most commonly used energized brake system is the drum brake. The brake pads pressing against a brake drum from the inside intensifies its pressing force in the leading direction of rotation.

There are also numerous self-energizing concepts for the disk brake system, the most well known being the wedge principle. Attention is again being focused on this concept following the development of the electromechanically controlled wedge brake and successful trials in cars /Aut06/. The self-energizing principle of the drum brake and the wedge principle are compared with one another in Figure 2.

![Figure 2: Comparison of the self-energizing principles of drum and wedge brakes](image)

The clamping force $F_{\text{clamp}}$ is introduced into the friction contact perpendicular to the direction of motion in the drum brake. The tangential frictional force is not borne directly, but via a lever in the bogie, so that torque builds up around the bearing point, increasing the pressing force of the brake pad. As far as the wedge brake is concerned, the clamping force acts tangential to the direction of motion on a wedge. A perpendicular force is introduced into the friction contact via the wedge level that supports the wedge. The resulting tangential frictional force acts in the direction of the wedge’s movement, intensifying the braking effect. Brake coefficient $C^*$ - the ratio between introduced clamping force and tangential frictional force - is a measure of
the intensification. The values for the drum brake and wedge brake systems shown in Figure 2 are given by $C_{\text{drum}}^* = \frac{\mu \cdot \tan \alpha}{\tan \alpha - \mu}$ and $C_{\text{wedge}}^* = \frac{\mu}{\tan \alpha - \mu}$. Drum brakes have a C* factor of between 1.5 and 20 according to the design /Bre04/. The wedge angle of an electromechanically controlled wedge brake is designed in such a way that the denominator of the gain factor approaches zero within the range of usual friction values. The gain becomes infinite and the brake therefore energizes itself. This means that the actuator of a self-energizing brake has to be capable of adjusting the wedge in both directions as the wedge can be drawn into the friction contact on its own. This type of brake is unstable without a closed-loop mechatronic control system. One of the challenges to be overcome with respect to the automatic brake control system for the electromechanical wedge brake is that the friction locking factor $\mu$ can only be estimated, it cannot be measured.

2 Hydraulic self-energizing principle

The concept of the hydraulic self-energizing brake described in this article, which operates according to the principle shown in Figure 5, utilizes the supporting force that is conducted into the undercarriage by the brake to generate hydraulic pressure. The special aspect of this concept is that the pressure can then be used to either release or apply the hydraulic brake by means of an electrically driven servo-valve. One of the most important features of the brake is that it is activated at a low power level via just one electrical control interface, which is capable of supplying power to the servo-valve and the necessary electronic closed-loop control system. There is no need for an external pressure supply. An exchange of hydraulic pressure is only required locally at the brake actuator. The functional concept is described in greater detail below on the basis of the fundamental hydrostatic principle.

The hydrostatic principle establishes a relationship between the pressure of a fluid and the force exerted by a piston via the piston’s surface. The selection of an area ratio between supporting piston $A_{\text{AS}}$ and brake actuator $A_B$ of less than 1 intensifies the pedal force acting on the brake lining by factor $\frac{A_B}{A_{\text{AS}}}$, refer to Figure 3.
Figure 3: Using the hydrostatic principle to apply the hydraulic brake

The perpendicular force $F_N$ introduced into the friction contact generates a tangential brake force $F_R$ via the friction coefficient. The hatching on the brake piston indicates that this brake force is supported in the undercarriage as the brake piston would otherwise move with the brake disk. The force can be converted into pressure by a hydraulic piston and used to further intensify the perpendicular brake force. This hydraulic self-energizing effect is a known phenomenon and is being utilized in some brake systems. A system like this can be characterized as being a mechanically operated, hydraulically self-energizing brake. However, the considerations outlined below imply that it can also be achieved without any external force (pedal force, electromechanical actuator, air pressure etc.) at all.

The idea is to use the supporting force of the brake against the undercarriage $F_{AS}$ as the sole source of brake power. This initially presupposes that the brake is engaging and that there is a supporting force available. The other case can be examined at a later stage. Figure 4 shows a drawing of such a system. The brake piston with surface $A_B$ is accommodated in a brake caliper (hatched) together with a supporting piston with surface $A_{AS}$. As indicated in the drawing, there is a guide which allows the caliper to be moved relative to the undercarriage. If a tangential frictional force acts on the brake lining, the piston of the supporting cylinder supports the lining on the fluid column shaded red, causing the pressure to rise. The pressure rise acts on the brake piston and in doing so influences the perpendicular force introduced into the friction contact. It is easy to imagine that a self-energizing effect can be achieved by choosing a suitable ratio between the surfaces of the supporting cylinder and brake actuator. This is illustrated by the concept described below:
A perpendicular force $F_N$ introduced into the friction contact at a specified time $t$ uses coefficient of friction $\mu$ to generate frictional force $F_R$, which has to be supported in the undercarriage. Frictional force $F_R$ and supporting force $F_{AS}$ are in equilibrium, as shown in Figure 4. The piston of the supporting cylinder is used to generate a pressure difference $\Delta p_{AS} = p_{HD} - p_N = \frac{F_R}{A_{AS}}$ in the closed hydraulic circuit and this is transferred to the brake piston by means of the hydraulic connection between supporting cylinder and brake cylinder and closes a circuit of gain or attenuation. At time $t + \Delta t$, the original perpendicular force $F_N$ has risen or dropped to $F_N' = \Delta p_{AS} \cdot A_B = F_N \cdot \mu \frac{A_B}{A_{AS}}$. This means that the originally applied clamping force $F_N$ leads to a changed clamping force $F_N'$ that is either intensified ($V_k > 1$) or attenuated ($V_k < 1$) by gain factor $V_k = \frac{A_B}{A_{AS}} \cdot \mu$. The rate of intensification or attenuation depends on the value of the gain factor, but is also limited by the hydraulic time constant.

**Electrohydraulically self-energizing brake**

In order to use the self-energizing principle in a brake system, it must be possible to regulate this unstable process by means of external intervention. There are various conceivable ways of doing this. As is evident from a study of the gain equation, one
of these involves impressing a controllable load pressure on the supporting cylinder by applying an external force to the piston of the supporting cylinder. Assuming that the brake had a gain factor in the order of 1, then the applied brake force would just about neither intensify nor attenuate. The additional actuator system is used to deliberately upset this balance in one direction or the other. The result is a servo-energized brake.

The considerably more sophisticated version involves using a servo-valve to control the brake pressure mechatronically. Figure 5 shows the principle of the electrohydraulically self-energizing brake, which is described in closer detail below. The kinematic principle of the supporting piston is the other way round here. Unlike Figure 4, in which the piston of the supporting cylinder is firmly anchored, the cylinder is connected to the undercarriage in this case.

![Figure 5: Schematic diagram of the hydraulically self-energizing brake](image)

As soon as a braking operation is initiated, pressure builds up in the supporting cylinder as this is used to support the retarding torque in the undercarriage. The pressure is released to the high-pressure side of the brake via the check valve. The pressure may be routed to the face of the piston or to the ring surface via the 4/3-way
servo-valve and used to apply or release the brake. The brake pressure build-up and release process is unstable and must be controlled automatically. A closed-loop control system is capable of stabilizing the process satisfactorily as verified by the recorded simulation results given in this article. Let us first describe the functions of the springs and accumulators shown in the drawing to make the brake construction easier to understand.

**Restoring springs to initialize the supporting cylinder**

The volume flow required to build up pressure in the brake actuator and to fill the high-pressure accumulator is output by the supporting cylinder during the braking operation. The inward movement of the piston rod is limited by the stroke of the supporting cylinder. This means that the supporting cylinder has to regenerate between the braking operation by the piston being returned to its central position by self-centring springs.

**Igniting the self-energizing process**

Another spring is needed to "ignite" the self-energizing process. An energy accumulator must provide the energy required to press the actuator against the brake disk in order to initiate a braking operation. This is a prerequisite for the effectiveness of the self-energizing principle. The energy may be made available by a stored-energy spring mechanism. The self-energizing process should be ignited automatically in the interests of safety, whereby automatic ignition means that the brakes should operate automatically if the energy drops below a certain level. In pneumatic brake systems, a drop in brake pressure causes a spring force to act on the brake actuator. Analogous to this, the spring between the brake cylinder and the brake lining in this system ensures automatic application of the brake lining and therefore leads to ignition of the self-energizing process. The servo-valve must be opened on the clamping side for this. This can be realized by means of an appropriate fail-safe valve setting even in the event of a failure in the electronic system.

**Extinguishing the self-energizing process**
The closed-loop power control system can only cancel out the brake force completely in theory. After all, it only works when the brake is engaged and is therefore generating a supporting force. Analogous to "ignition", lifting the braking actuator away from the brake disk may be referred to as "extinguishing" the self-energizing process.

The actuating energy required to extinguish the self-energizing process and set a defined air clearance must be provided by an energy accumulator. The energy is needed to lift the braking actuator away from the brake disk against the spring force that is required to ignite the self-energizing process. It may be provided by a hydraulic accumulator on the high-pressure side. A control pulse to the servo-valve causes its pressure to be routed to the surface of the piston ring and the brake piston is retracted. One important factor to be taken into consideration when designing and dimensioning the accumulator is that it must be capable of discharging sufficient volume at the pertinent pressure. The volume can only be discharged if the pressure on the high-pressure side $p_{HD}$ remains higher than that on the low-pressure side $p_{ND}$ divided by area ratio $\alpha$ of the brake actuator: $p_{HD} < \frac{p_{ND}}{\alpha}$. The high-pressure line is pressurized during the braking operation that always precedes the brake release process, however, which means that the high-pressure accumulator is being charged. A combination of normal and pilot-controlled check valves, which allows the accumulator to be charged at any time but only permits discharging as needed, may be fitted upstream of the piston accumulator to prevent the piston accumulator discharging during a braking operation at low brake pressure. Once the air clearance has been set, the servo-valve is closed and the brake actuator remains in its position.

**Low-pressure accumulator used as an expansion tank**

As shown in Figure 5, the brake actuator is a differential thruster. The difference in piston surface area is such that it takes in more volume when the piston is extended on the piston face side than it discharges to the surface of the piston ring. In an enclosed system without accumulator and tank, this would lead to a situation in which a vacuum is produced in the low-pressure zone when the brake is applied (extended) because more volume is extracted via the supporting cylinder than is returned via the brake actuator. An expansion tank is required in order to prevent this situation
occurring. The pressure applied to the expansion tank should be low, just sufficient to prevent the build-up of a vacuum with the associated outgassing of the pressure medium, /Mur05/.

**Closed-loop control: braking torque instead of perpendicular force**

As already explained above, the self-energizing brake force build-up process is unstable and needs to be controlled. However, instead of adjusting the difference in pressure in the brake actuator and with it the clamping force, it is more expedient to regulate the difference in pressure in the supporting cylinder. Disregarding friction and inertia, this is directly related to the tangential brake force, i.e. the currently effective retardation torque. The current retardation torque of the wheels is an unknown factor in a conventional brake system. The coefficient of friction is subject to fluctuations and is determined by temperature and speed, as well as weather conditions. These uncertainties are such that the effective brake retardation can only be estimated for a conventional brake system, even if the brake pressure / clamping force is known. The supporting torque of the brake can be calculated by measuring the pressure values in the supporting cylinder and by means of subtraction. With the servo-valve controlled in an appropriate manner, the closed-loop control system is required to follow the specified retardation torque as well as possible.

As far as the systematic approach is concerned, the closed-loop control function described above may be regarded as being a pressure control system with variable supply pressure. 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 /Mur02/. The controller may be supplemented by a derivative-action component in order to exert additional influence on the system's attenuation. A proportional-action controller should be used for initial simulation-assisted experiments, however. Although the infeed movement of the brake piston constitutes a small disturbance for a brief period, this can be regarded as being negligible initially. The changing supply pressure corresponds to a closed-loop gain that depends on the operating point. The closed-loop control system can be optimized by adapting the controller gain factor to the system pressure.

**Simulation model**
A simple simulation model was built in order to verify the variability of the self-energizing brake system. We used the DSHplus simulation software for this, a program that is particularly suitable for the simulation of hydraulic systems, also refer to /fluidon/. It offers a means of modelling and parameterizing hydraulic systems by connecting standardized hydraulic components via a graphic user interface. Figure 6 illustrates the layout of the model. It comprises a hydraulic section with blue connecting lines, a mechanical section shown in grey and a green closed-loop control section, which are described briefly below.

![Diagram of the brake simulation model](image)

**Figure 6:** Layout of the brake simulation model in DSHplus

The brake actuator is a differential thruster with 82.2 mm piston diameter, 60 mm piston rod diameter and 30 mm stroke. It is pretensioned in the run-out direction by a spring on the piston face side. The spring exerts forces of 400 N with the piston in the extended position. The brake actuator is a zero-overlapped 4/3-way control valve with a 2 l/min. nominal volume flow connected. It has been parameterized to react relatively slowly with a natural electromechanical frequency of 30 Hz. The low value is intended to accentuate the fact that robust, reasonably priced actuators can be used for the closed-loop control system. The high-pressure and low-pressure zones are on the other side of the valve. Check valves protect the high-pressure zone from the supporting piston, ensuring that the chamber with the highest pressure is always opened to the high-pressure side. The high-pressure accumulator is a spring-preloaded piston accumulator with 8 ml storage capacity. The supporting cylinder has
a piston diameter of 30 mm, 15 mm rod diameter and ± 100 mm stroke. The area ratio compared with the face of the piston amounts to 1 to 10, which means that the gain factor $\frac{A_{BK}}{A_{AS}} \mu$ for friction coefficients $\mu \geq 0.1$ fulfils the condition for self-energizing. The restoring springs lock the supporting piston in the middle position with 100 N and have a spring stiffness of 2 N/mm. The expansion tank with 35 ml storage capacity is connected to the low-pressure side. Fully charged, the accumulator generates a system pressure of around 3 bar on the low-pressure side. The accumulator is fully charged when the brake piston is completely retracted.

The stiffness of the brake caliper is modelled in the mechanical section. The brake caliper, the brake linings and the brake disk are not ideally stiff and are represented by a spring with play. The resulting stiffness has been parameterized as 50 kN/mm, while the play corresponds to the air clearance and amounts to approx. 0.5 mm. The frictional force is calculated from the perpendicular force of the actuator, using the proportional friction coefficient relation and assuming a constant coefficient of friction of $\mu = 0.4$, and this is applied to the supporting piston as an external force.

The closed-loop control section compares the difference in pressure at the supporting piston with a predetermined specified frictional force value. The closed-loop controller, a simple proportional-action transfer element in this case, passes the system deviation to the valve as the manipulated variable.

**Simulation results**

The simulation results presented in this article provide evidence of the dynamic efficiency of the brake in the controlled state. In this respect, the dynamic performance essentially depends on whether the high-pressure accumulator has been charged by a previous braking operation or not. It also gives an insight into the system's dead time. This is the period between brake operation and the first reaction of the brake, which is required to overcome the air clearance.

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 is not sudden for reasons related to passenger comfort and safety, of course. The achievable dynamic
performance of the brake plays an important role in terms of non-skid braking in spite of jerk limitation and constitutes one of the main advantages of hydraulic systems over pneumatic brakes. The exactness with which the non-skid braking system operates improves with the speed of the brake's response to brake signal corrections, shortening the braking distance without producing flat areas on the wheels. Three simulations with sudden changes in the reference input variable give information about the expected dynamic performance of the brake:

1. Braking with maximum force with high-pressure accumulator completely discharged
2. Venting the brake and setting the air clearance
3. Braking with maximum force and with preloaded high-pressure accumulator

The reference input variable is the specified value for brake force $F_{\text{brake}}$, which is converted into a pressure difference by the supporting cylinder.

Assuming that each wheel is subjected to a load of 9 t and the train is retarded by 1 m/s\(^2\), the radius ratio between friction ring and wheel of 247/460 gives rise to a supporting force of 18,437 kN on an even surface, allowing for the rotating inertia with a factor of 1.1, refer to Figure 7. This supporting force may be regarded as being the top limit for the brake force.

**Braking in relaxed state**
Under worst-case conditions, the high-pressure accumulator is completely empty and cannot make any contribution towards overcoming the air clearance. The spring in the brake actuator is parameterized to be relatively weak, which means that it cannot apply the brake as quickly. It is not only acting against the inertia of the actuator, but also against that of the supporting piston, the pressure in the low-pressure accumulator and the hydraulic resistors (servo-valve and check valves). Figure 8 shows the time characteristic of the specified and actual variables, as well as the movement of the brake actuator and the supporting piston.

**Figure 8:** Simulation of the relaxed brake in the event of a sudden change in the reference input variable

The braking operation is divided into two sections: the brake piston first overcomes the air clearance of 0.5 mm. It is evident that the supporting piston is also moved with the actual brake actuator. The valve is open to the maximum extent because of the large difference between specified and actual values. The friction lining makes contact with the disk after half a second and the self-energizing process is ignited. Initially weak, the frictional force is intensified exponentially so that the valve enters the control range relatively quickly and the frictional force of **18.437 kN** is finally
established. It takes 0.43 s to reach the required brake force once the self-energizing process has ignited. This means that the total control response time amounts to 0.93 s. As we will show below, this time can be reduced substantially if the high-pressure accumulator is preloaded. The supporting piston has given way by a total of 36.6 mm during the control response time. The delivered volume is used to compress the fluid volume (50.2 %), charge the high-pressure accumulator (10.7 %) and move the brake actuator (39.1 %). These figures illustrate the extensive influence exerted by the so-called equivalent bulk modulus of the pressure medium, which accounts for the elasticity of the cylinder and line walls. This value is much poorer where hoses are used, which is why hoses should not be used for the brake system.

**Venting/releasing the brake**

If the specified braking value suddenly changes to 0 kN supporting force, the servo-valve opens in the other direction and relaxes the compressed fluid of the brake piston in the low-pressure zone. The brake piston retracts, as shown in Figure 9.

**Figure 9:** Simulation of brake venting
The high-pressure zone is connected to the surface of the brake actuator piston ring at the same time, 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 36 N supporting force. At this moment (t = 5.5 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. In the final brake release phase, the supporting piston is returned to its initial position by its centring springs. The pilot valve on the supporting piston is opened for a limited period for this. After a little more than half a second, the brake is back in its initial position and the high-pressure accumulator is still charged, ready for the next braking operation. It goes without saying that the supporting piston should be designed in such a way as to enable another braking operation immediately, even without the return movement.

**Braking with preloaded high-pressure accumulator**

Under normal operating conditions, it may be assumed that the high-pressure accumulator is still preloaded from a previous braking operation. **Figure 10** shows the way in which the response time improves when the high-pressure accumulator is preloaded.
Figure 10: Simulation of the preloaded brake in response to a sudden change in reference input variable

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 66 ms. Compared with the characteristic shown in Figure 8, the self-energizing process ignites with a much steeper initial gradient and has already reached the target value of 18.437 kN after 0.33 s. The total period, including the dead time required to overcome the air clearance, amounts to 0.396 s. This means that the dead time has been reduced by 86 % and the clamping time by 23 %. The supporting piston then only gives way by 27.5 mm. Its delivery volume is divided between the high-pressure accumulator (10.1 %), the stroke of the brake actuator (51.5 %) and compression of the fluid volume (38.4 %). It is clearly evident that the supporting piston is required to deliver less volume and the consumed volume can be better used if the brake is preloaded.

Summary and prospects

The simulation results provide evidence of the ease with which the self-energizing hydraulic brake can be controlled using a proportional-action controller. They draw
attention to the dynamic power potential, which can be improved still further by means of adaptive control concepts and a higher-level open-loop brake control system. The achieved time constants are significantly smaller than those of the pneumatic brake systems used up to now. This offers a means of improving anti-skid braking considerably. The simulation results also provide information regarding the anticipated strokes of the supporting cylinder, which must be designed and dimensioned in such a way as to enable several braking operations without return strokes in the middle position.

For further research within the framework of the project sponsored by the German Research Committee (DFG), the model will be supplemented by a simulation of vehicle inertia and a dynamic friction contact in order to study the dynamic effects brought about by the additional degree of freedom provided by the guided brake mounting. The simulation model will also be used to develop improved closed-loop control concepts.

Current plans include building a model test rig for a smaller brake disk and reduced braking energy at the IFAS to enable verification of the presented concept. It is intended to incorporate the findings and experience gained with the model test rig into the design and construction of an initial prototype, which is to be tested on the rail vehicle roller dynamometer on the premises of the Institute of Rail Vehicles and Materials Handling Technology (IFS), one of the partner institutes working on the DFG project, at the end of 2007.

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