DESIGN OF THE SELF-ENERGISING ELECTRO-HYDRAULIC BRAKE (SEHB)*

Dipl.-Ing. Julian Ewald (1), Dr.-Ing. Matthias Liermann (2), Univ.-Prof. Dr.-Ing. H. Murrenhoff (1)

(1): RWTH Aachen University, Institute for Fluid Power Drives and Controls

(2): American University of Beirut, Lebanon

1   Introduction

The Self energising Electro-Hydraulic Brake (SEHB) is developed at the Institute for Fluid Power Drives and Controls of RWTH Aachen University. The idea was developed during the research project "Intelligent, Integrated Single-Wheel Traction and Brake Module (EABM) /Her08/ and was awarded with the science award of North Rhine-Westphalia /AN07/. In spite of it being originally designed for use in railway vehicles it is also adequate for motor vehicles and industrial applications. This article presents the basics about the dimensioning and design of the SEHB. It also shows the conflicts due to contradictory optimisation aims and how they can be solved by compromises. Moreover, a test facility and test results are presented.

1.1   Hydraulic Brakes

Until the 1930s, in some vehicles even until the 1960s, mechanical brakes were used. Bowden cables, leverages and redirections transferred the force from the brake pedal to the wheel brakes. A disadvantage was the high amount of maintenance. Furthermore the irregular force transmission to the brake callipers caused uneven wear /Bre04/. The first hydraulic wheel brake actuator was patented in 1917 by Malcolm Loughead, a civil engineer. This invention started the triumph of hydraulic brakes as it doubled the brakes' efficiency. In the beginning drum brakes represented the state of the art.

Their main advantage is the capability of implementing a mechanical boosting of the brake force. Consequently the activating force can be kept on a low level.

The drum brake has disadvantages because of insufficient cooling, and other disadvantages such as difficult dosability, friction variation and squeaking sounds. When higher brake power was needed, the use of disc brakes became common. However, in the beginning disc brakes with a booster were not available. The necessary actuating force for disk brakes was very high compared to drum brakes. Not until the invention of the pneumatic brake booster, disc brakes succeeded on the market. The vacuum reinforcement provided for high brake power, easy dosability and a low pedal force. A central brake booster between brake pedal and main brake cylinder replaced the local intensification at each drum brake. Different reinforcement factors at the wheel brakes were prevented, allowing an equal distribution of brake forces. Soon ideas for self energising disc brakes were developed /Tou64/, but they could not compete with brake booster systems.

1.2   Mechatronical Brake Systems

The requirements for brake systems of motor and railway vehicles are becoming more difficult to fulfil. For both types of vehicles high brake power, low energy demand, good controllability and the ability of brake force feedback are requested.

In addition, there are specific requirements such as a flexible brake management or a distributed brake system with interfaces for railway applications. Pneumatic brake systems or brake systems with vacuum boosters cannot meet these requirements or can only fulfil them with high effort. This is why the development of mechatronical brake systems is promoted. The wedge brake is an example for a highly developed brake system that works in the region of critical self energisation. The central element of the wedge brake is the wedge bearing. It is located between the moving brake pad and the brake calliper. It allows for the conversion from friction force to clamping force. A control unit influences the position of the wedge by an electric motor with a spindle system /Gom06/.

1.3 Working principle of the SEHB

The principal idea of a self energising brake is to use the kinetic energy of the vehicle to clamp the brake shoes towards the brake disk. The brake calliper is moveable in the direction of the friction force. The brake torque is supported by an hydraulic cylinder. In contrast to other existing brake systems, the SEHB is controlled in the postcritical reinforcement region /Lie08/. Depending on the driving direction one of the two supporting cylinders is pressed in. -**figure11** shows the SEHB in top and side view. The functionality of the SEHB will be explained by describing a brake cycle. Initially the brake is released with a gap between the disc and the brake pads. The valves NC V3 and NO V4 are closed, so that a hydraulic force holds back the prestressed spring, integrated into the brake actuator (BA). As soon as the valve NO V4 opens, chamber B of the brake actuator is connected to low pressure.

The spring pushes the brake actuator out and oil is drawn through the check valve into chamber A. The spring force presses the brake pads against the brake disc, building up a low friction force. This force is transferred to the vehicle by one of the supporting cylinders. The other cylinder stays in its position allowing a pressure build-up in both directly connected cylinders. The configuration with two plunger cylinders is considerably simpler than preceding systems /Lie06/.

The hydraulic self energisation is due to the connection of the supporting cylinders with the brake actuator. The oil flows through the high pressure check valve HPCV and the NO V1 valve. The process of self energisation is interrupted as soon as the NO V1 valve closes. The SEHB always needs to be controlled, because of the postcritical self energisation. If it is not controlled, the brake clamps until the pressure limiting valve opens.

The pressure p_{AZ} in the supporting cylinder is the controlled parameter. If the dimensions are known, the retarding force can be directly calculated with this pressure. If the mass of the vehicle is known, the deceleration can be calculated as well.

The four shift valves NO V1, NC V2, NC V3 and NO V4 connect the two chambers of the brake actuator either with high or low pressure. By choosing normally open (NO) and normally closed (NC) valves, a fail-closed behaviour is guaranteed, so that the brake clamps in case of a failure of the electric power supply.
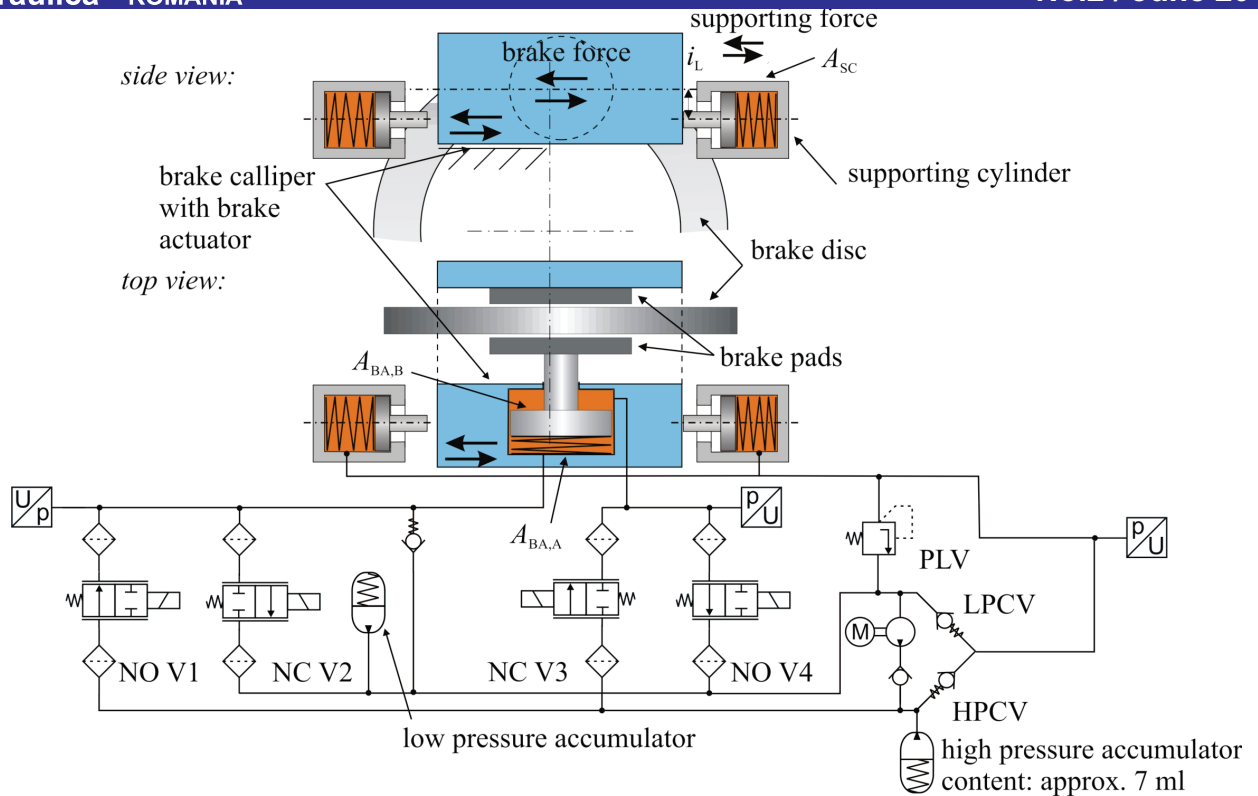
figure 1-1:Working principle of the SEHB with simplified support

The low pressure is kept on a low value by an accumulator. Thus, a higher stiffness of the fluid is achieved /Mur08/. At the same time cavitation in the suction part is prevented.

The high pressure accumulator is needed to release the SEHB completely against the spring force of the brake actuator. While braking, the accumulator is charged automatically as soon as the pressure in the high pressure part rises above the loading pressure of the accumulator. To release the brake, the accumulator's volume of a few millilitres is led into chamber B so that the brake opens completely against the spring force. A gap between the disc and the brake pads is set. If the vehicle is parked and the high pressure accumulator is empty, the vehicle has to be started up against a low braking force from the preloaded spring. Another possibility is to release the brake by the pump shown in

- figure1-1.

2   Basics of dimensioning and design

In order to dimension the diameter of the brake actuator (BA) and of the supporting cylinder (SC) the path of self energisation is considered. By choosing the diameters it is ensured that for all friction coefficients which can occur in normal operation, the pressure in the supporting cylinder is always higher than in the brake actuator. Thus, the brake can energise itself when valve NO V1 is open. From this consideration the simplified condition for the relation of the hydraulic surfaces can be deduced, as presented in /Lie06/.

$$A_{BA,A} > \frac{1}{2 \cdot \mu \cdot i} \cdot A_{SC} \qquad \text{eq. 2-1}$$

$$\mu_{min} = \frac{1}{2 \cdot i} \cdot \frac{A_{SC}}{A_{BA,A}} \qquad \text{eq. 2-2}$$

Certain effects, which restrain the self energisation have not been considered in this calculation. These effects are: friction, spring forces and flow losses at the valves.

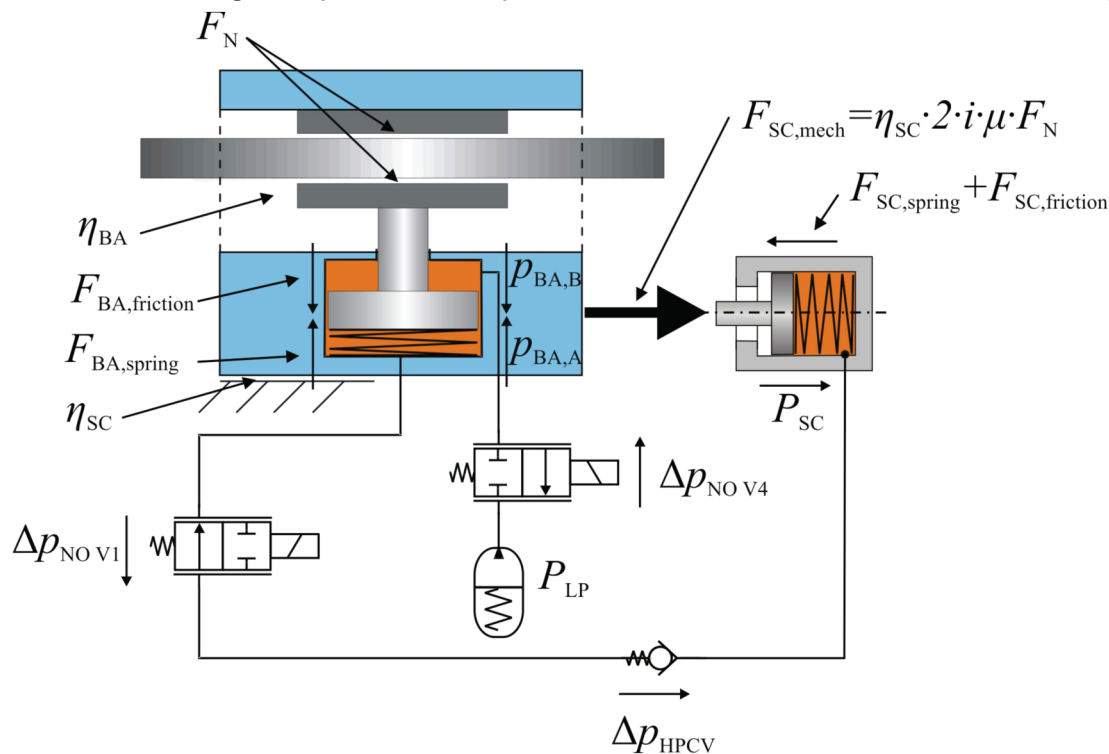Therefore, in the following, - eq.21 will be expanded. The used variables are defined in figure 2-1.

figure 2-1:Definition of hydraulic and mechanic variables

The pressure losses at the high pressure check valve, consisting of the opening pressure and of a part depending on the volume flow, are considered together as the pressure Δp_{HPCV}. The volume flow through the high pressure check valve also flows through the valve NO V1 causing another pressure loss Δp_{NOV1}. The pressure losses depend on the operating point, the surface of the brake actuator $A_{BA,A}$ and on the compliance of the brake calliper.

$$p_{SC} = p_{BA,A} + \Delta p_{HPCV} + \Delta p_{NOV1} \qquad \text{eq. 2-3}$$

The mechanical force at the supporting cylinder $F_{SC,mech}$ is balanced to the hydraulic force, the friction forces and to the spring force which presses the piston back to its initial position.

$$F_{SC,mech} = F_{SC,hydr} + F_{SC,Reib} + F_{SC,Feder} =$$
$$= p_{SC} \cdot A_{SC} + F_{SC,friction} + F_{SC,spring} \qquad \text{eq. 2-4}$$

The force at the supporting cylinder is also balanced to the friction force actuating in the friction radius. The friction force is calculated with the normal force per brake pad F_N, the friction parameter between brake disc and brake pad μ, the number of friction surfaces (2) and the transmission i. Mass forces are neglected. The conduction of the brake force to the supporting force is considered with an efficiency factor.

$$F_{SC,mech} = \eta_{SC} \cdot 2 \cdot i \cdot \mu \cdot F_N \qquad \text{eq. 2-5}$$

The transmission *i* is defined as the quotient between friction radius and the distance between the centre of the friction radius and the supporting cylinder's force application point.

$$i = \frac{r_B}{r_{SC}} \qquad \text{eq. 2-6}$$

The force applied to the brake pad depends on the efficiency of the brake calliper η_{BA}. The efficiency factor describes the quotient of the normal force at the friction pad and the force at the brake actuator. This force results from the hydraulic force in chamber A and the spring force reduced by friction forces and by the hydraulic force in chamber B. The hydraulic chamber B can be used for control or can be connected to low pressure by the valve NO V4. In this case, the pressure in chamber B is equal to the low pressure plus a pressure loss at valve NO V4 caused by the flow.

$$F_{SC,mech} = \eta_{SC} \cdot 2 \cdot i \cdot \mu \cdot \eta_{BA} \cdot \left(p_{BA,A} \cdot A_{BA,A} - p_{BA,B} \cdot A_{BA,B} - F_{BA,friction} + F_{BA,spring} \right) \qquad \text{eq. 2-7}$$

The minimal friction value is calculated by rearranging eq.2-7 and inserting it into eq.2-1 for the critical case.

$$\mu_{min} = \frac{F_{SC,mech}}{\eta_{SC} \cdot 2 \cdot i \cdot \mu \cdot \eta_{BA} \cdot \left(p_{BA,A} \cdot A_{BA,A} - p_{BA,B} \cdot A_{BA,B} - F_{BA,friction} + F_{BA,spring} \right)} \qquad \text{eq. 2-8}$$

To achieve $F_{SC,mech}$ eq.2-4 and eq.2-3 are inserted into eq.2-8. Furthermore it is assumed that chamber B of the brake actuator is connected to low pressure and that the pressure loss at valve NO V4 is known.

$$\mu_{min} = \frac{\left(p_{BA,A} + \Delta p_{HPCV} + \Delta p_{SOV1} \right) A_{SC} + F_{SC,friction} + F_{AZ,spring}}{\eta_{SC} \cdot 2 \cdot i \cdot \eta_{BA} \cdot \left(p_{BA,A} \cdot A_{BA,A} - \left(p_{LP} + \Delta p_{EOV4} \right) \cdot A_{BA,B} - F_{BA,friction} + F_{BA,spring} \right)} \qquad \text{eq. 2-9}$$

The minimal friction value depends on the operating point which is expressed by the parameter $p_{BA,A}$ in this context. For the dimensioning it is interesting to know the minimal friction coefficient which still allows self energisation. The pressure $p_{BA,A}$ in chamber A adopts values between low pressure and maximum pressure that are known from the requirements. To start with an example for the dimensioning of the SEHB the idealised minimal friction coefficient is calculated with eq.2-2. The resulting value is 0.097. For the calculation with eq.2-9 the parameters listed in table2-1 are needed as well.

The pressure loss Δp_{HPCV} of each high pressure check valve can be identified from its data sheet. The spring forces in the brake actuator and the supporting cylinder are determined during the design and can be presumed as known.

It is clearly more complex to detect the friction forces at the cylinders and the mechanical efficiency factors. They can be determined from measurements. Since they depend heavily on external factors, they should be estimated to the safe side.

The minimal friction coefficient depending on the pressure in the brake actuator is shown in figure2-2. As shown, the minimal friction coefficient is in most operating points higher when using the extended calculations. Through this, the dimensioning must be done with a calculation based on eq.2-9.

efficiency from friction force to supporting force	η_{SC}	0,95
efficiency of the brake calliper	η_{BA}	0,95
spring force of the brake actuator	$F_{BA,spring}$	3000 N
friction force of the brake actuator	$F_{BA,friction}$	500 N
area of the brake actuator, chamber A	$A_{BA,A}$	5026 mm²
area of the brake actuator, chamber B	$A_{BA,B}$	3063 mm²
area of the supporting cylinder	A_{SC}	706,9 mm²
spring force of the supporting cylinder	$F_{SC,spring}$	150 N
friction force of the supporting cylinder	$F_{SC,friction}$	50 N / 100 N
transmission from friction force to supporting force	i	0,725
pressure difference at the high pressure check valve	Δp_{HPCV}	0,2 bar
pressure difference at valve NO V1	Δp_{EOV1}	0,1 bar
pressure difference at valve NO V4	Δp_{EOV4}	0,1 bar
low pressure	P_{LP}	2,9 bar

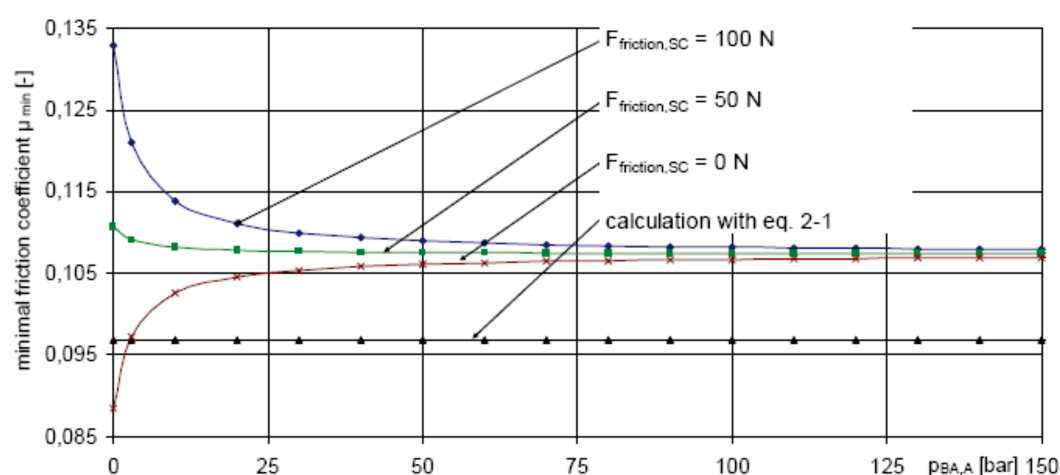*table 2-1:Parameters for the dimensioning*

figure 2-2: Minimum friction coefficient in simplified and extended calculation

2.1   Fail-Closed Concept

The SEHB shown in **-figure11** brakes if the electric control fails. The limit is set up by the opening pressure of the relief valve. The brake calliper closes until an adjustable friction force is reached. By opening the pressure limiting valve, the fluid from the supporting cylinder is let to the low pressure part of the SEHB. The pressure in the brake actuator remains or can be limited by another pressure relief at the brake actuator. When the supporting cylinder is empty, the drag force can no longer be increased. For this reason the brake force cannot be readjusted if the friction factor decreases. In this case, the brake force can only be regulated by the pump shown.

Because of the targeted railway application the SEHB incorporates a fail-closed principle. This means that in case of a failure of the power supply it has to brake with a defined brake force. The high force level is achieved by the self energisation.

The initialisation of the braking process can be implemented in different ways. Apart from a mechanical spring, a hydraulic force can be considered.   **figure23** shows three other possibilities for the implementation. As an alternative for the spring a hydraulic reservoir could be used. It either actuates the brake directly (1), activates it through an electrically operated valve (2) or actuates a second cylinder (3). The advantage of the hydraulic solution is that it can easily be switched off by a valve. In addition, the hydro accumulator can be positioned separately from the brake actuator to fit into the package. This allows for a very compact brake actuator. All variants have in common that they form a system which is hydraulically separated. No connections between the brakes are necessary, the SEHB can be installed as a module without any exterior hydraulic connections. For service issues the SEHB can be maintained when taken off the vehicle.
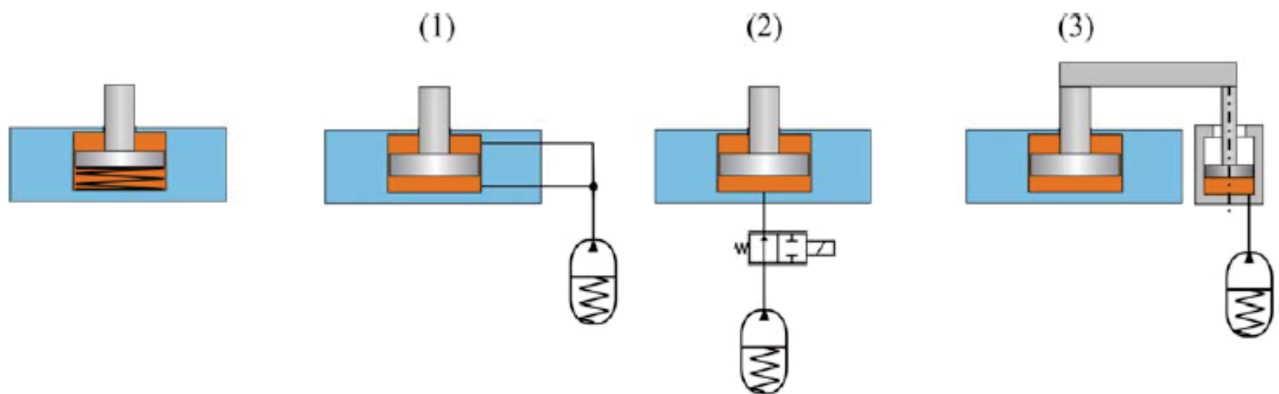

figure 2-3:Possibilities for the hydraulic "spring"-force

Another possibility to start the braking process is to use a pump as shown in **-figure11**. In this case the pump would have to start for every braking process. As an alternative the use of a(n) (electric) magnet is possible /Hof07/. However, these alternatives require a safe energy supply.

2.2   Fail-Open Concept

In contrast to the fail-closed concept there is the fail-open-concept. It is used in other applications, such as passenger cars. If the control unit fails, there must not be a brake intervention as a suddenly blocked wheel can mean a higher risk than the partial fail like of the braking system.

For a fail-open-concept, an unintentional closing of the brake must be prevented. If the electric control fails, any self energisation of the brake must be prohibited. This is why the normally open and the normally closed valves are used interchanged compared to the fail-closed system. In case of a failure in the electric control, the brake actuator is connected with the low pressure. Consequently, the closed brake is released if the electric control fails.

For a fail-open application the brake calliper is significantly simplified because, as in conventional automobile brakes, plunger cylinders can be used as actuators.

Instead of providing a calliper independent fallback for the actuator, a direct hydraulic connection to the master brake cylinder can be created, in analogy to automobile brakes when the vacuum booster has failed. Experience shows that this connection can be configured so that the safety requirements are met. **figure2-4** shows a breaking system with four SEHB callipers. Still, they are directly connected to a brake pedal. As in conventional brake callipers the brake release is passive. The elastic deformation of the seals serves for an automatic set up of a gap between brake disc and brake pads.
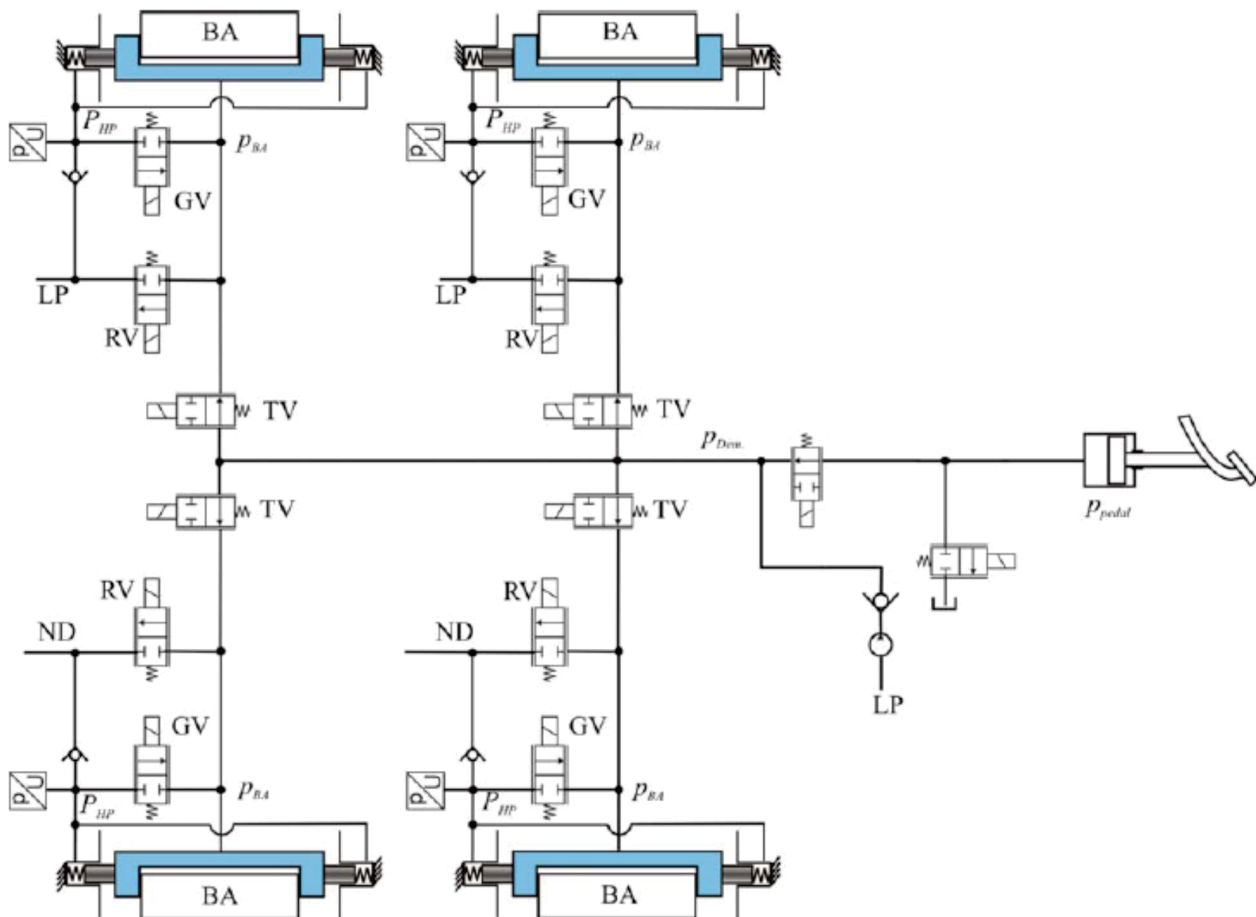

figure 2-4:SEHB for automobile applications with fail-open concept

If the driver wants to decelerate the vehicle, he initiates the brake with the pedal. Additionally, the concept provides for a pump so that the SEHB can brake independently from the driver. This is for example needed for the stability program or a driver assistance system. The pressure is led to the brake callipers which are connected to the pedal or the pump by the normally open separation valve (TV). Consequently, the brake pads are pushed towards the brake discs. As soon as a pressure slope is measured, the separation valve TV closes and the valve VV opens, enabling the self energisation. The pressure at the supporting cylinder is controlled so that the SEHB features a brake torque control in closed loop operation. The demanded deceleration is deduced from the measured pressure value at the brake pedal or set by a control device.

The brake pedal is connected to the reservoir and to the pump by two valves. If a brake process is initiated by the control unit and the pump, effects on the brake pedal are suppressed. In addition, because of the two valves, a pedal simulator can be easily implemented. For this purpose, compliance at the brake pedal has to be provided in the form of a small accumulator.

If there is a long ABS braking on fast changing surfaces the oil from the supporting cylinder is used up. In this case the brake control unit closes the brake calliper further by actuating the pump, as in a conventional electro-hydraulic brake.

2.3   Conflicts of aims

The requirements a brake system must satisfy can be summarized in five categories: forces and dynamics, security, package and weight, comfort and costs. In this chapter contradictions are discussed which arise from the optimisation.

The central requirement is to guarantee a specified stopping distance. For this purpose, the brake is defined. Due to the braking principle, the highest pressure in the supporting cylinder occurs at the maximum brake force. As the maximum system pressure is limited by the system components, the area of the supporting cylinder and the transmission ratio *i* are dimensioned first. This first step is shown in top position of **-figure25**. The transmission ratio describes the ratio between friction force and supporting force. It results from the system configuration and especially from the arrangement of the supporting cylinder. The area of the supporting cylinder is then set by the cylinder's force and by the allowed pressure.
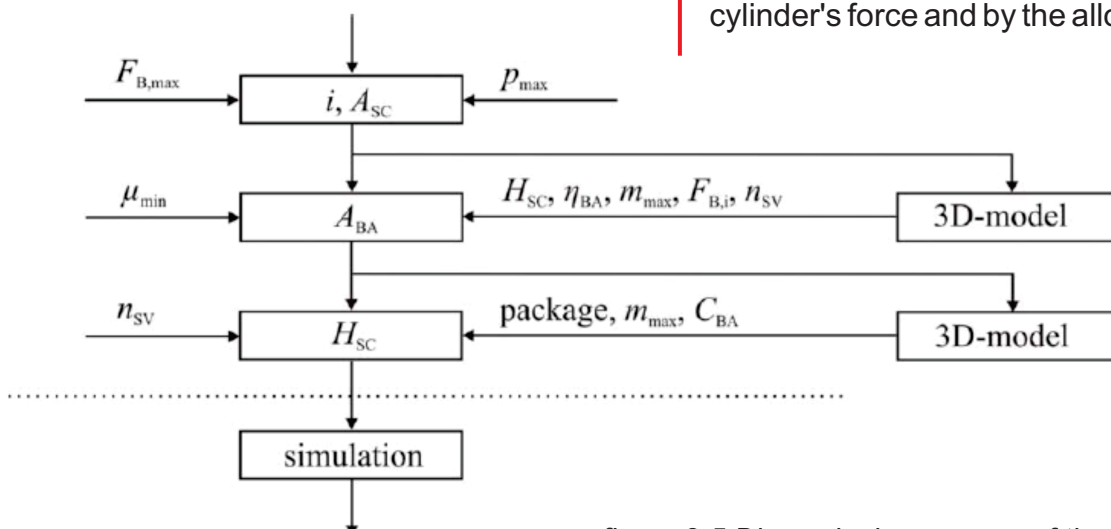

figure 2-5:Dimensioning process of the SEHB

The components, particularly the cylinders' seals, the poppet valves and the check valves have to leak as little as possible. Poppet valves show minimal leakage due to surface roughness of the valve seats and tappet /Smi08/. Plastic seals at the valve's seat minimise leakage. However, these seats are limited in the maximum pressure range. As pilot-operated valves have a long response time they can only be used in the SEHB under very limited conditions. Spool valves always leak because of the radial gap between the spool and the housing. A smaller gap reduces the leakage but increases the valve's manufacturing costs considerably. In addition, the valve becomes more vulnerable to dirt in the oil.

The second step shown in -figure25 is the dimensioning of the brake actuator's diameter A_{BA}. It can be derived from the minimal friction coefficient μ_{min} which still guarantees the functionality of the SEHB. The relation between the minimal friction coefficient and the area of the brake actuator has been derived in -eq.29. The friction coefficient during the operation of the brake depends on many influences. A water film on the brake disc is an example for a condition that implies a very low friction coefficient. To ensure the brake's functionality, the minimum brake coefficient has to be set sufficiently low. Choosing a low friction coefficient means a large braking actuator. The needed oil volume for clamping the brake increases proportionally with the area of the brake actuator. However, the volume of oil supplied by the supporting cylinder is limited. Choosing a very low minimal friction coefficient, either the supporting cylinder's stroke must be very long or the number of possible braking processes n_{SV} is small.

After this number of braking processes the supporting cylinder is exhausted and has to be reset to its original position or the pump must be activated. The supporting cylinder's stroke is mainly restricted by the available package and the allowed weight. As the brake calliper has to follow the stroke of the supporting cylinder, the needed space matches the traversed envelope. The SEHB's weight has to be particularly small if the brake calliper has to be added to the unsuspended mass, e.g. in passenger cars. In railway vehicles the brake callipers are attached to the bogie which is located behind the primary suspension. In this case a compromise has to be found, which, on the one hand guarantees the functionality at very low friction coefficients and on the other hand does not unnecessarily increase the package and the weight.

When dimensioning the supporting cylinder's stroke, the third step, further restrictions have to be considered in addition to limits of the package. First of all, to calculate the needed oil volume for clamping the brake, the stiffness of the brake calliper C_{BA}, including the brake pads and the brake disc, is decisive. During the supporting cylinder's stroke the radiuses' ratio of supporting force and brake force can change. This deviation also has to be considered when the minimal friction coefficient is set. If the directions of the supporting force and brake force vectors change, the allowed shear force at the supporting cylinder must be taken into consideration. -figure26 shows a solution to combine a small package with a long stroke of the supporting cylinder. The curved path prevents that the brake calliper exceeds the brake disc significantly. The straight parts of the path are fitted to the supporting cylinder so that the shear force is minimal.
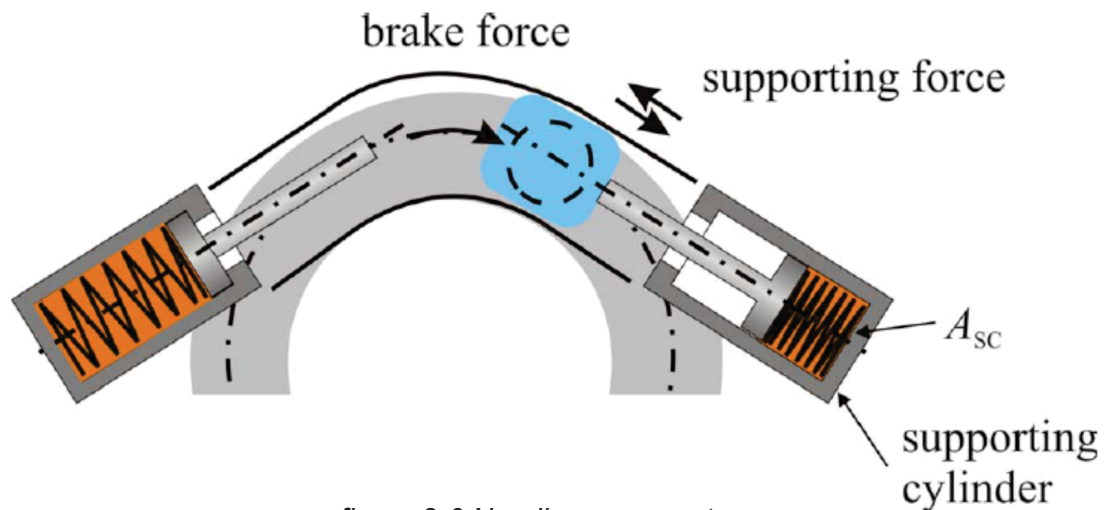
figure 2-6:Non-linear support

The minimal friction coefficient also influences the choice of the valves. When the friction coefficient is low, the valve needs a large flow area. Near the minimal friction coefficient the pressure difference at the valve is small. At the same time the brake calliper needs its maximum clamping force for deceleration. Additionally, because of the small stiffness, more compression oil is needed. In contrast, at high friction coefficients a high pressure difference at the valve occurs. A small volume flow at high pressure differences is desirable to allow a constant dynamic of the brake /Lie08/. This requirement can be fulfilled by proportionally operating valves or by using a flow control valve. Another possibility to influence the dynamic at different operating points, is to change the working areas at the cylinders.

For this purpose a brake actuator can be built up from several cylinders. The cyclic self energisation of the SEHB can be adjusted in steps if the brake is controlled adequately /Hof07/.
This idea can also be implemented in another way. figure2-7 shows a brake actuator consisting of a differential cylinder whose cylinder housing forms the piston of a plunger cylinder which is pushed in by a spring. When the brake is released the spring presses the cylinder housing to the inner stroke limit. The differential cylinder is hold back against the spring because of a pressurised chamber B. At high friction coefficients the cylinder housing stays in this position and only the differential cylinder is active. If friction values are low, chamber A is closed by valves 1b and 2b and the supporting cylinder is connected to the plunger cylinder's chamber.
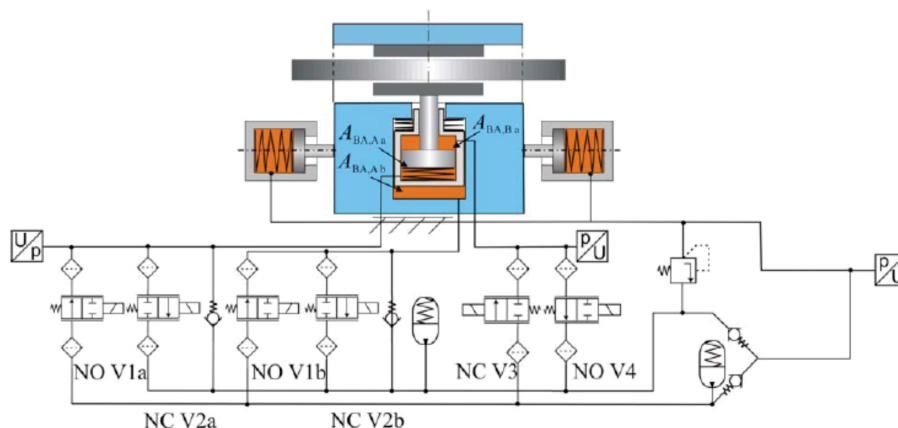

figure 2-7:Brake actuator with reinforcement cylinder

Apart from changing the area of the brake actuator, the supporting cylinder can be adjusted to the current friction value. For this purpose several supporting cylinders with the same or different areas can be used. They can be connected to high or low pressure by valves. Another possibility is designing them like the brake actuator in **- figure27**. If friction coefficients are low, only a small area of the supporting cylinder is activated. Thus, a high closed-loop gain is implemented. In case of high friction coefficients, the active area of the supporting cylinder is enlarged, improving the controllability.

It is also possible to design the supporting cylinder in a way that at first a small area is active. After a short stroke, the first cylinder step moves against a stroke limiter and the friction force is supported by the whole cylinder area.

1   SEHB prototype

 figure31 shows the prototype built up at the IFAS. The brake calliper is a floating calliper with a differential cylinder as brake actuator. It rides on guide bolts. A purchased block cylinder is used as the clamping cylinder. On its backside, depicted on the left side of figure31, the valve block is directly flanged. On the front side of the cylinder a tensioned spring is installed in axial direction to provide for the fail-closed behaviour.
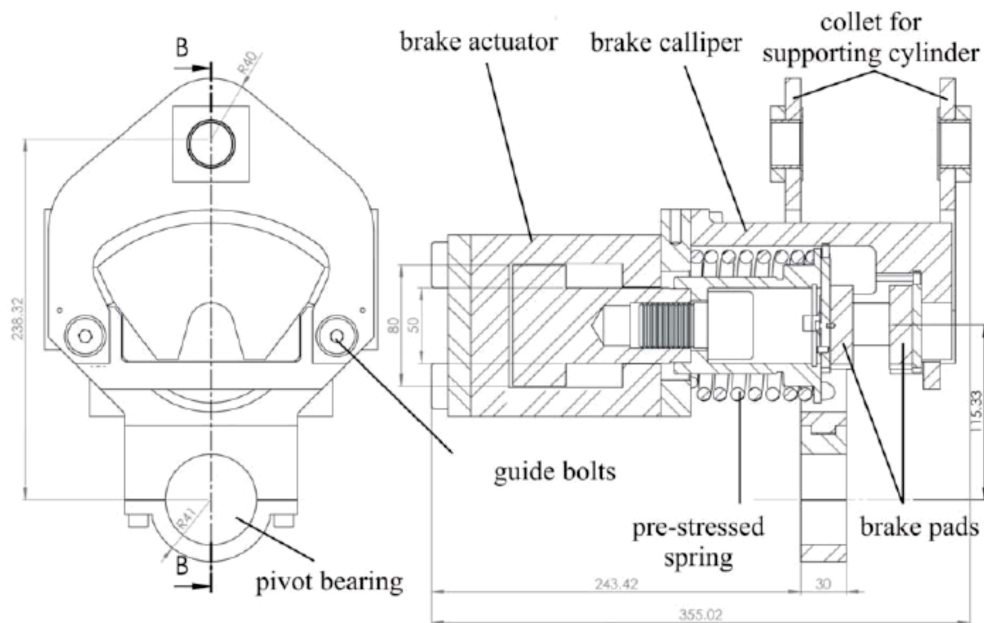

figure 3-1:Brake actuator of the SEHB prototype

A pivot bearing connects the floating calliper to the machine bed. The rotating axis is coaxial to the rotating axis of the brake disk. Due to this fact, the brake pad is directed on the brake disc with a constant friction radius. Contrarily to the brake principle shown in   - figure11, the supporting cylinder is designed as a double rod cylinder. Thus, a standard hydraulic cylinder is used. The spring reset is not integrated into the cylinder, it is mounted to one side of the cylinder.

The resulting demand for more construction space is accepted for the prototype. Both chambers of the cylinder are connected to the valve block by hoses.

The high and the low pressure accumulators are connected to the valve block by hoses, too. They are mounted onto the machine bed next to the brake. A piston accumulator with a volume of about 7 ml is used as the high pressure accumulator. The volume of the low pressure accumulator is much larger due to the necessary compensation of wear resulting in the brake actuator's stroke.

The prototype is mounted in the institute's test field as shown in figure3-2. It is driven by a flywheel powered by a hydraulic motor, shown on the right side of figure3-2. The friction force is controlled by a real-time rapid prototype control processor.

The test stand software includes maps that compensate the valves' pressure-dependant behaviour allowing for a delicate brake force control. The calculated control signal for the valves is the current in the valves' solenoids
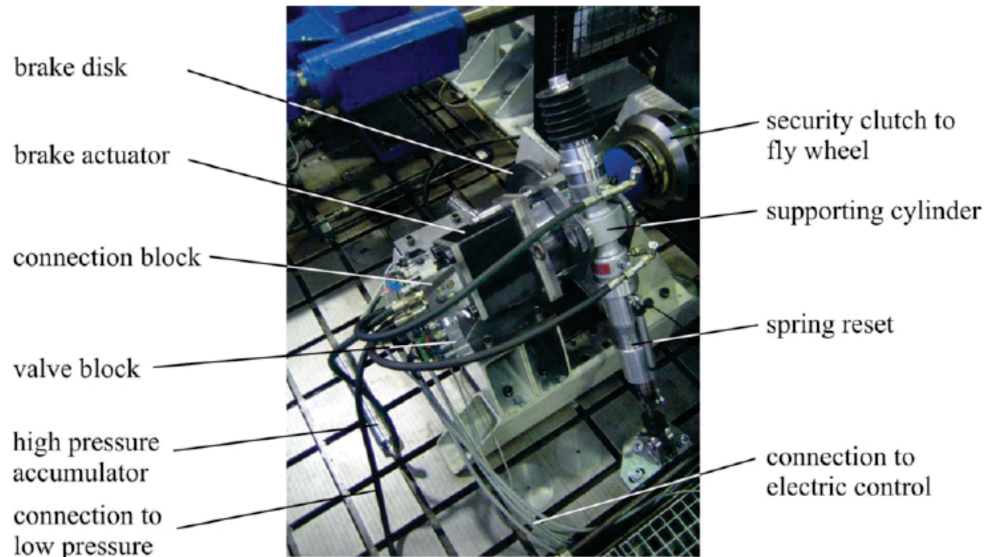

figure 3-2: Prototype on machine bed

3.1   Valve unit

The valve unit consists of a connection block which is directly screwed on the brake actuator and of the valve block itself. The connection block is manufactured from steel. It includes the check valves as well as the connections of the hydraulic lines to the accumulators and to the supporting cylinder. In addition, the pressure sensors are installed onto the connection block.

The solenoid valves depicted in figure11 are fit into the valve block. To control the flow at the valve unit in a broad range, the valves are doubled. 2/2-way valves from an automotive brake application are used. These valves feature a low price, low leakage, high dynamics and a small need of package. The valves are fitted into the valve block as shown in **figure33**. When pressed into the block, the valves' housing seals by a knife edge.
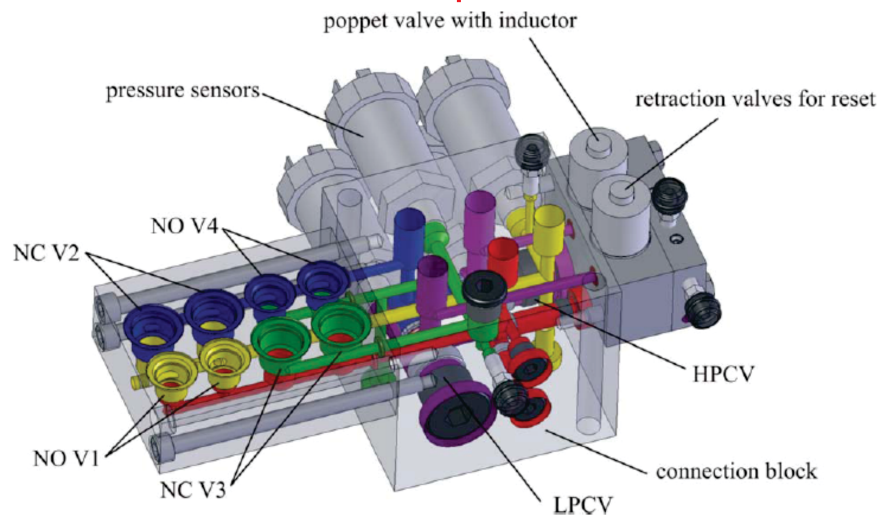

figure 3-3: valve block

Since a double rod cylinder is used, an additional valve is necessary. It connects the two chambers of the supporting cylinder to reset it to its original position. This valve function is included in a small valve block which is mounted on the connection block, too. As reset valves, two more normally closed valves are installed back to back. This arrangement is necessary because the normally closed valves open in one direction like check valves.   - figure33 shows both valves with attached solenoid.

4   Measurement results

In the last chapter a measurement is exemplarily presented and analysed. **figure41** shows a measurement during which the brake force is varied between two levels in the form of a ramp /Ewa08/. The upper measurement plot shows the reference brake force and the actual brake force calculated from the pressure sensor at the supporting cylinder.
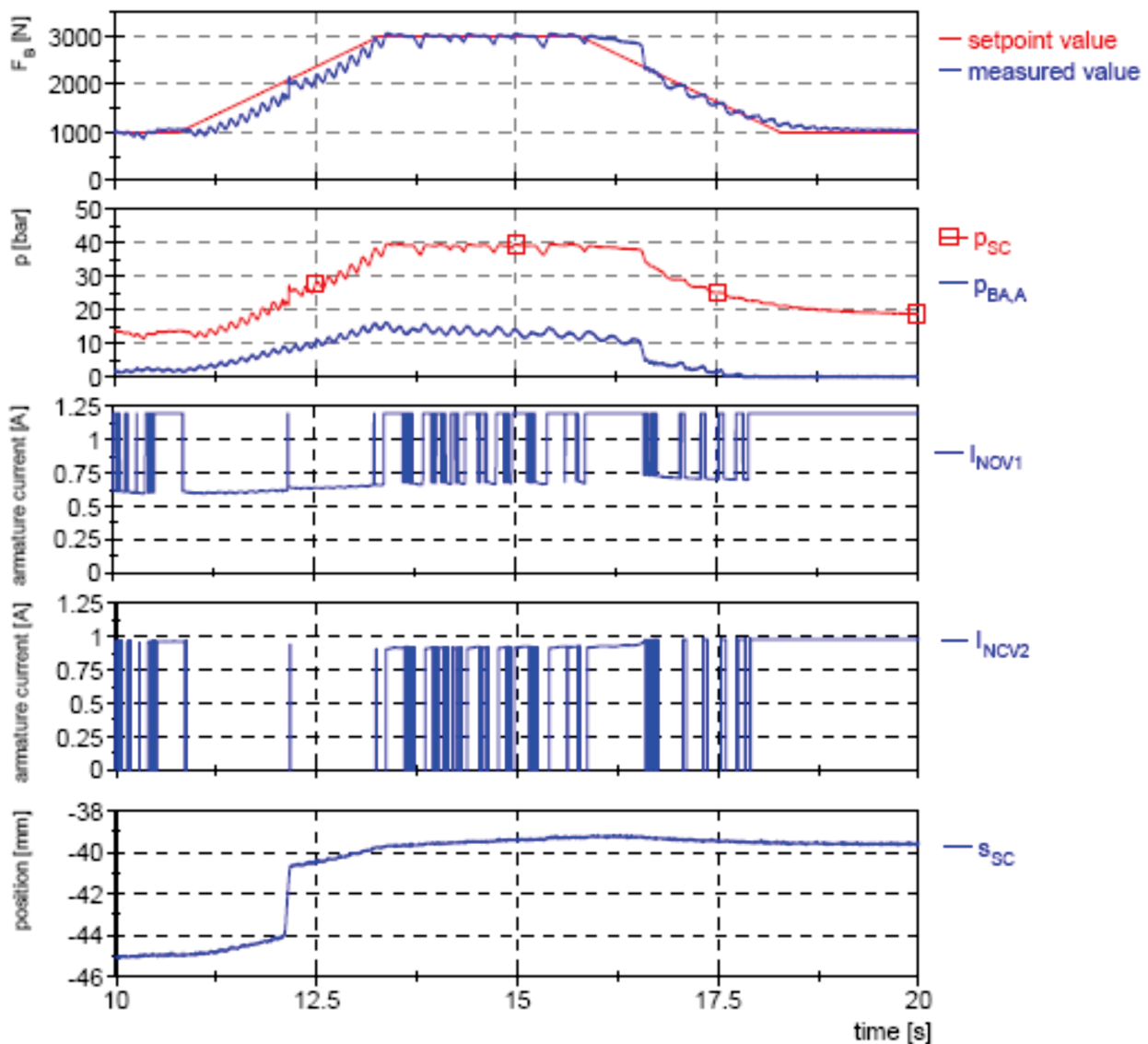
figure 4-1:measurement results /Ewa08/

In the second plot the pressure values in the brake actuator and in the supporting cylinder are shown. The pressure in the supporting cylinder is directly proportional to the braking force. Therefore it is constant when the braking force does not change. The slight pressure decline in the brake actuator at a brake force of 3000 N is explained by an increasing friction value due to the heating up of the brake disc and the brake pads.

The two measurement plots in the middle show the electric actuation in the valve solenoids. These currents are given by the control unit and generated by an amplifier card. The value of the current grows when the reference brake force increases since the poppet valves are influenced by the pressure difference at the valve's tappet. The steps in the current profile are due to changes of the valve's operating mode. There is a hysteresis between the value of the current, at which the valve is safely closed and the value at which it starts opening. Starting in the middle of these values, the control unit adjusts the current to set the desired opening of the valve.

The measurement plot at the bottom shows the position of the supporting cylinder. The high slope at about 12 s is a consequence of filling the high pressure accumulator. The stroke of the supporting cylinder during this measurement is 5.5 mm. Thereby 3 mm are needed to fill the accumulator. Since the stored oil is needed to release the brake completely, this part of the supporting cylinder's stroke only occurs once during a braking process.

1   Summary and Perspective

The presented dimensioning of the SEHB provides a good insight to the mechatronical system of the SEHB. The dimensioning requires that all parameters are considered carefully which influence the self energisation. The examination of the SEHB at low friction coefficients is indispensable to guarantee the functionality of the SEHB in all situations.

The presented prototype demonstrates the functionality of the SEHB. In further development steps the characteristics of the reinforcement can be examined with the prototype. A simulation model can be calibrated to measurements at different operating points. The verified simulation model allows realistic conclusions about dynamic characteristics and controllability. In a static dimensioning this is only possible in terms of quality.

The SEHB can be built in different system configurations. Therefore it is appropriate for applications in different environments with different requirements and security concepts. In this article a fail-closed security concept is presented, which meets the security concept of railway vehicles. The SEHB for automobile applications pursues a fail-open security concept. It also includes a fall back level with a direct hydraulic connection between thebrake pedal and the wheel brakes.

6. Bibliography

/AN07/---„Aachener bremsen besser", Aachener Nachrichten, 08.05.2007

/Bre04/ Breuer, Bert; Bill, Karlheinz H.- „Bremsenhandbuch – Grundlagen, Komponenten, Systeme, Fahrdynamik"; 2. edition 2004, Vieweg Verlag

/Ewa08/ Ewald, Julian; Liermann, Matthias; Stammen, Christian; Murrenhoff,Hubertus - „Application of proportional seat valves to a self-energising electro-hydraulic brake", in: *Fluid power and motion control*: (FPMC 2008), Bath, GB, Centre for Power Transmission and Motion Control

/Gom06/ Gombert, Bernd; Gutenberg, Philipp - „Die elektronische Keilbremse – Meilensteine auf dem Weg zum elektronischen Radantrieb", in *ATZ*, 108(11), 2006

/Her08/ Hermanns, Marcel; Hennen, Martin; Liermann, Matthias; Stützle, Thorsten - „Intelligentes, integriertes Einzelrad-Antriebs-Brems-Modul (EABM)", in: *Eisenbahntechnische Rundschau*. Hamburg : DVV Media Group, Eurailpress. - 57 (2008)

/Hof07/ Hofmann, Dirk; Baumann, DietmarDisk brake, patent DE102007016250A1, 04.04.2007

/Lie06/ Liermann, Matthias; Stammen, Christian - „Selbstverstärkende hydraulische Bremse für Schienenfahrzeuge: Intelligentes, Integriertes Einzelrad-Antriebs-Brems-Modul", in: *O + P* Nr. 50 2006)

/Lie08/ Liermann, Matthias - „Self-engergizing Electro-Hydraulic Brake", dissertation, 2008, Shaker Verlag

/Mur08/ Murrenhoff, Hubertus - „Servohydraulik – Geregelte hydraulische Antriebe", Shaker Verlag 2008

/Smi08/ Schmidt, Matthias; Murrenhoff, Hubertus; Lohrberg, Henrik; Körber, Franz-Josef - "Influencing parameters on tightness of hydraulic seat valves", in: *Fluid power and motion control*: (FPMC 2008), Bath, GB, Centre for Power Transmission and Motion Control

/Tou64/ Tournier, Yves; Henri, Beauchamp - Disk brake, patent FR19630941849, 18. Juli 1963