Application of proportional seat valves to a Self-energising Electro-Hydraulic Brake

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ABSTRACT

A new hydraulic brake utilising a self-energising effect has been developed at the Institute for Fluid Power Drives and Controls (IFAS). The Self-energising Electro-Hydraulic Brake (SEHB) generates the brake pressure by supporting the brake torque via a hydraulic cylinder and hence doesn’t need an external power supply. Until now, the SEHB has been used with seat-type switching valves for control of its brake torque only. Spool type valves cannot be used for SEHB because of leakage in the closed position due to radial clearance.

For high requirements concerning comfort and dynamics this paper presents a valve concept using 2/2 way proportional seat valves. The major advantage over previous concepts using switching valves is the adjustable closed loop gain. As the result of a simulation study regarding the requirements of the target application, a configuration of eight 2/2 way valves is set up. Measurements of the valve tappet’s position with a laser vibrometer show the dynamics of the used valve-types. A map of the flow is measured to regard the pressure dependency in the SEHB controller. The valves are integrated in a compact unit for the SEHB prototype. The paper finishes with first results of closed loop brake force control.

1. Introduction

The brake concept of Self-energising Electro-Hydraulic Brake (SEHB) combines high dynamics and high force to weight ratio of closed-loop controlled hydraulic actuation with high efficiency by using the principle of self-energisation. It has been developed at the Institute for Fluid Power and Controls (IFAS, RWTH Aachen University) within a research project funded by the DFG (German Research Foundation). Its concept has been introduced in (1), (2). The brake calliper is mounted pivoting around the wheelset to use the brake torque as the source of power to supply hydraulic pressure for braking. Only low electric power for the operation of hydraulic valves, pressure sensors and controller electronics is required to operate the brake.
The design of the prototype has been published in (4). In first tests the brake force control has been realised using seat-type switching valves from an anti-lock brake system. For applications with higher requirements like electronic stability systems (ESP) or the electro-hydraulic brake (EHB) proportional seat valves have been developed (5). They allow a more continuous control of the flow and pressures.

1.1 SEHB concept using single-acting supporting cylinders

The idea of SEHB is that the pressure needed for actuation of a hydraulic disc brake is gained from the hydraulic support of the friction force. Unlike conventional brakes, where the brake calliper is fixed, in the SEHB concept it is movable tangential to the friction contact. In the case of braking, the friction force acts on the supporting cylinder causing a pressure build-up. In previous publications (4) a brake concept has been presented where the brake calliper is supported by a synchronising supporting cylinder, where the two pressure chambers are mechanically connected. Another option, presented in Figure 1, is to use two single-acting cylinders mounted on both sides of the brake calliper. Both of them are fully extended in the middle position. The brake in Figure 1 is shown for the case of braking, indicated by the pushed-in single acting supporting cylinder on the left.

Beginning with the open brake set up to a defined clearance between brake pads and brake disk, valves PV 2 and PV 3 are opened. This enables the preloaded spring to press the brake linings against the brake disk. As soon as the brake has moved beyond the clearance, the brake pads are pressed with the spring force towards the brake disk.

Dependent on the direction of rotation, one of the single-acting supporting cylinders is pressurised. The other supporting cylinder remains extended at its full stroke due to a mechanical stroke limitation and releases from the frame. The high pressure check valve conducts the fluid to the high pressure accumulator.

For increasing the brake force, valves PV 1 and PV 3 are opened. High pressure is applied on the piston face side of the brake actuator, while the ring side is connected to low pressure. The increased compression of the actuator increases the force acting on the supporting cylinder. This process is self-energising dependent on the ratio of piston areas of supporting cylinder and brake actuator.

Decreasing the brake force is done by opening valves PV 2 and PV 4. The piston face chamber is released, while the piston ring side is charged, reducing the actuator force and yielding a negative feedback of the supporting pressure on the braking pressure. By this mode of operation the brake force cannot be reduced completely and the brake pads cannot be lifted off from the brake disk. The preloaded spring in the brake actuator has to be pushed back. However, the pressure in the supporting cylinder is now too low. Therefore the high pressure check valve disconnects the supporting cylinder from the high pressure accumulator. With the oil supplied by the high pressure accumulator a defined clearance is set between brake pads and brake disk.

The advantage of this supporting cylinder configuration is that some components of the system presented in (4) are no longer required. Through the direct connection between both
supporting cylinders there is only one hydraulic capacity which provides the pressurised oil to the valves in both driving directions. Two of the original four check valves can be saved. Only a high pressure check valve for separation of supporting pressure and high pressure accumulator and a suction check valve for separation of supporting pressure and low pressure reservoir are needed. This suction valve is needed during the retraction of the supporting cylinders. It connects the supporting cylinder with the low pressure part after braking. The pushed in single acting supporting cylinder is retarded into its end position by a spring, while being filled with oil from the low pressure reservoir.

**Figure 1:** Principle of Self-energising Electro-Hydraulic Brake (SEHB), using four proportional seat-type valves

The SEHB, which so far has been developed for a train application, is designed with a fail-closed concept. This means, that in case of a failure like the loss of electric power, the brakes apply and the vehicle is stopped safely. This is realised hydraulically by a configuration of normally open and normally closed valves depicted in (3). Without electric power the seat valves are either opened or closed by an integrated spring. The power-off state is shown in Figure 1.
2 Proportional valves from automotive applications

The normally opened (NO) and normally closed (NC) types of the seat-type 2/2 way proportional valves used for this study were provided by Continental Teves. They are originally used for automotive brake applications. The NC valve is used for the electro hydraulic brake (EHB). The NO valve has been designed for a traction control system (ASR), where the wheel slip is controlled by braking the wheel in case of loss of traction. The valves are made as cartridges for being press-fitted into a valve block.

Three forces are applied to the valve tappet. For the normally closed valve, shown in Figure 2 on the left side, the spring force $F_{Spring}$ closes the valve, while the force generated by the pressure difference acts opening the valve as well as the magnetic actuation force $F_{magnetic}$. The valves have a pressure-dependent characteristic due to the pressure drop at the seat. In comparison to seat-type switching valves the force between armature and solenoid is constant over the tappet movement. By this, partially open positions of the valve tappet are achieved.

![Figure 2: Normally open and normally closed seat-type valve](image)

The NC valve has a rated flow of 40 cm³/s, the NO valve has 52 cm³/s at a pressure drop of 100 bar using DOT 4 brake fluid. However, the SEHB prototype is using HLP46 hydraulic fluid instead. The different specific weights result in a different rated flow, which can be calculated by Eq. 1.

$$Q_{HLP} = \sqrt{\frac{\Delta p_{HLP} \cdot \rho_{DOT4}}{\Delta p_{DOT4} \cdot \rho_{HLP}}} \cdot Q_{DOT4}$$

*Eq. 1*
Table 1: Rated flow for a normally open and a normally closed valve

<table>
<thead>
<tr>
<th>Valve type</th>
<th>flow rate for DOT 4 @ 100 bar ($\rho_{\text{DOT-4}} = 1060 \text{ kg/m}^3$)</th>
<th>flow rate for HLP 46 @ 35 bar ($\rho_{\text{HLP}} = 870 \text{ kg/m}^3$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[ml/s] [l/min]</td>
<td>[ml/s] [l/min]</td>
</tr>
<tr>
<td>NC EHB</td>
<td>40 2.4</td>
<td>26 1.56</td>
</tr>
<tr>
<td>NO ASR</td>
<td>52 3.12</td>
<td>34 2.04</td>
</tr>
</tbody>
</table>

Concerning the needed flow, two main requirements can be identified. A high rated flow is needed at low brake forces or in case when the brake is not engaged yet. In this operating point the pressure difference at the valve is low, thus the valve has to provide a wide flow section. On the other hand, at high brake force, the applied pressure difference at the valve is high. The bulk modulus at high pressure is high hence the quotient $\frac{dp}{dQ}$ describing the pressure build-up is high as well. Therefore a high resolution of the valve is needed.

The valve has to combine the two characteristics composed of a high rated flow and a good resolution at small openings. To be able to make use of the valve resolution, the pressure dependent behaviour must be known very well.

The first step is the determination of the required rated flow. A high rated flow can be achieved by using more than one valve in parallel. The number of valves needed can be derived by requirements given form the desired railway application. The time from an open brake calliper to a defined brake force is limited. Based on the requirements given from the EABM framework a limit of half a second is set. With the rated flows given by the above described valves a simulation model in DSHplus is build up to get a measure for the closing time using one or two valves.

Table 2: Tested valve configurations

<table>
<thead>
<tr>
<th>Configuration</th>
<th>PV 1</th>
<th>PV 2</th>
<th>PV 3</th>
<th>PV 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. 1</td>
<td>1 NO</td>
<td>1 NC</td>
<td>1 NC</td>
<td>1 NO</td>
</tr>
<tr>
<td>No. 2</td>
<td>2 NO</td>
<td>2 NC</td>
<td>2 NC</td>
<td>2 NO</td>
</tr>
<tr>
<td>No. 3</td>
<td>2 NO</td>
<td>2 NC</td>
<td>1 NC</td>
<td>1 NO</td>
</tr>
<tr>
<td>No. 4</td>
<td>2 NO</td>
<td>1 NC</td>
<td>1 NC</td>
<td>1 NO</td>
</tr>
</tbody>
</table>

The possible valve configurations with a number of valves between four and eight are shown in Table 2. In configuration No. 2 all valves are doubled. In configuration No. 3 and No. 4 the valves connected to the piston rod side are not doubled. For benchmarking these configurations, the simulation is run with a reference signal curve of the demanded brake force as shown in Figure 3.
Figure 3: Simulation to verify valve configuration (Configuration No. 2)

From the simulation results five characteristic parameters are extracted, these are:

- $T_A$: Time to get over a clearance of 1 mm between brake pad and disk
- $T_B$: Time for step response from 0% to 90% of the maximum brake force
- $T_C$: $T_A + T_B$
- $P_{OS}$: Overshooting at step response from 0% to 90% of the maximum brake force

<table>
<thead>
<tr>
<th>No.</th>
<th>$T_A$</th>
<th>$T_B$</th>
<th>$T_C$</th>
<th>$P_{OS}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.662 s</td>
<td>0.142 s</td>
<td>0.804 s</td>
<td>1.4%</td>
</tr>
<tr>
<td>2</td>
<td>0.402 s</td>
<td>0.074 s</td>
<td>0.476 s</td>
<td>11.2%</td>
</tr>
<tr>
<td>3</td>
<td>0.675 s</td>
<td>0.084 s</td>
<td>0.759 s</td>
<td>8.7%</td>
</tr>
<tr>
<td>4</td>
<td>0.662 s</td>
<td>0.085 s</td>
<td>0.747 s</td>
<td>9.2%</td>
</tr>
</tbody>
</table>

With a configuration of four singular valves (No. 1) the dynamics at low brake forces are too slow. The configurations where only one or two valves are doubled (No. 3 and No. 4) cannot offer the short time to get other over the clearance between brake pad and brake force ($T_A$), although they do well in reaching 90% of the maximum brake force after the brake has travelled over the clearance ($T_B$). Configuration No. 2 is chosen to be used with the SEHB prototype because it has the lowest values for $T_A$, $T_B$, and $T_C$. The overshooting with this configuration can be reduced to the valve of configuration Nr. 1 by using the parallel valve only to get over the clearance and at low brake forces. By this a fast response of the brake with a very small overshoot can be realised.
4. Valve measurement

In this chapter a measurement of the valve tappet position and a measurement of the flow are presented. With a laser vibrometer the valve tappet’s position of a NC valve is measured. Due to the design of the cartridge valves only the NC valve could be opened in such a way that the laser could be focused onto the tappet. The NO valve has a filter element above the valve tappet which could not be removed without damaging it. The optical principle allows a measurement up to a very high dynamic which the flow sensor cannot achieve. However, the laser vibrometer needs an optical access to the valve tappet hence the measurement cannot be used for the pressurised valve.

The test bench with the laser vibrometer is depicted in Figure 4. The reference signal generated by the measuring computer is given to the current driver. The current driver supplies a closed loop controlled current to the valve’s solenoid. By the laser vibrometer and its amplifier the tappet position is given back to the measuring computer.

![Figure 4: Test bench with laser vibrometer](image-url)

At first, a static measurement is done to determine the current where the valve starts to open. As shown in Figure 5, without pressure, the valve is beginning to open at about 0.9 A and is reaching its maximum opening of 125 µm at about 1.1 A. The upper graph shows the measured position of the valve tappet over time. The lower plot shows the displacement of the tappet as a function of the current for both directions of actuation, opening and closing the valve. The measured step which occurs in the tappet position at 0.92 A and 0.86 A could not be settled conclusively. There are three theories: One is that due to high friction, for the nearly closed valve, the tappet stops. Another explanation is that the tappet is tilting in its seat. The third possibility is that another part of the valve is moving prior the tappet starts moving out of its seat.

There is a hysteresis between the opening and closing of the valve. Due to the friction force, which is always directed against the direction of movement, the force for pulling the tappet out of the seat against the spring is higher than the force for closing it. The valve is
designed such that the valve is closed by a spring pressing the tappet into the seat, see Figure 2. The magnetic force pulls the tappet against the spring out of its seat. While the valve is opened, the friction force is directed against the magnetic force. When closing the valve, due to reducing the force from the solenoid, the friction force is aligned to the same direction as the magnetic force.

**Figure 5:** Static measurement of tappet position

### 4.1 Tappet dynamics

For an evaluation of the tappet dynamics, the solenoid is actuated with a sinusoidal signal and the response is measured. The sinusoidal signal increases its frequency exponentially to provide equal number of oscillations through the frequency bandwidth.

**Figure 6:** Sinusoidal signal for the bode diagram
Beginning with the half open valve, the reference signal is applied as shown in Figure 6. The position of the tappet is measured. The data acquisition is done with a dSPACE rapid control prototyping board where the data is logged at a rate of 40 kHz. By Fast Fourier Transformations (FFT) the measured data is transformed to a bode diagram. The measurement is done for 10%, 25%, 50% and 100% of the possible tappet movement. Figure 7 shows the bode diagram for 10% of the possible tappet movement. The bode plot shows increasing noise for amplitude and phase values above 200 Hz. This is caused by different dynamics of the valve in closing and opening direction, as can be seen from Figure 6. The difference between closing and opening dynamics results in two values for phase lag and amplitude for each excited frequency. This effect becomes more dominant for higher frequencies as Figure 7 shows.

![Figure 7: Frequency response to sinusoidal signal for 10% tappet movement](image)

From the whole measurements a characteristic of a PT1T1 can be identified. The first-order time-delay element is identified from the falling amplitude at increasing frequency. The dead time element results from the phase response with a steady rising gradient. A PT1 would reach the maximum gradient in its phase response at -45° and then would tend to -90° phase delay. The measurements, however, show a rising gradient far beyond this value of the phase delay. Measurements of the step response for closing and opening valve show that the valve is faster in closing than in opening.

### 4.2 Flow rate measurement

In addition to the dry measurement, the flow rate through both types of the used valves is measured against pressure drop and solenoid current. The valve unit is connected between two pressure controlled servo valves. The pressures at both sides of the valve unit can be adjusted to the demanded pressure with a high accuracy by these two servo valves. The oil for this measurement is supplied by a pump and not from the supporting cylinder.
Basically three variables can be set individually: The pressures before and after the valve and the solenoid current. Changing every parameter individually leads to a four-dimensional map. Comparing the flow at the same pressure drop but different absolute values, the flow is not significantly different, neither for the normally open valve nor for the normally closed valve. Therefore the degree of freedom can be reduced to two; the flow is now depended on the current and the pressure drop only.

**Figure 8:** valve flow maps for NO valve

**Figure 9:** valve flow maps for NC valve
The flow rate is displayed in Figure 8 and Figure 9 for various pressure drops over the current. A characteristic parameter for the valve is the flow at a pressure drop of 35 bar. The NO valve has a nominal flow of 1.95 lpm and the NC valve a flow of 1.6 lpm. This data complies with the data from the manufacturer. The flow rate for a fully open valve at 1.2 A is shown in Figure 8 and Figure 9 calculated by the basic equation of an orifice. A distinctive hysteresis between opening and closing is shown especially for the normally open valve.

4 Mechanical design and current driver

The eight valves are pressed as cartridges into a valve block made from aluminium as shown in Figure 10. The valves are mounted as closely together as possible for a low weight of the valve unit and a good package of the complete brake calliper where the hydraulics are mounted onto. Like at a common 4/3 way valve, the four hydraulic connections are lead out at one side. On this side the valve is put on a mounting plate with integrated check valves.

![Valve Unit](image10)

3.4 Electronic control

Each valve is driven by a current controller. The current driver is based on an asymmetric half bridge built up from two transistors and a comparator. The circuit diagram is depicted in Figure 11 consisting of a transistor between the solenoid and the power source and the second transistor between the solenoid and ground. Three states are realised. For rising current both transistors are switched on and current is conducted from source over the solenoid to ground. If the measured current is higher than a dead band around the desired current, the transistor $S_1$ between source and coil is opened. By the inductance of the solenoid the current is falling slowly while the circuit is closed by the free wheeling diode $D_1$ and a shunt. For cutting down the current quickly, the energy stored in the inductance is moved to the capacitor. Therefore both transistors are turned off and the solenoid is
“pushing” its current over the diode D₂ to the capacitor with the other port connected to ground by diode D₁. The magnetic force applied from the solenoid to the valve tapped is cut off fast.

![Wiring scheme of current driver](image)

**Figure 11:** wiring scheme of current driver

### 4. Brake force control

The flow maps depicted in Figure 8 and Figure 9 are implemented to a brake force control software using a rapid control prototyping board. The closed loop controller is set up with a PI controller. From simulations a P controller has been considered to work well with the SEHB (4). The hysteresis shown in the flow maps can be compensated in a first step with a switched integral controller.

**Figure 12** shows he measurement of three values, the brake force, the pressure in the piston side of the brake actuator and the stroke of the supporting cylinder. The reference signal is a ramp rising from 1000 N to 3000 N tangential brake force. The measured brake force is following the reference signal well. Only at 16 s, when the falling ramp begins, a greater control deviation occurs. This should be eliminated with some optimisation on the controller setup.

The pressure signal shows a ripple. By closing all valves manually the ripple is still present hence the ripple can be considered to be caused by the friction contact between brake disk and brake lining. Future research will attend to the question of decreasing this effect with a high dynamic brake force control.

The supporting cylinder stroke, shown in the plot at the bottom of Figure 12, points out the low needed oil supply from the supporting cylinder. For the shown ramp rising from 1000 N to 3000 N only 5.5 mm (11%) of the maximum stroke of 50 mm are needed.
Included in these 5.5 mm are about 3 mm for charging the high pressure accumulator, which is only needed once during a brake operation.

**Figure 12:** Brake force measurement
5. Conclusion and Outlook

This article presents the development of a valve unit for the SEHB prototype. Four doubled 2/2 way seat valves take over the functionality of a 4/3 way valve. The advantage in comparison to spool type 4/3 way valves is the leakage free seat and the higher degree of freedom in control.

By simulation the needed nominal flow is evaluated and the valve configuration is determined. Without the exact knowledge of the valve characteristic the controller cannot be set up to offer the desired smooth characteristic and the exact brake force control which is needed by comfort and safety demands. Therefore, the tappet dynamics of the used seat-type 2/2 way valve is measured. Additionally the flow is measured at different pressure drops over the solenoid current. The data from the measurement is considered in the SEHB controller and depicted in a flow map for the used normally open and normally closed valve.

The valves are fitted to a valve block and used for a closed loop brake force control with the SEHB prototype. The valve measurements are the basis for a fitted controller design enabling a good performance. Future work is concerned with a controller design compensating the hysteresis of the valve between closing and opening current in a better way.

The authors thank the German Research Foundation (DFG) for funding this project and Continental Teves for supply of the valves.

6. References


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