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DISC BRAKE WITH HYDROMECHANICALLY CONTROLLED BRAKE TORQUE FOR RAILWAY APPLICATIONS

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Abstract. Hydraulic disc brake systems are widely spread in commuter train applications. Due to their operation with an open loop control system, they are not able to deal with the influences of varying conditions surrounding the frictional contact zone between the brake pads and the disc. This results in a variable friction coefficient and affects the system performance. To detect the variation a closed loop control system with a feedback signal needs to be implemented in existing brake solutions to maintain high reliability and operating permit. This paper examines the design features of an existing system and finds a suitable mechanical feedback signal, whose eligibility is experimentally verified. Subsequently the geometrical calculations of the system are made and the control system is simulated. Based on the simulation results a preliminary design of the system is presented.

Keywords: Brake system; brake torque control; closed loop system; railway application.

INTRODUCTION

Railway vehicles are an efficient, reliable and eco-friendly mass transportation solution, which still provide a vast potential for further development despite their long existence. The brake system is an essential part of a functional railway vehicle. For the implementation of various braking functions, such as general operation, emergency, holding or parking, both passive spring-applied actuators and actively acting brakes are used. The adaption of the contact force between brake pads and disc or wheel can be affected both gradually and continuously. Especially in case of brake operations with wheel slide protection, a continuously variable control of the contact force offers the possibility of an improved brake performance. However, all conventional disk brake systems have the common inability to record the influence of varying conditions in the frictional contact zone between brake pads and disc. Thus, the brake performance of these systems is dependent on a changing frictional coefficient due to external disturbance variables. Additionally, the prevailing operating conditions of commuter trains are characterised by harsh weather conditions, severe vibrations, strong mechanical loads and a high number of actuation cycles. **Figure 1 (a)** illustrates a conventional hydraulic brake system which is affected by a large number of various disturbances.

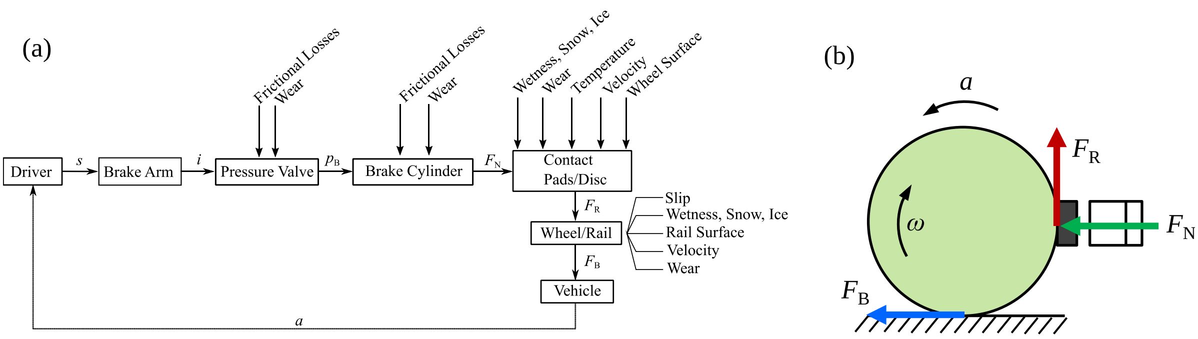


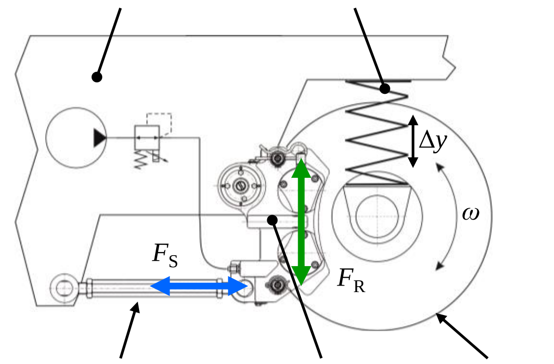
FIGURE 1. (a) Factors affecting the brake system; (b) Brake parameters

The brake cycle of a consisting system begins with the signal from the brake arm shifted by the train driver. The displacement s is transferred into the electrical signal i to the brake pressure valve. According to the signal the valve increases or decreases the hydraulic brake pressure p_B in the brake cylinder. Inside the brake cylinder, the normal force F_N is generated and applied on the brake pads and subsequently the brake disc. Throughout those processes the brake signal is disturbed due to frictional losses and wear. The normal force and the friction coefficient between the pads and the disc induces the frictional force F_R on the wheels, resulting in reducing the angular velocity of the wheel ω as depicted in **figure 1 (b)**. Due to the contact between the slipping wheel and

the rail a brake force F_B affects the vehicle. In the contact zones wetness, wear, temperature, vehicle velocity, and the surface roughness of the wheel cause disturbances. The driver perceives the deceleration a of the vehicle as a feedback from the open loop and builds a man-made closed loop brake system.
One option to compensate the introduced disturbances and to control the actual brake torque is to use the self-energised hydraulic brake (SEHB) which was developed at the Institute for Fluid Power Drives and Controls (IFAS) [1][2]. It does not require an external power supply and provides a reduced installation space. Nonetheless, the primary structure of the SEHB differs considerably from a conventional hydraulic brake system, which accompanies the market introduction by various uncertainties and difficulties. At the same time, narrow specifications request brake performances with more stringent criteria. While the brake system has to make the optimal use out of the frictional contact, the compliance with minimum brake distances for comfort reasons present challenges for the design and operation of modern brake systems. An innovative option is the integration of a hydromechanical brake torque control system into an established hydraulic brake design for commuter trains. For this purpose, in the first section of this paper the structure of a selected brake system is analysed for its ability to detect the real frictional force F_R with high reliability. A suitable mechanical feedback signal is found to build a closed loop system, which is developed and simulated in the following sections. The last section gives an overview of the preliminary design.

ANALYSIS OF AN EXISTING BRAKE SYSTEM

The selected brake system consists of a brake cylinder, a caliper, brake pads, and a supporting structure. **Figure 2 (a)** shows the integration of the brake into the bogie which includes the hydraulic supply and control elements. The brake is supported by two bearings in the bogie. The first one at the brake enclosure leads to a rotatable connection of the brake while the second one builds the supporting structure with a lever arm and fixes the brake position. During a brake process a frictional force F_R is generated and directed through the supporting structure towards the bogie resulting in a supporting force F_S.



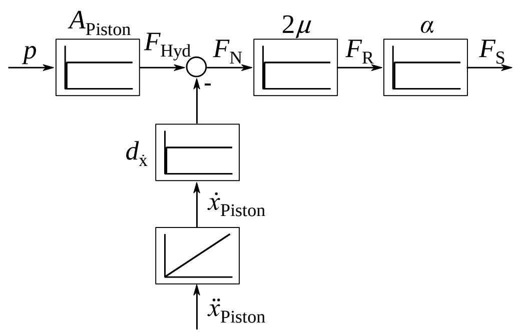


FIGURE 2. (a) Working concept; (b) Block diagram of the actual open loop system.

The open loop system is presented in **Figure 2 (b)** as a block diagram. The brake pressure p is applied on the piston of the brake cylinder resulting in the hydraulic force F_N as shown in equation (1).

$$F_N = A_{\text{Piston}} \cdot p - d_{\dot{x}} \cdot \dot{x}_{\text{Piston}} \tag{1}$$

Hereby the hydraulic force F_{Hyd} is diminished by the frictional force of the brake cylinder depending on the cylinder speed $\dot{x}$ and its friction coefficient $d_{\dot{x}}$. Multiplied by the number of brake pads and the friction coefficient μ, the frictional force F_R is achieved. The supporting force F_S can be calculated with a factor α that depends on geometric relationships like the suspension Δy of the brake disc axle towards the bogie as described in equation (2).

$$\alpha = \frac{F_S}{F_R} = f(\Delta y, \dots) \tag{2}$$

With the supporting force F_S as representative value for the real friction force the development of a closed loop control system is possible. However, bringing a control mechanism into the supporting structure will lead to a length variation Δx of the lever arm, which in turn affects the factor α. To investigate the influence of possible deflections the mechanical structure of the brake system is implemented into the geometrical processing software Geogebra. **Figure 3** shows the brake model in its end positions on the brake disc due to maximal deflection values.

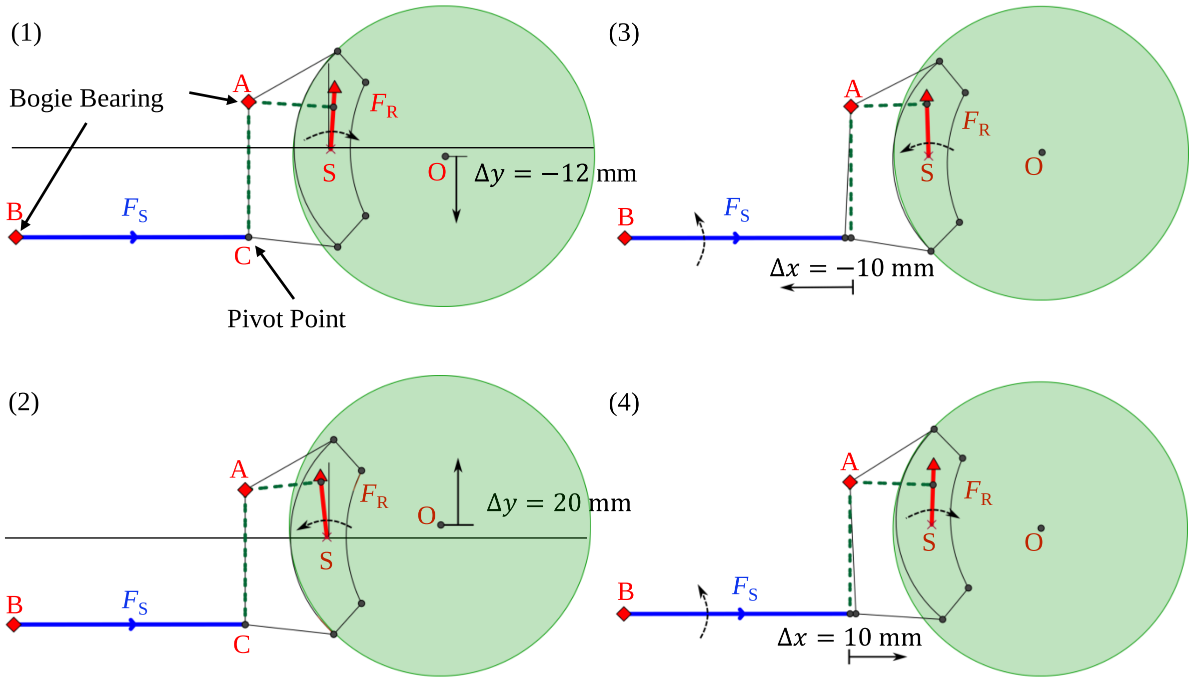


FIGURE 3. Force and position reaction due to suspension travel Δy (1),(2) and lever arm length Δx (3),(4)

The suspension travel Δy between the geometrical center point of the brake pads and the brake disc variates between -12 mm (1) and 20 mm (2) while the length Δx of the supporting structure will be changed between -10 mm (3) and 10 mm (4). Obviously, a change of the suspension travel Δy leads to a rotating friction force F_R and thus to a varying lever transmission between F_R and the supporting force F_S. The change of the lever arm length Δx also directly influences the transmission ratio due to shifting the brake pad's position on the brake disc. **Figure 4** illustrates the results for the factor α calculated by Geogebra.

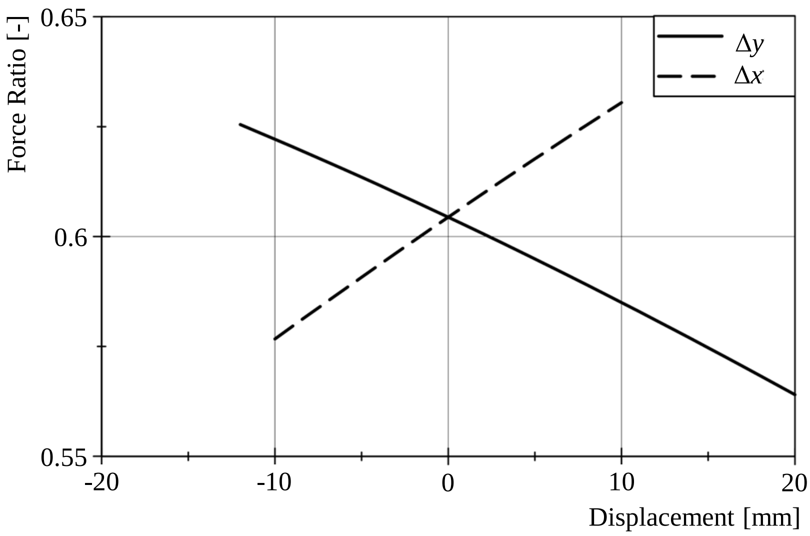


FIGURE 4. Force ratio α depending on suspension travel Δy and supporting structure length Δx

The factor α changes linearly for both displacements and shows only small deviations of 5% to 7.5%. In real operation, only small deflections are expected. Therefore, it is assumed that the supporting force F_S is nearly proportional to the frictional force F_R and can serve as a feedback signal and indicator for the real brake force of the system. Nonetheless, possible effects will be considered in simulation by varying displacement values.
The quality of the supporting force F_S is experimentally verified using a brake test rig shown in **Figure 5 (a)**.

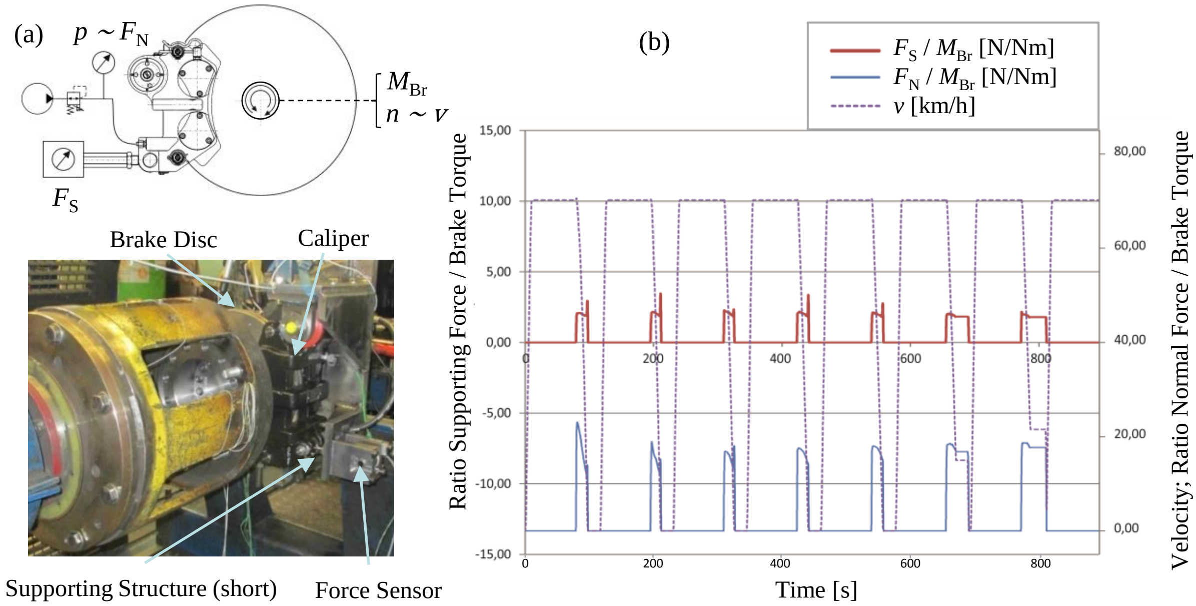


FIGURE 5. (a) Brake test rig; (b) Measurement results

The brake disc and an additional inertia mass are driven electrically. The supporting structure with shortened lever arm is connected to the test rig frame with a force sensor. **Figure 5 (b)** displays the results, which show the ratio between the measured supporting force F_S and the brake torque M_{Br} respectively the ratio of the normal force F_N and M_{Br} next to the calculated velocity. While the ratio F_S/M_{Br} is nearly constant during brake, the ratio F_N/M_{Br} changes crucially between and during the brake intervals. Additionally, the measured supporting force occurs in a technically useful span with approximately 10% of the normal force's value. Hence, the supporting force F_S can indicate the real brake torque M_{Br} and the supporting structure can be used for a hydromechanical control system.

CLOSED LOOP SYSTEM CONCEPT AND SIMULATION

After verifying the eligibility of the feedback signal F_S, a principle concept with hydromechanical elements is defined. **Figure 6** shows the simplified working concept of a closed loop brake system.

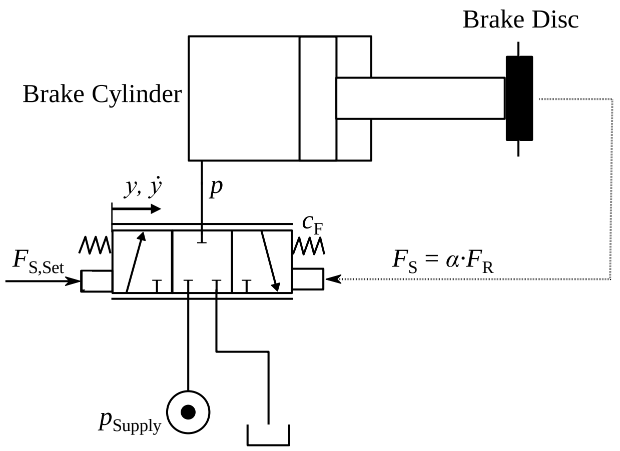


FIGURE 6. Closed loop concept

The control loop is closed using a 3/3 proportional hydraulic valve balancing between a demanded supporting force $F_{S,set}$ and the current supporting force F_S. The brake force demand is provided by the original hydraulic control system of the vehicle. The current brake force is transmitted using the supporting structure with the factor α. In case of over braking the supporting force exceeds the set brake force. Thus, the valve connects the brake cylinder with the tank resulting in reduced brake force until the equilibrium is reached again. In case of under braking the set brake force is higher than the supporting force and the valve connects the brake cylinder with the supply pressure. A mathematical model of the system is built to determine significant parameters for the system design. First, the equilibrium of forces around the valve shaft is considered in equation (3)

$$m_{\mathrm{Valve}} \cdot \ddot{y} = \sum F = F_{\mathrm{S,Set}} - F_{\mathrm{S}} - c_{\mathrm{f}} \cdot y - d_{\mathrm{friction}} \cdot \dot{y} \tag{3}$$

where m_{Valve} represents the mass of the valve spool and d_{friction} a friction parameter depending on the spool velocity. The linearized equation of the hydraulic flow in a valve is introduced in equation (4) [3] and applies for both the inlet and the outlet flow.

$$Q = \left.\frac{\partial Q}{\partial y}\right|_{p} \cdot y - \left.\frac{\partial Q}{\partial p}\right|_{y} \cdot p = V_{\mathrm{Qy}} \cdot y - V_{\mathrm{Qp}} \cdot p \tag{4}$$

The flow factors V_{Qy} and V_{Qp} express the dependence on valve opening y respectively the change of pressure p. The leakage flow is determined using the leakage pressure factor K_{lecp} as follows in equation (5).

$$Q_{\mathrm{lec}} = K_{\mathrm{lecp}} \cdot p \tag{5}$$

Subsequently equation (6) gives the pressure in the system with C_{H} as the hydraulic capacity.

$$\dot{p} = \frac{1}{C_{\mathrm{H}}} \cdot \sum Q = \frac{1}{C_{\mathrm{H}}} \cdot (Q_{\mathrm{in}} - Q_{\mathrm{out}} - Q_{\mathrm{lec}}) \tag{6}$$

Figure 7 visualizes the hydromechanical control system in a block diagram. The system is divided into four subsystems: The pilot valve, the mechanical modelling of the main valve, the hydraulic modelling of the main valve and the brake mechanics. The control loop is closed by returning the feedback signal F_{S} to the valve forces sum point.

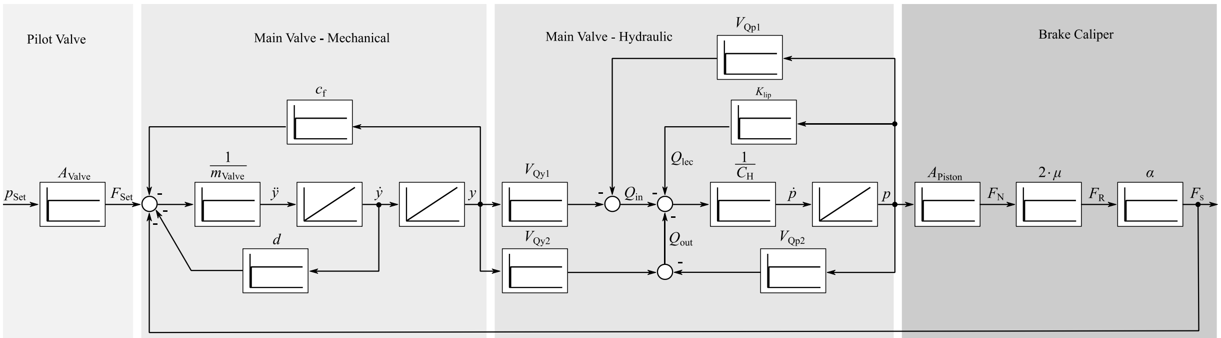


FIGURE 7. Block diagram of the closed loop control system

Equation (7) states the transfer function of the system after linearization.

$$G(s) = \frac{F_{\mathrm{S}}}{F_{\mathrm{Set}}} = \frac{(V_{\mathrm{Qy1}} - V_{\mathrm{Qy2}}) \cdot A_{\mathrm{Piston}} \cdot 2\mu \cdot \alpha}{A \cdot s^3 + B \cdot s^2 + C \cdot s + D} \tag{7}$$

with

$$A = m_{\mathrm{Valve}} \cdot C_{\mathrm{H}} \tag{8}$$

$$B = m_{\mathrm{Valve}} \cdot \left(K_{\mathrm{lecp}} \cdot V_{\mathrm{Qp1}} \cdot V_{\mathrm{Qp2}}\right) + d \cdot C_{\mathrm{H}} \tag{9}$$

$$C = d \cdot \left(K_{\mathrm{lecp}} \cdot V_{\mathrm{Qp1}} \cdot V_{\mathrm{Qp2}}\right) + c_{\mathrm{f}} \cdot C_{\mathrm{H}} \tag{10}$$

$$D = c_{\mathrm{f}} \cdot \left(K_{\mathrm{lecp}} \cdot V_{\mathrm{Qp1}} \cdot V_{\mathrm{Qp2}}\right) + \left(V_{\mathrm{Qy1}} - V_{\mathrm{Qy2}}\right) \cdot A_{\mathrm{Piston}} \cdot 2\mu \cdot \alpha \tag{11}$$

Using Matlab/Simulink, a parameter set is found which leads to stable system behaviour. Nonlinearities such as the volume flow's square root dependence on the pressure difference and spool way limitations are considered in a simulation model in the one-dimensional simulation software DSHplus. **Figure 8** illustrates the implementation of the brake system.

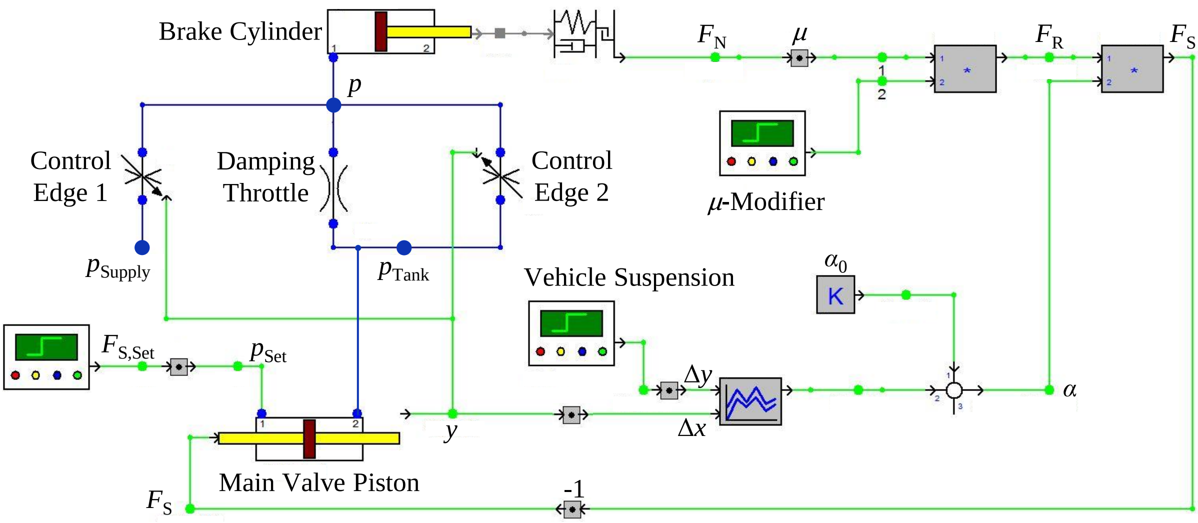


FIGURE 8. Simulation model in DSHplus

The mechanical part of the brake is implemented by using two cylinder models for both the main valve and the brake cylinder. The two chambers of the main valve piston are connected with the set pressure affected by a force demand, and the tank pressure. The supporting force signal F_S is applied on the main valve piston so that the resulting hydraulic force equilibrium balances it. The output displacement y of the main valve piston is connected with both control edges of the valve that represents the hydraulic part of the valve. Both control edges are connected to the same brake piston chamber and to the supply pressure and tank pressure on the other side respectively. When the main valve moves in positive direction control edge 1 opens to increase the braking pressure p while control edge 2 closes. With a negative displacement y the system reacts inversely and the braking pressure decreases. A throttle between braking pressure and tank leads to more damping. The brake piston is mechanically coupled with a spring-damper element to represent material properties of the pads and the disc [4]. Variations of the friction coefficient μ between brake pads and disc and the force factor α are set by additional signal generators.

SIMULATION RESULTS

Figure 9 depicts the simulation results of step responses due to switching force demands (a) and error response provoked by a decreasing friction coefficient μ (b).

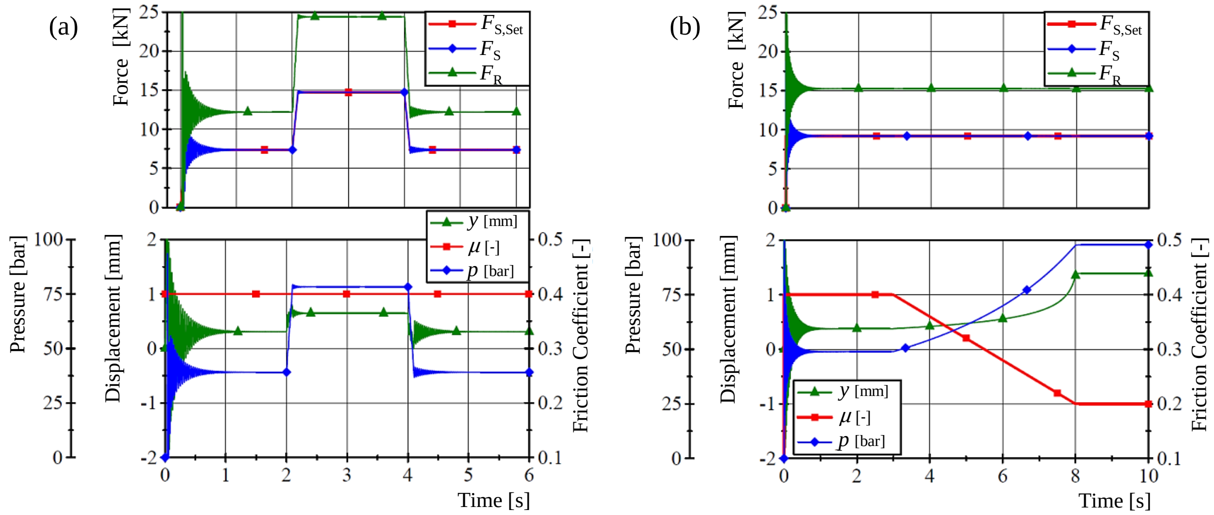


FIGURE 9. System response to (a) changing force demand (step response); (b) changing friction coefficient (error response)

The supply pressure p_{Supply} of 100 bar corresponds to the maximum system pressure of the original brake. According to the force demand $F_{S,Set}$ the valve opens and the pressure p increases. After short settling time the friction force F_R builds up and the current supporting force F_S follows the demand $F_{S,Set}$ precisely. At a higher pressure level the valve opens more to compensate the increasing pressure depending leakage flow. While in this simulation the friction coefficient remains constant, for the investigation of the error behaviour the conditions between brake pads and disc are continuously changing. **Figure 9 (b)** shows the results achieved with a constantly reduced friction coefficient μ to introduce a disturbance to the system. The system compensates the loss of the frictional force F_R directly by opening the main valve and increasing the braking pressure p. As a result, the frictional force F_R and the supporting force F_S remain constant during the whole brake time. Hence, the system behaves as expected and fulfils its purpose as well. **Figure 10** illustrates the results of the dynamic analysis of the system for various levels of the pressure p_{Set}.

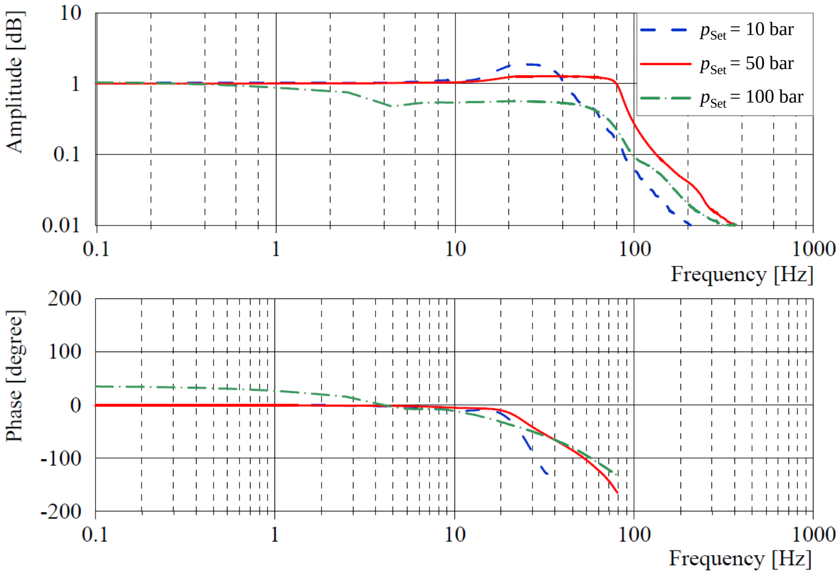


FIGURE 10. Dynamic response of the system at various pressures p_{Set}

Using the Bode diagram, the dependence of the dynamic response on the input values can be observed. Sweeps of increasing frequencies at three distinct pressures p_{Set} (10, 50 and 100 bar) are performed. The supply pressure p_{Supply} is 100 bar again. Due to the nonlinearities of the hydromechanical system the dynamic response and the resonance frequency change between 20 and 80 Hz depending on p_{Set}. While the dynamics of the system decrease as expected with rising set pressure, they are also reduced at very low pressure demands. Main reason for this behaviour is the flow factor V_{Qy} which depends on the pressure difference. Measures to improve the dynamics for this operation area would be a smaller valve opening or the reduction of the supply pressure.

PRELIMINARY DESIGN

An innovative design is created, that integrates the sensing tool of the feedback signal F_S and the control mechanisms of the brake pressure into the supporting structure. One of the aspects that needs to be considered while designing, is the direction of the supporting force F_S which changes with the direction of the vehicle wheels' rotation from forwards to backwards. Therefore, the capture mechanism has to be able to indicate the supporting force F_S for both directions. It is also essential that, independent of the direction of the supporting force, an increase in the supporting force F_S is opposed with a decreased opening y of the valve, connecting the brake chamber to the tank and closing the supply pressure. For this function the relative motion between the valve sleeve and the valve spool must be in the same direction independently of the supporting force direction. **Figure 11** schematically shows the concept of the bidirectional supporting force capture mechanism for increasing supporting force F_S.

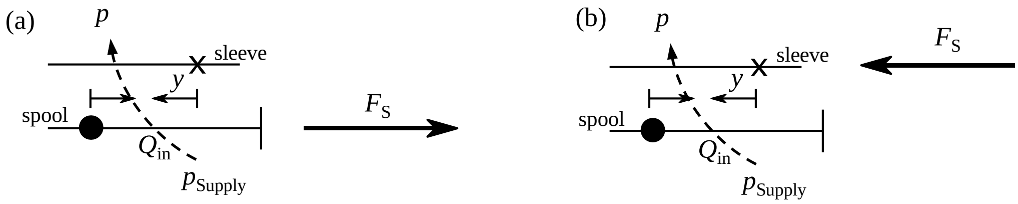


FIGURE 11. Relative displacement in case of (a) pulling or (b) pushing supporting force F_S

Figure 12 presents the main components of the valve integrated into the supporting structure in completely extended position for a pulling supporting force F_S.

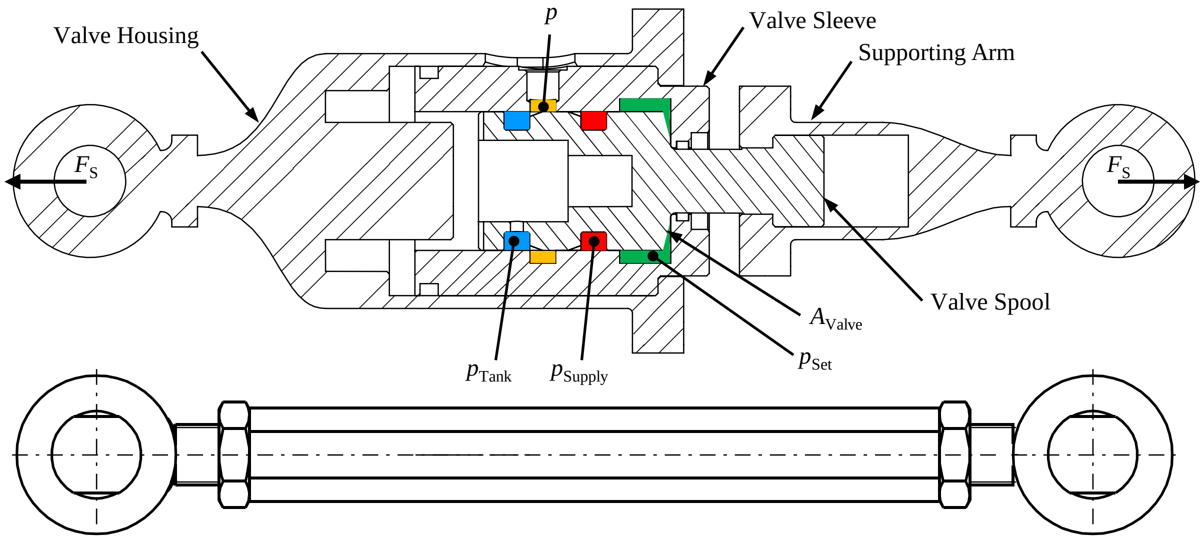


FIGURE 12. Preliminary design (above) and original supporting structure (bottom)

The valve mainly consists of four parts:

1. The valve spool with control grooves.
2. The valve sleeve that forms the metering edges as variable orifices with the valve spool. Additionally, it contains the ports for the four needed hydraulic connectors.
3. The supporting arm which is connected to the brake caliper.
4. The valve housing which is coupled with the vehicle bogie.

In pulling direction the supporting force F_S moves the supporting arm and the valve spool to the right side. Due to the set pressure p_{Set} and the area A_{Valve}, valve spool and sleeve are moving relative to a position where the metering edges between the supply, braking, and tank pressure achieve an equilibrium with the current supporting force F_S. In pushing direction the supporting arm moves left up to the valve sleeve and pushes it further - relative to the valve spool - until an equilibrium with the hydraulic force due to the pressure p_{Set} and the area A_{Valve} is reached again.

CONCLUSION AND OUTLOOK

Based on a conventional hydraulic brake system for commuter trains an advanced brake with hydromechanical brake torque control is presented. The force transmitted through the brakes supporting structure is identified and experimentally verified as representative value for the current brake torque. Simulation results show the functionality and good dynamics of the concept which is elaborated in a preliminary design.
Future research lays the focus on completion of the design process and building prototypes for an experimental verification of the concept. After gaining experience of this innovative brake system on a test rig a commuter train will be equipped with prototypes for extended field testing.

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