DEVELOPMENT OF A VALVE FOR A BRAKING-TORQUE CONTROL USED ON RAILWAY BRAKES

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A novel hydraulic disc brake system with a closed-loop control of the braking torque is under development at the Institute for Fluid Power Drives and Systems (ifas) of RWTH Aachen University. During the braking process, the torque on the caliper acts on a supporting rod. It has been found that the resulting supporting force is proportional to the braking torque. Thus, this force is utilized as a feedback variable for a hydro-mechanical control unit.

In this paper, a simulation model of a spring-applied disc brake is introduced and the behavior of this brake is investigated. Based on those results, necessary requirements on the control unit are deduced and a suitable prototype for the investigation on a test rig is developed. Finally, the behavior of the control unit is analyzed using the simulation model.

KEYWORDS

spring-applied disc brake, hydro-mechanical braking-torque control, valve design, railway application, simulation

1 INTRODUCTION

The friction coefficient in the contact between brake pads and brake disc is subjected to various influences such as temperature, wear, humidity or frozen friction contacts [Lee2017]. Currently, open-loop braking systems are commonly in use and only passive countermeasures, such as optimization of the friction materials, are taken [Breuer2017].

Within a research project at if as, a self-energizing hydraulic brake for railway vehicles (SEHB) was developed. Petry et. al. [Petry 2016] presented a closed-loop control for the braking torque of the SEHB and described the potential and advantages of a closed-loop brake system. Based on that, a closed-loop control of a conventional brake was introduced [Petry 2017]. It has been found in a preliminary study that the supporting force of the brake caliper is proportional to the braking torque. Thus, it has been proposed to utilize the supporting force as a feedback variable for a hydro-mechanical control unit. The control unit can be implemented by integrating a hydraulic valve in the supporting rod. This concept has been further investigated by Bordovsky et al. [Bordovsky2018].

So far, the theoretical research has been done and only active brake systems have been regarded. Even though, spring-applied brakes are often preferred to active brakes due to their inherent fail-safe behavior. An application of the closed-loop braking torque control for a spring-applied brake is still missing.

This paper applies the concept of a closed-loop control of the braking torque with the supporting force as a feedback variable to a spring-applied disc brake. Furthermore, a prototype valve for an upcoming experimental verification on a test rig and on a reference vehicle is developed.

2 CONVENTIONAL BRAKE SYSTEM

In this paper, a novel concept of a hydro-mechanical braking-torque control is analyzed on an existing disc brake HYS 258 by the company Hanning & Kahl GmbH & Co KG (see Fig. 1). It is a hydraulic spring-applied brake system with a floating caliper, which is mounted on a bogie via a sliding pivot joint B and a supporting rod connected to the pivot joint A_0. Such systems are used in low-floor vehicles as fail-safe brakes since they provide braking power by themselves without any other input energy.

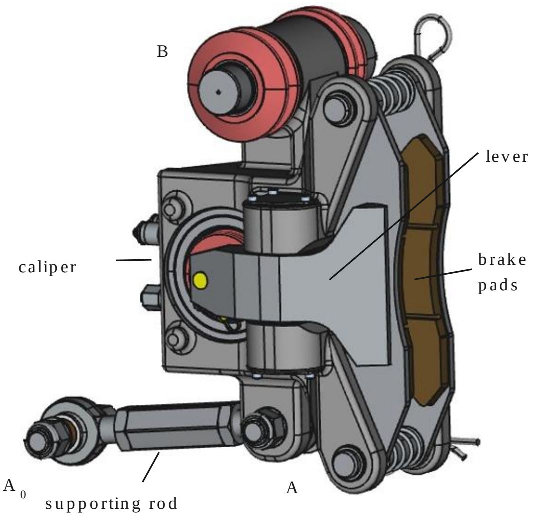

Figure 1. Original brake system

Fig. 2 shows a functional schematic of the original brake system. When closing the brake by reducing the braking pressure p_B, the brake pad I is moved against the brake disc. Then, the brake pad I pulls the caliper including the brake pad II against the opposite side of the brake disc, which results in a braking force, respectively a braking torque. This torque acts on the brake disc as well as on the caliper and is taken up by the supporting rod.

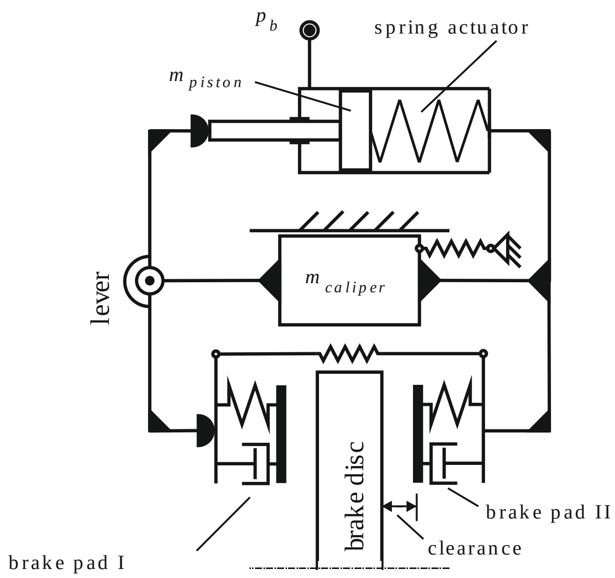

Figure 2. Functional schematic of the original brake system

2.1 Simulation model

A simulation model of the original brake is implemented in the software AMEsim to assess the system behavior. Using the three-dimensional mechanical library, a simplified kinematic model of the brake system is built based on the functional schematic shown in Fig. 2. The hydraulic system is simplified for the purpose of the simulation. An ideal pressure source is assumed and a proportional pressure reducing valve, which controls the pressure in the hydraulic actuator, is modelled as a first-order-delay element with an orifice. The valve characteristic is implemented according to the manufacturer data. A pipe, which connects the valve to the hydraulic actuator, is modelled by considering its resistance and capacitance. The model of the hydraulic actuator is composed of a mass-envelope module, which represents the piston and the caliper masses respectively, a return spring, and a piston with a moving body.

Compared to the similar model presented in the publication [Bordovsky2018], this model is extended by a complex braking-torque calculation. In previous work, e.g. [Petry 2017], it has been assumed that the friction force acting between brake pads and the brake disc acts in the center of their contact area and that there are no additional forces or torques transmitted. However, in reality, friction forces arise over the whole contact area and act in the direction of the local friction velocity. Thus, it is necessary to integrate the local friction torque over the whole contact area to evaluate the overall torque. This effect has been described for example by Haag [Haag 2012]. Though, he disregarded the translational relative movement between the brake disc and brake pads. For the regarded system, such movement is generated by the suspension of the vehicle. Since this has to be taken into account for the investigated brake system, Haag's formulas are extended by corresponding terms.

Fig. 3 illustrates friction velocities acting on a pad-disc contact area for a displaced brake disc related to a fixed brake pad. In any point on this area, the friction velocity is composed of the rotational velocity component $\vec{v}_{rot}$ and the translational velocity component $\vec{v}_t$ of the brake disc. The center of the brake disc $M'$is displaced from its ideal positionM_0 in the direction of x and z-axis by distancesM_x'andM_z'.

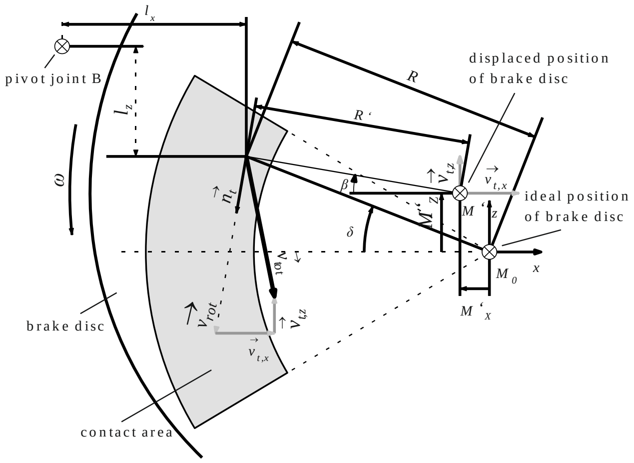

Figure 3. Friction velocity acting on a pad-disc contact area for a displaced brake disc

The local torquet_B related to the pivot joint B can be expressed according to eq. (1)

$$t_B = p \cdot \frac{\mu}{|\vec{v}_{tot}|} \cdot \left(l_x v_{tot,z} + l_z v_{tot,x}\right) \tag{1}$$

where p is the local contact pressure, μ the local friction coefficient, $l_x$and l_z are the local levers related to the point B. Furthermore, $v_{tot,x}$and $v_{tot,z}$ are the local x and z-components of the vector field$\vec{v}_{tot}$, which is composed of the rotational and the translational velocity fields $\vec{v}_{rot}$ and $\vec{v}_t$ as follows

$$\vec{v}_{tot} = \vec{v}_{rot} + \vec{v}_t = -R'\omega \begin{pmatrix} \sin\beta \\ \cos\beta \end{pmatrix} + \begin{pmatrix} v_{t,x} \\ v_{t,z} \end{pmatrix} \tag{2}$$

with the angle β

$$\beta = arctan\left(\frac{R\sin\delta - M_z'}{(R\cos\delta + M_x')}\right) \tag{3}$$

and the local friction radius R'

$$R' = \sqrt{(R\sin\delta - M_z')^2 + (R\cos\delta + M_x')^2}. \tag{4}$$

The overall torque $T_B$related to the pivot joint B can be calculated by integrating the local torque t_B over the whole pad-disc contact area.

$$T_B = \int t_B \, dA = \int\int t_B \cdot R \; dR \, d\delta \tag{5}$$

The local torque on the brake disc $t_{M'}$ can be written as

$$t_{M'} = \frac{\vec{v}_{tot} \cdot \vec{n}}{|\vec{v}_{tot}|} \mu \, p \, R' \tag{6}$$

with the unit normal vector $\vec{n}$

$$\vec{n} = \begin{pmatrix} -\sin\beta \\ -\cos\beta \end{pmatrix} \tag{7}$$

Finally, the overall torque$T_{M'}$is obtained usingeq. (8).

$$T_{M'} = \int t_{M'} dA = \int \frac{\vec{v}_{tot} \cdot \vec{n}}{|\vec{v}_{tot}|} \mu \, p \, R' \, dA \tag{8}$$

Because no analytical solution was found for those expressions, they were numerically solved. Since this could not be done directly in the software AMEsim, the calculation was implemented using the modelling language Modelica within AMEsim.

2.2 *Behavior of the open-loop system*

Main influences and disturbances affecting the braking process are identified using the simulation model. To examine the behavior of the original open-loop system, a parametric study is set up using the Monte-Carlo approach. The following variables are evaluated: the maximum braking torque, the brake reaction time, and the ratio between the braking torque and the supporting force (hereinafter referred to as factor α). The braking torque is normalized with the mean braking torque obtained from the Monte-Carlo run.

Fig. 4 – 6 depict results of the parametric study in the form of response surfaces. Two major influences on the maximum braking torque are identified. As expected, the friction coefficient has the main influence on the braking torque (see Fig. 4). Furthermore, the braking torque is lowered by a rising clearance b between brake pads and the brake disc because the spring actuator has to overcome this clearance before generating a braking force. Due to this fact, the reaction time of the system is affected as well, as shown in Fig. 5. The axis of b is shown reversed in order to depict the response surface. Moreover, the oil temperature has a large impact on the reaction time of the system as the maximum flow rates decrease with an increasing viscosity at lower fluid temperatures.

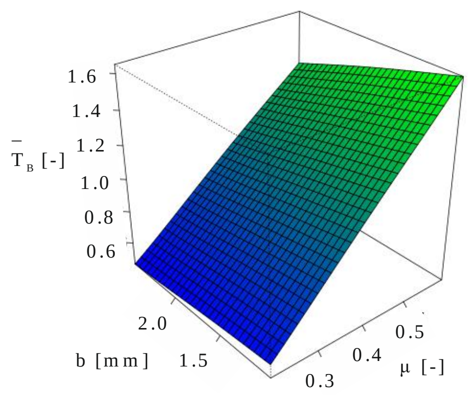

Figure 4. Response surface of the normalized braking torque $\bar{T}_B$as a function of the clearance b and friction coefficient μ

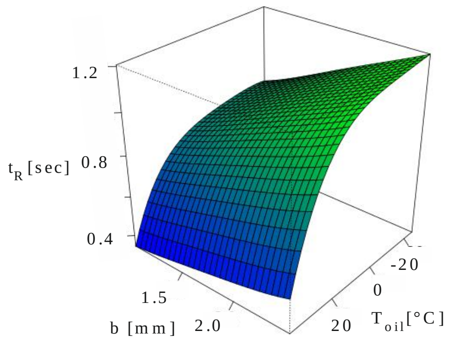

Figure 5. Response surface of the reaction time t_R as a function of the clearance b and the oil temperature T_{oil}

As can be seen in Fig. 6, the factor α greatly depends on the vertical M_z disc displacement. It results from the extended calculation of the braking torque as introduced above. Consequently, this finding significantly differs from the results found in previous works [Bordovsky2018] and [Petry 2017].

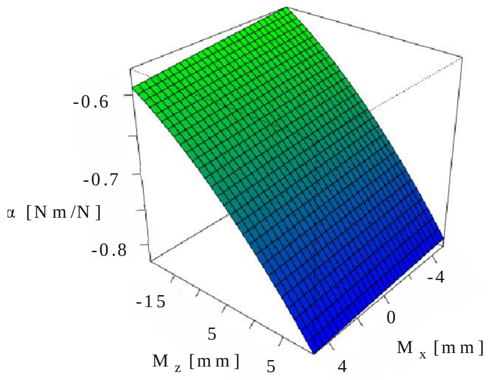

Figure 6. Response surface of the factor α as a function of the horizontal M_x and the vertical M_z disc displacements

For the calculation of the friction torques, some assumptions are made. For instance, a uniform pressure is assumed over the pad-disc contact area. In reality, the pressure can be larger in the center of the contact-area because the pads can deform due to the clamping force applied and due to the buckle of the brake pads resulting from the heat generated during braking. Both influences may lead to a redistribution of the local friction forces to the center of the contact area, and hence to a reduction of the influence of the displacements. Since those effects could not be estimated in a satisfying manner, measurements are necessary to evaluate their scope.

3 NOVEL BRAKE SYSTEM

3.1 Development of a control concept

The concept of the closed-loop braking-torque control has been introduced in previous works[Petry 2017] and [Bordovsky2018]. Therefore, it is only briefly described here. The supporting force along the supporting rod is utilized as a feedback variable for the closed-loop control. The braking torque is controlled by a hydro-mechanical unit, which represents a prototype valve consisting of spool, sleeve and body. Those parts are moved against each other in order to keep an equilibrium between the set and the actual braking torque.

To realize a closed-loop control, the following sub-functions are necessary. The actual variable has to be fed back, the set variable and the actual variable have to be balanced, and the control variable has to be applied. Within this study, the feedback variable is the supporting force, the set variable is a set pressure and the control variable is the braking pressure, i.e. the pressure in the brake actuator. Since the supporting force can either be a tensile or a compressive force, depending on the direction of the vehicle motion, a rectification as a fourth sub-function is implemented.

Based on this, a mechanical rectification is implemented, which enables a compression of the spring at compressive and tensile forces. This can be achieved by varying the active areas for the force transmission, as illustrated in Fig. 7. In the upper case, a compressive force acts on the structure. Therefore, the left joint acts on the main body and the right joint presses against the central rod to the left. In the lower case, a tensile force is acting. Therefore, the active interfaces change. Now the right joint acts on the main body and the left joint pulls on the central rod to the left. Thus, in this example, the spring is always subjected to compressive stress, independent of the direction of the acting forces.

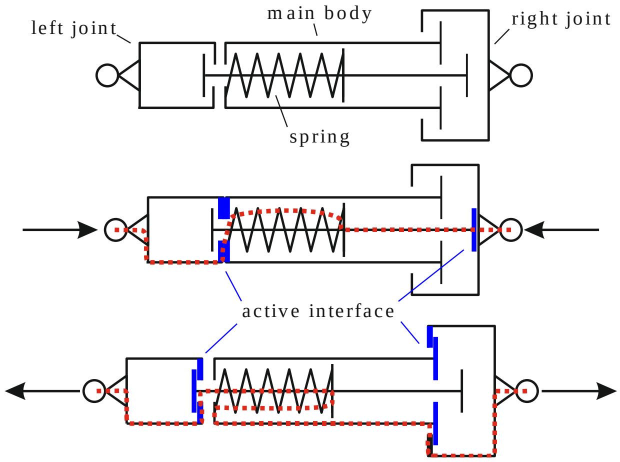

Figure 7. Concept of a mechanical rectification by variation of the active area

Since the analyzed brake system is of a spring-applied type, a lower braking pressure leads to higher braking torques, and thus to higher supporting forces. To realize a force balance on the valve spool, the supporting and the set pressure force are summed up and act against a spring. A disturbance leads to a spool displacement, which results in a connection of the brake-actuator port with either the pressure-supply or the return-line port depending on the difference between the set pressure and the supporting force. The principle of the control unit is depicted in Fig. 8.

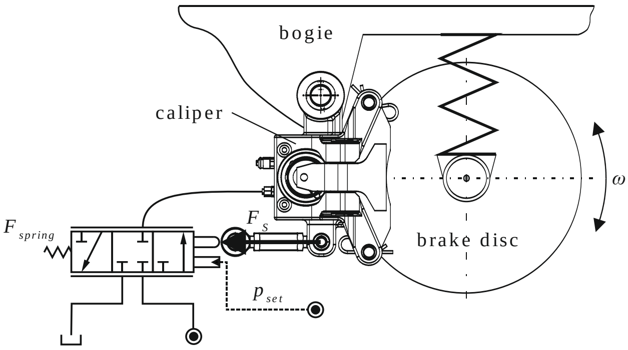

Figure 8. Schematic of the closed-loop brake system

3.2 Parametrization of the control unit

The characteristic between the set pressure and the normal force of the conventional brake is used for the parametrization of the control valve. For a set pressure $p_{set} = 0$ bar, the brake is completely closed. On the control unit, the spring has to keep the valve completely open to the tank against the maximum possible supporting force. Thus, the spring pre-tension for the maximal spool displacement y_{max} has to be equal or greater than the maximum supporting force.

$$F_{Spring}(y = -y_{max}) \geq F_{S,max} \tag{9}$$

Furthermore, the brake is barely closed for a set pressure $p_{set} = p_{th}$ acting on the valve area A_V. Therefore, no supporting force is acting and the spool is balanced in its mid-position.

$$F_{Spring}(y = 0) = A_V \cdot p_{th} \tag{10}$$

Finally, if the maximum set pressure $p_{set} = p_{max}$ is applied, the valve fully opens to the pressure supply to completely unclamp the brake.

$$F_{Spring}(y = y_{max}) \leq A_V \cdot p_{max} \tag{11}$$

When assuming a constant spring stiffness, the necessary spool area A_V can be obtained as follows

$$A_V \geq \frac{F_{S,max}}{2 \cdot p_{th} - p_{max}}. \tag{12}$$

The maximum supporting force $F_{S,max}$ can be found from a torque equilibrium on the caliper. Subsequently, either a maximum spool displacement y_{max} or a spring stiffness can be chosen. Since standard springs are only available with certain stiffnesses, while the maximum displacement can be chosen rather freely, a suitable spring is selected and the displacement is set according to the spring characteristic. In order to achieve limited displacements of the caliper, Belleville springs are selected for this concept because they feature a compact design and a large stiffness.

To achieve a suitable volume-flow-signal gain for the valve, metering notches have to be machined on the metering edge of the spool. Their geometry and size strongly influence the valve performance. The geometry defines the volume-flow-signal gain depending on the spool displacement. Typical forms are triangular, rectangular and cylindrical notches. In case of a rectangular notch, the gain linearly rises with the displacement, while cylindrical and triangular notches feature progressive gain characteristics, which allow for a fine metering around the valve mid-position [Findeisen 2015]. In this case, the spring significantly restricts spool displacements due to disturbance variables in the control loop, and thus limits the disturbance reaction. To avoid this, the gain must not be too low for small displacements. Therefore, rectangular notches are chosen for this application.

Finally, the volume-flow-signal gain is optimized to achieve a robust control behavior and acceptable system dynamics. For this purpose, a genetic algorithm is implemented in the software AMESim. The goal of the optimization is to minimize the integral square error (ITSE) of a step response.

3.3 Final design

Fig. 9 shows the final design of the control unit. It replaces the supporting rod of the original system. On the spool, the supporting force, the set pressure and the force generated by the spring are in balance. Due to the mechanical rectification, the supporting force acts on the spool always in the same direction. In this figure, it acts on the right side.

In order to reduce leakage, the spool is sealed with O-rings. Note that components such as fittings, seals, joints and the spring are not shown.

3.4 Behavior of the closed-loop system

The behaviorof the closed-loop system is analyzed based on a state-space representation, which is obtained from the mathematical model of the normal force and of the controller (not described in this paper). First, the stability and then the dynamic behavior are investigated. Subsequently, a parametric study is conducted for the closed-loop system to assess its behavior by evaluating the normalized braking torque and the reaction time.

The state-space representation is implemented as a script in the software Matlab. For the parameters derived from the design and stored in the script, the requirement for exclusively negative real parts of the eigenvalues of the system matrix is fulfilled. Thus, the linearized system is stable.

Using the state-space representation, the frequency behavior of the system is analyzed. The behavior of the closed-loop system is finally shown in bodeplots (Fig. 10 and Fig. 11). In case of a change of the set variable, the large gain of the system results from the mathematical description. The input signal is normalized to compare the original system, whose input signal is current, with the closed-loop system, whose input signal is pressure. On the other hand, the system response is not normalized.

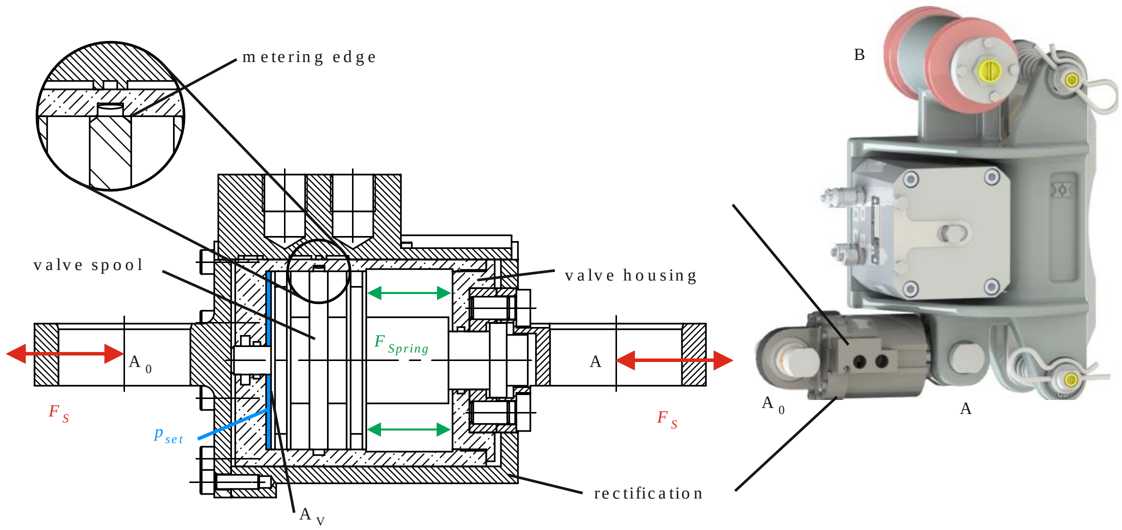

Figure 9. Novel brake with the control unit

In case of a change of the set variable, the first peak of the magnitude at approximately 90 Hz results from the natural frequency of the control unit. The second peak of the magnitude with a subsequent drop at about 700 Hz is caused by the proper motion of the brake caliper and the piston against the brake pads. Since the damping of the brake pads is unknown, and it is assumed to be very low for the calculation of the frequency responses, it can be concluded that this peak is significantly weaker in reality.

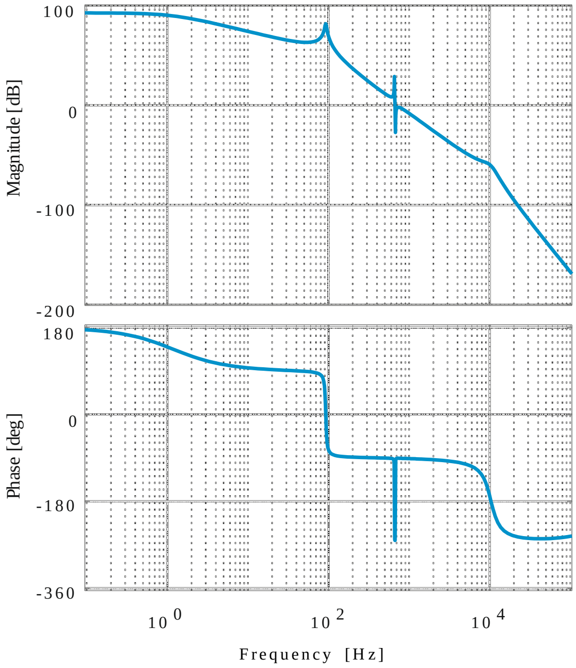

Figure 9. Frequency behavior of the closed-loop system subjected to a change of the set variable

It can be seen in Fig. 11 that the disturbance variables are first damped. However, the attenuation decreases sharply with an increasing frequency. Above 5 Hz, there is no attenuation present. A small increase of the natural frequency of the control unit can be seen at a frequency of approximately 90 Hz.

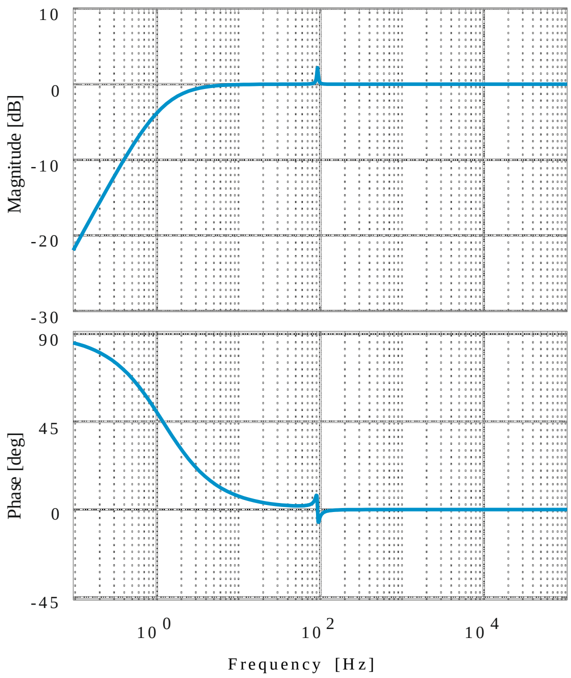

Figure 10. Frequency behavior of the closed-loop system subjected to a change of the disturbance variable

For the parametric study, the braking torque is again normalized with the mean braking torque of the study. The results of the parametric study are shown in Fig. 12 and Fig. 13.

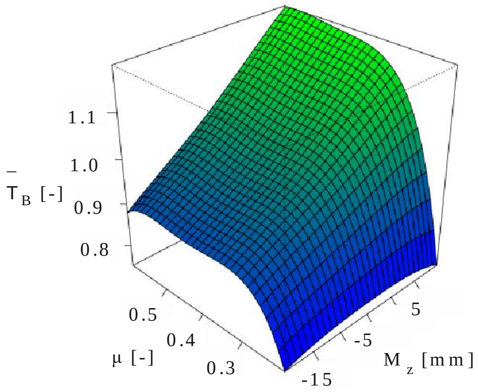

Figure 11. Response surface of the normalized braking torque $\overline{T}_B$ over vertical disc displacement M_z and friction coefficient μ

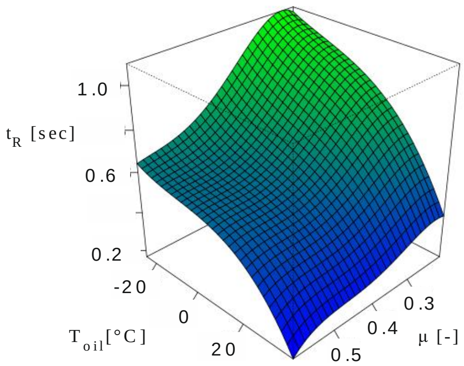

Figure 12. Response surface of the reaction time $t_R$over the oil temperature T_{Oil} and the friction coefficient μ

Compared to the open-loop system, influences of disturbances on both responses of the closed-loop system changed. The braking torque remains constant for a large range of the friction coefficient.

Based on the influences on the factor α, the generated braking torque depends on the vertical displacement of the brake disc. Though, compared to the conventional open-loop system, the braking torque variation is significantly reduced (compare Fig. 4). Nevertheless, the impact of the vertical disc displacement on the braking torque will be experimentally investigated. If necessary, compensation will be carried out.

The oil temperature still impacts the reaction time. There is an additional dependency on the friction coefficient since lower friction coefficients have to be compensated by a higher clamping force. A direct comparison of the reaction time of the conventional and the novel braking system is difficult since the main influences are now different. For a given working point, the reaction time of the closed-loop system is a bit lower.

4 CONCLUSIONS

In this paper, a novel prototype valve for a hydro-mechanical brake torque control was introduced. The behavior of the open-loop and the closed-loop system was analyzed with the help of a parametric study. Based on the results, the introduced closed-loop control is able to compensate for the influence of a variable friction coefficient on the braking performance.

It was found that the vertical displacement of the brake disc has a major influence on the control loop. This effect was considered already in the design phase by extending the calculation of the braking torque.

Future work will include the manufacturing of the prototype valve. First, the prototype will be examined on a test rig. Then, a reference vehicle will be equipped with the novel brake system to investigate its brake performance under real operating conditions.

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