Hydro-Mechanical Closed-Loop Brake Torque Control for Railway Disc Brakes

Contribution to the topic novel vehicles and components

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Zusammenfassung

The generated braking torque of disc brakes is highly dependent on the tribological contact between brake disc and brake pads. Since the actual friction forces are subjected to various disturbances, the resulting brake effect is similarly influenced. Common countermeasures try to optimize the friction pairing to limit the deviation. Within an ongoing research project at the Institute for Fluid Power Drives and Systems (ifas) of RWTH Aachen University, a novel approach is taken. To compensate the disturbances, an innovative hydraulic disc brake with closed loop control of the braking torque is being developed. This allows a higher utilization of the physical braking potential, an improved and more reliable braking process and possibly the use of cheaper, less optimized friction pairings. In this project, the closed loop control is achieved by a hydro-mechanical control unit embedded in the supporting structure of a conventional hydraulic disc brake, using the supporting forces of the caliper as feedback variable for the control unit. Due to the hydro-mechanical implementation of the control, a highly robust and reliable control system can be realized. In the scope of this article, the dynamic brake behavior of the novel closed loop system is closely regarded and suitable models for the description are introduced. A control concept is researched and the resulting prototypes are presented.

Keywords: disc brake, hydraulics, closed loop control, braking torque, valve design

1 Introduction

An intrinsic problem of all friction based brakes and thus disc brakes, is their dependency on the friction coefficient in the contact between brake pads and disc. Since the actual friction coefficient is highly dependent on the operating parameters, the brake performance and especially the reliability of the braking process is limited. Typical disturbances are changing temperatures in the tribological contact, wet or frozen contact areas, manufacturing flaws or even vitrification of the brake pads [5]. Usually, for conventional brake systems, it is tried to limit the disturbances by optimizing the friction pairing. Much effort is spent in trying to find friction pairings with ideally constant friction properties. To the day, no ideal friction pairing has been found and while there have been significant advances, the problem still limits the achievable brake performance. [2] In order to further increase the performance and reliablity of the braking process, novel approaches need to be applied. A basic solution to the problem is the implementation of a closed-loop braking torque control. Unfortunately, a direct measurement of the braking torque on the brake disc is technically demanding and expensive. Further on, to achieve higher reliability, robustness and predictability, in safety-critical applications such as brakes, a purely mechanical system is preferable.

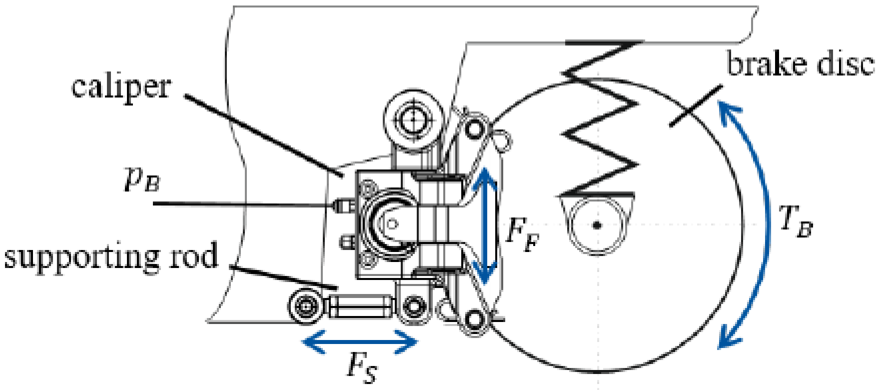
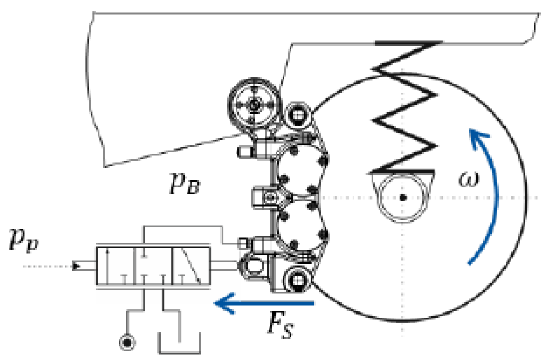


Figure 1: Acting forces on a typical disc brake

At ifas, a hydro-mechanical closed-loop control system is being researched. In previous works, it has been found that the supporting force F_s of the caliper is proportional to the acting braking torque T_B on the brake disc, since the friction forces F_F act on both components (fig. 1). The supporting force is therefore suitable as a feedback variable for the implementation of a closed-loop braking torque control. Using the supporting force as a feedback variable allows a measurement on fixed parts of the brake and solves the problem of a data transfer from a rotating shaft to a fixed control unit. The use of the supporting force as a feedback variable for an active brake torque control has been researched in previous research projects on the Self-Energizing Hydraulic Brake (SEHB) invented at ifas [6, 7]. So far, the regarded control systems were strictly electronic. Furthermore the SEHB is a highly innovative brake type with low public awareness and acceptance. In the ongoing research project, a purely hydro-mechanical closed-loop control for conventional hydraulic disc brakes is being researched. Thus, a higher reliability and a better integration in existing brake systems can be achieved. The main objective of the control system is to compensate external disturbances on the braking torque and thus mainly on the friction coefficient. Those are typically friction speed and temperature, pressure, wetness or vitrification of the brake pads. Those disturbances do not have high dynamics [3, 9], so the implemented controller does not need to be fast either. There are of course disturbances with higher dynamics, such as disc thickness variations. Those are explicitly not focused in this research and will not be compensated by the proposed control system. Compensation of highly dynamic disturbances has been analyzed in [6] and [7]. It could be shown that a simple proportional controller was not able to achieve the necessary level of control. Finally, to achieve satisfying results, a predictive status control was applied. Advanced control strategies like this are not achievable with a hydro-mechanical control system and will have to be left to electronic control systems.

2 Novel brake system

The novel brake system consists of a conventional disc brake and a novel control unit. This way, compatibility to existing brake systems can be achieved.



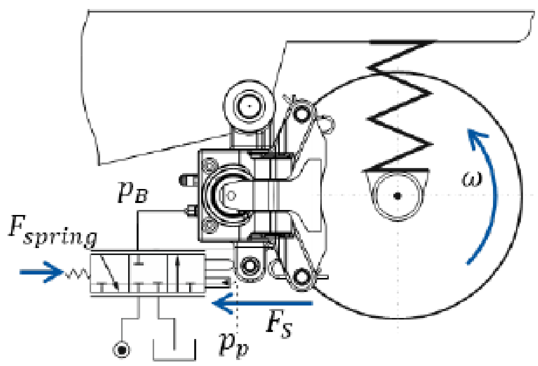


Figure 2: Suitable control valves for an active and a passive brake system

For the control unit, a distinction has to be made between a control unit for an active brake (fig. 2(a)) and one for a passive brake (fig. 2(b)). An active brake system is closed by a rising pressure in the hydraulic cylinder of the brake. These brake systems are very compact and highly dynamic, but they are not fail-safe. Without further safety measures an active brake system will fail in case of a loss of the hydraulic power supply. In contrast, passive brake systems are closed by a prestressed spring and kept open by a pressure in the braking cylinder. Such a brake closes if the power supply fails. Thus it is inherently fail-safe. On the downside, the necessary spring accumulator builds significantly bigger than a single cylinder. The initial control system design has been proposed in [8]. A control valve is suggested, which compares the supporting force F_s with a pilot pressure p_p and supplies the actual braking pressure p_B as a regulating variable to adjust the braking torque according to the pilot pressure. As shown in figure 2, the control valve can be implemented directly into the supporting structure of the brake system.

2.1 Control units for active and passive brake systems

Looking at figure 2 it can be seen that a change in the vehicle's driving direction results in a switch of the direction of the supporting force Fs. Therefore, without the use of a rectification, a closed-loop control system with the valve described above would have a positive feedback for one driving direction, leading to an unstable, self-enforcing system.

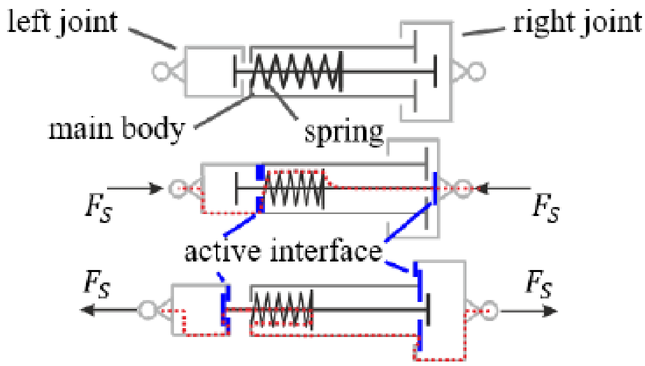
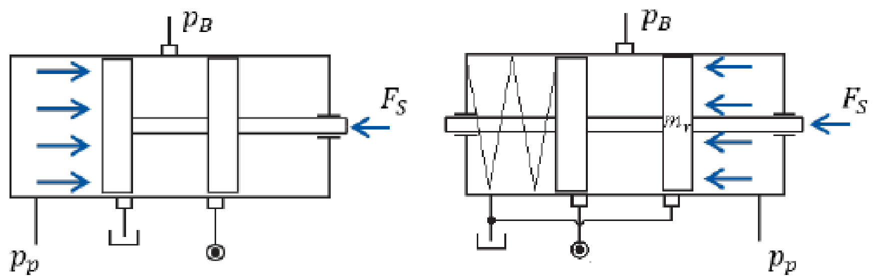


Figure 3: Concept for the mechanical rectification

Such a behavior is undesirable and a rectification needs to be implemented. For this application, a mechanical rectification is the most promising concept. The concept is depicted in figure 3. By variation of the active interfaces, the spring in the sketch is always kept under compressive stress independent of the direction of the acting forces. In the top view of the figure, no external forces act on the structure and the separate parts can be seen. Under compressive stress, the left joint is on block with the main body and the right joint pushes on the spool. Under tensile stress, the acting interfaces change. Now, the left joint pulls on the spool and the right joint is on block with the main body.



(a) Valve for active brake (b) Valve for passive brake

Figure 4: Schematic of the control valve for an active and a passive brake

The concept for a control valve for an active brake is shown in figure 4(a). To realize a control, the force from the pilot pressure pp and the supporting force of the brake F_s can be directly balanced on the spool of the valve. If the force equilibrium is disturbed, the spool of the valve is displaced and the pressure in the brake p_B is controlled accordingly. In case of the passive brake, the necessary control valve is slightly more complex. In order to maintain the passive brake's fail-safe behavior, a similar fail-safe concept has to be implemented in the control valve. In this case the valve has to open by a secondary force if the hydraulic power supply fails. To maintain an easy integration in existing systems, the valve has to retain the pressure-to-braking force characteristics of the original passive brake. This leads to the basic valve schematic depicted in figure 4(b). Both, the supporting force of the brake F_s, as well as the force resulting from the pilot pressure p_p act in the same axial direction of the spool. A prestressed spring counterbalances the forces. This valve will be opened by the prestressed spring if the pilot pressure is lost. At the same time, the pressure-to force characteristic of the passive brake is kept. From those considerations, prototypes for the control units can be derived. A sketch of the prototype for the active brake system is shown in figure 5. For the passive brake, a prototype for a control unit based on the preliminary design in [8] has been introduced in [1].

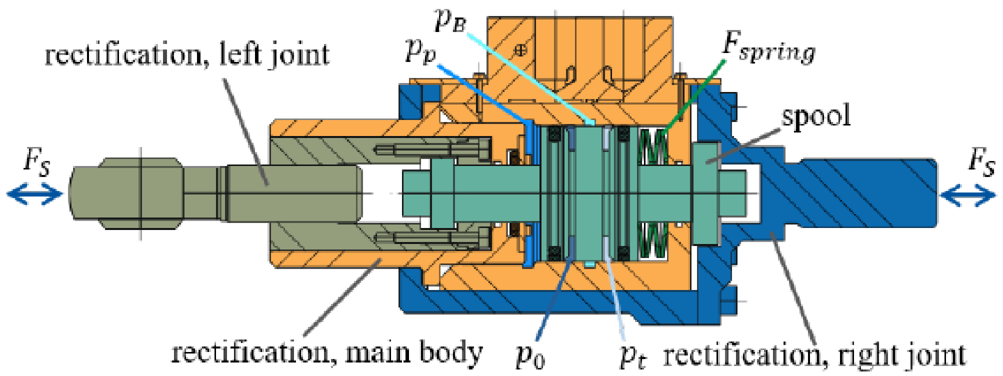


Figure 5: Cross section of the prototype of the control unit for the active brake system

3 Modeling of the closed-loop control

For both control systems the dynamic modeling is quite similar. Therefore only the active system will be presented in detail. The whole system consists of two parts: the novel control unit and the conventional brake system. Both parts can be modeled separately. Afterwards both models are combined.

3.1 Conventional brake system

First, the dynamics of the actual brake need to be regarded. They can be described by the substitute model in figure 6. System input is the flow into the brake Q_B and output is the supporting force Fs. The flow Q_B causes a pressure build-up in the cylinder. To describe the pressure build-up, it is assumed that the dead volume in the hydraulic circuit is significantly bigger than the volume change caused by the movement of the cylinder. This assumption is plausible, since the cylinder in the actual system has a very short stroke length - only the clearance between pads and disc has to be covered. In this case, the hydraulic capacitance C_H is independent of the position of the piston and thus constant.

$$C_H = \frac{\delta V}{\delta p} = \frac{V_0}{E} \tag{1}$$

Thereof, the pressure buildup in the cylinder can be described with the sum of all flows into the cylinder $\sum Q_B$ and the movement of cylinder and piston $\dot{x}_c$ and $\dot{x}_p$.

$$C_H \cdot \dot{p}_B = \sum Q_B + A_p(\dot{x}_p - \dot{x}_c) \tag{2}$$

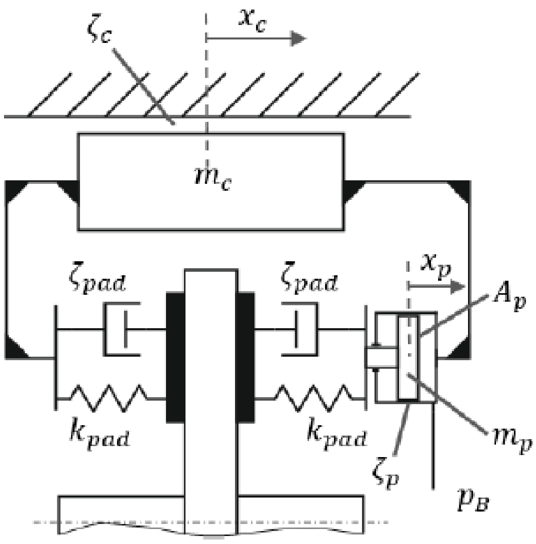


Figure 6: Dynamic model of the active brake

For a closed brake, the brake dynamics can be described with the following equations. Force equilibria for the caliper and the piston:

$$m_c \cdot \ddot{x}_c = -F_{pad,l} + p_B \cdot A_p - \zeta_C \cdot \dot{x}_c - \zeta_p(\dot{x}_c - \dot{x}_p) \quad (3)$$

$$m_p \cdot \ddot{x}_p = F_{pad,r} - p_B \cdot A_p - \zeta_p(\dot{x}_p - \dot{x}_c) \quad (4)$$

The material characteristics of the pads are modeled with a simple Voigt element. This leads to the normal forces on the pads $F_{pad;l}$ and $F_{pad;r}$.

$$F_{pad,l} = k_{pad} \cdot x_c + \zeta_{pad} \cdot \dot{x}_c \qquad \forall \quad x_c \geq 0 \quad (5)$$

$$F_{pad,r} = -k_{pad} \cdot x_p - \zeta_{pad} \cdot \dot{x}_p \qquad \forall \quad x_p \leq 0 \quad (6)$$

As shown in [8, 1], the supporting force can be regarded as directly proportional to the braking torque and thus to the normal forces on the brake pads. Disturbances on the proportionality as described in [1] are omitted at this point. In this case, the supporting force is a function of the normal forces on the brake pads $F_{pad;l}$ and $F_{pad;r}$.

$$F_s = K_S\left(F_{pad,r} + F_{pad,l}\right) = K_S\left(k_{pad}(x_c - x_p) + \zeta_{pad}(\dot{x}_c - \dot{x}_p)\right) \quad (7)$$

3.2 Control unit

The control unit consists of two parts: the mechanical rectifier and the control valve. The mechanical rectifier is only active if there is a switch in the direction of movement of the vehicle. During a standard braking process, no changes of the direction of movement occur. Therefore, the rectification is inactive during the braking process and does not need to be regarded in the dynamic modeling. A special case is the stopping process on a rising slope. Here, the direction of the braking torque changes in the moment of stand-still due to gravity. During such a switch, the rectification will move and no supporting force act on the valve. This is has the effect of a disturbance that reduces the braking torque, the brake will close stronger. Since this effect will only occure during stand-still it can be omitted at this point in order to apply linear control theory. The valve dynamics can be described with the model shown in figure 7. The system input is the pilot pressure pp and the supporting force F_s. The output is the flow Q_B. Additionally, Q_B is dependent on the acting braking pressure p_B which can be regarded as a disturbing variable at this point. The supporting force Fs, the force resulting

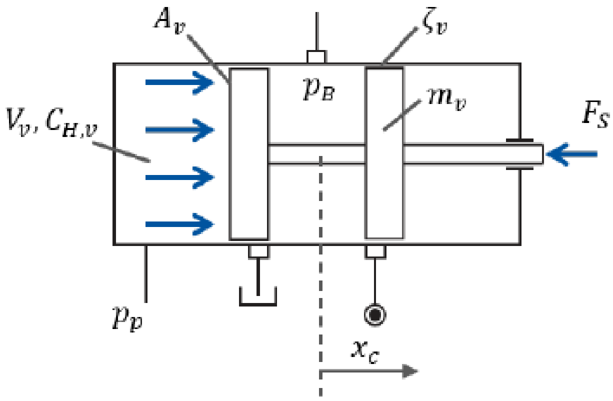


Figure 7: Dynamic model of the active control valve

from the pilot pressure, the force of the valve's spring, friction forces and a force resulting from the flow through the valve F_f are acting on the spool.

$$\ddot{x}_v \cdot m_v = p_p \cdot A_v - k_V \cdot x_V - F_f - K_v \cdot x_v - \zeta_v \cdot \dot{x}_v \qquad (8)$$

Since the supporting force Fs and the force from the pressure $F_{p,V}$ are very high, while the required flow rates are quite small, it can be assumed that the forces from the flow F_f are negligible. The flow Q_B can be described as a function of the spool displacement x_v and the pressure difference across the valve *Dp*.

$$Q_B = c_V \cdot x_v \cdot \sqrt{\Delta p} \qquad (9)$$

In order to apply linear control theory, equation (9) has to be linearized. From a Taylor series, so called valve sensitivity coefficients can be derived for this purpose. The valve coefficients are dependent on the operating point. For system stability, the most critical point of operation in hydraulic systems is the load-free state. Therefore it is used as operation point for the linearization. [4]

$$\Delta Q = \left.\frac{\partial Q}{\partial x_v}\right|_{p0} \Delta x_v + \left.\frac{\partial Q}{\partial p}\right|_{p0} \Delta p + \ldots \approx K_{Qx} \cdot \Delta x_v - K_{Qp} \cdot \Delta p \tag{10}$$

3.3 Complete model of the control system

The subsystems are coupled by the supporting force Fs (equation (7)) and the flow Q_B from valve to brake (eq. (10)). The complete system can be represented as a linear statespace model.

$$\dot{\mathbf{x}} = \mathbf{A} \cdot \mathbf{x} + \mathbf{B} \cdot \mathbf{u} \tag{11}$$

$$y = \mathbf{C} \cdot \mathbf{x} + \mathbf{D} \cdot \mathbf{u} \tag{12}$$

with the definition of the state vector **x** (eq. 13) and the input vector **u** (eq. 14), the system matrix **A** (eq. 15), the input matrix **B** (eq. 16), the output matrix **C** (eq. 17) and the feed-through matrix **D** (eq. 18) can be determined.

$$\mathbf{x} = \begin{pmatrix} x_p & \dot{x}_p & x_c & \dot{x}_c & p_B & x_v & \dot{x}_v \end{pmatrix}^T \tag{13}$$

$$\mathbf{u} = (p_p) \tag{14}$$

$$\mathbf{A} = \begin{pmatrix} 0 & 1 & 0 & 0 & 0 & 0 & 0 \\ -\frac{k_{pad}}{m_p} & -\frac{\zeta_{pad}+\zeta_p}{m_p} & 0 & -\frac{\zeta_p}{m_p} & -\frac{A_p}{m_p} & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & -\frac{\zeta_p}{m_c} & -\frac{\zeta_{pad}}{m_c} & -\frac{\zeta_{pad}+\zeta_c+\zeta_p}{m_c} & \frac{A_p}{m_c} & 0 & 0 \\ 0 & \frac{A_p}{C_H} & 0 & -\frac{A_p}{C_H} & -\frac{K_{Qp}}{C_H} & \frac{K_{Qx}}{C_H} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 \\ \frac{K_s k_{pad}}{m_v} & -\frac{\zeta_{pad}}{m_v} & -\frac{K_s k_{pad}}{m_v} & \frac{\zeta_{pad}}{m_v} & 0 & -\frac{K_v}{m_v} & -\frac{\zeta_v}{m_v} \end{pmatrix} \tag{15}$$

$$\mathbf{B} = \begin{pmatrix} 0 & 0 & 0 & 0 & 0 & 0 & \frac{A_v}{m_v} \end{pmatrix}^T \tag{16}$$

$$\mathbf{C} = \begin{pmatrix} -K_s k_{pad} & -K_s \zeta_{pad} & K_s k_{pad} & K_s \zeta_{pad} & 0 & 0 & 0 \end{pmatrix} \tag{17}$$

$$\mathbf{D} = \begin{pmatrix} 0 \end{pmatrix} \tag{18}$$

To describe the disturbance reaction of the system, the equations 3 to 10 need to be altered. In case of a disturbance reaction, the system is exited by a perturbing effect and not the pilot pressure. For simplification, a generalized perturbing force on the supporting force is introduced to represent different kinds of disturbances. Since the disturbance is assumed to act on the supporting force, the disturbance is directly fed through to the system output. In this case, the input and feedthrough matrices given in equations (19) and (20) can be utilized.

$$\mathbf{B_{dis}} = \begin{pmatrix} 0 & 0 & 0 & 0 & 0 & 0 & -\frac{1}{m_v} \end{pmatrix}^T \tag{19}$$

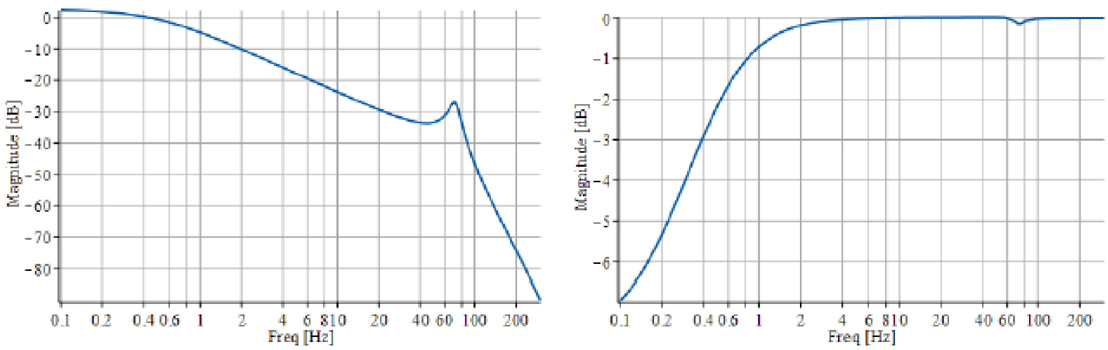
$$\mathbf{D_{dis}} = (1) \tag{20}$$

3.4 Parameter identification

To analyze the system behavior, the introduced model needs to be parametrized. Most of the parameters, such as masses, dimensions and stiffnesses are well known. Nevertheless, the friction coefficients, the material parameters of the brake pads and the valve sensitivity coefficients K_{Qp} and K_{Qx} are more difficult to discern. Especially K_{Qx} is dependent on the operating point and can be strongly influenced by the geometry of the control valve. Values for the valve sensitivity coefficients can be found for given valve geometries and operating points from the flow through the valve. Since the actual prototypes have not been produced yet, the actual friction forces can not be measured. Therefore, the expected friction needs to be estimated from the specifications of the sealings and from similar systems. Finally, the material properties of the brake pads are not well known and utilization of the Voigt model is a significant simplification of the actual material behavior. The calculated dynamics can therefore be expected to deviate from the actual material behavior. The pole/zero configuration of the linearized system reveals that the poles stemming from the stiffness of the brake pads occur at frequencies that are several orders higher than the frequencies of the control system's poles. The poles of the control system are therefore dominant and the dynamics of the brake pads will not effect the dynamics of the control system in any significant way. Therefore, uncertainties in the description of the dynamic material behavior of the brake pads can be tolerated.

4 Dynamic system behavior

From the model described in chapter 3, the dynamic system behavior can be calculated. The results for the frequency response (fig. 8(a)) as well as the disturbance response (fig. 8(b)) are given in figure 8. For the most parts, the frequency response is similar



(a) Frequency response (b) Disturbance response

Figure 8: Calculated frequency and disturbance responses for the controlled system.

to that of an over-damped second-order system. First, there is a linear part in the transfer function with no reduction of the system response for lower frequencies, then the system response is increasingly diminished. Around 70 Hz, the eigenfrequency of the valve can be seen. Here, the system behavior differs from that of a second-order system. Afterwards, the damping is further increased and the system behavior is again similar to the mentioned second-order system. Looking at the disturbance response, a corresponding behavior can be observed. Disturbances are strongly damped for small frequencies. For rising frequencies, the damping reduces until the disturbance is completely uncompensated. As in figure 8(a), the eigenfrequencies of the control valve can be seen around 70 Hz. From both figures, it can be easily seen that the regarded control system is relatively slow. For example, the calculated -3 dB frequency lies slightly below 1 Hz. As mentioned in section 1, requirements for the control unit are to compensate slow disturbances and an easy integration in existing systems. Therefore, the control unit does not need to be fast. Furthermore, in order to utilize the original hydraulic supply systems, the volume flow demand of the control unit may not exceed certain limits. Since fast hydraulic control systems do have a high volume flow consumption, a fast control unit would actually undesirable in this application.

5 Conclusion and outlook

In this paper, a novel brake system with hydro-mechanical control of the braking torque has been described. The system is designed for robustness and easy integration into existing brake systems. The dynamics of the proposed control system have been closely analyzed. While the control unit cannot be designed for high dynamics, satisfying results for the intended working conditions can be achieved. Further works will include the manufacturing of the prototypes and measurements on a specially designed test bench. Finally, deployment of the prototypes in a reference vehicle is planed to examine the effects on the vehicle dynamics.

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