

Evaluation of a State of the Art Hydraulic Brake System with Regard to Future Requirements

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ABSTRACT

New technologies like alternative power trains and driver assistance systems have a big impact on brake system development. Most of the development work aims at the improvement of the actuation and modulation components of the brake system. The basic hydraulic network remained nearly the same over decades and still has to meet these new requirements.

Previous papers have focused mainly on studying the behavior of single components, like for example the brake hose fluid consumption in detail. Other papers studied the complete system but simplified it extremely, so that some relevant effects are neglected. In this work, one focus is to study the influence of single relevant components, like the hydraulic unit and the hoses on the overall system performance. For this measurements with a complete hydraulic brake system, including a state of the art electromechanical brake booster and single component measurements for identification, are conducted. A variety of ambient temperatures, from low to high, is included in this study by using a temperature chamber for measurements. The dynamic pressure propagation through the system is studied, as it is crucial for future high dynamic brake systems. Besides that, key figures which not only focus on the dynamics but also on the efficiency of the system are introduced. As braking maneuvers are often short term and extremely dynamic processes, measurement equipment, like flow meters, can hardly be integrated without affecting the system behavior significantly. Therefore simulation models with different degrees of component detail are generated from an ideal system to the complete detailed system to get a better understanding of the relevant components and their effects on the before introduced key figures.

This work could support developers to get a better understanding of the effects of certain components on the system overall performance and how to improve them to meet future requirements.

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INTRODUCTION

The hydraulic brake system with its centralized architecture has been established and further improved over the last decades. The boosting device to amplify the driver's foot force might be different from system to system, for example pneumatically, electromechanically or even hydraulically. What stays the same for all of them is that the hydraulic pressure due to this amplified driver foot force is generated at a centralized position in the engine compartment, and from there transferred over the pipes and hoses to the wheel brakes.

The hydraulic brake line system itself has not been in the focus of researchers in the last years anymore. Therefore only few studies can be found addressing this part of the brake system. Even though not all the effects and their relevance to the overall system performance are understood yet, researchers and developers focus mainly on the active part of the brake system which is relevant for the pressure build up

itself, like the actuation (booster) or modulation (hydraulic control unit) devices. With permanently increasing requirements for reaching high ratings in safety tests, the influence of the pipes, hoses and valves on the overall brake system performance could offer insights on how to further decrease pressure build up times. Furthermore efficiency is getting more and more into the focus for all auxiliary systems with regard to electric vehicles. The electrification of the powertrain and braking through recuperation leads to a shift of the overall load collective towards lower pressures.

Pang and Agnew [1] studied the flow resistances in the brake system. They reported that the main affecting factors of fluid flow rate are the brake tube size, hydraulic control unit (HCU) valve size, caliper size and temperature. In their studies they first introduce a simplified model of the brake system to predict the pressure difference through the different components of the brake system. To simulate the

pressure drop from the master cylinder, to the spot where brake fluid is leaving the hydraulic control unit (HCU), they used the nozzle equation and adapted it by a factor, which is fitted by using measurement data. For modeling the part from the HCU-output to the wheel brakes, they simplified this part by using the equation for flow inside a tube. To take flow resistances into account, the equivalent length of the resistances the brake fluid has to pass, are added on top of the regular length of the pipe and brake hose. This model is applied to a test vehicle, which is accordingly equipped with pressure sensors and compared to measurement results. Therefore one of the main simplifications here is that the brake hoses are modeled as pipes with rigid walls, hose expansion is neglected.

Due to the disadvantages of a flow meter, the assumption is used, that the flow rate is linearly correlated to the master cylinders piston area and the speed of travel. The studied braking maneuver has a maximum input rod velocity of ~ 170 mm/s (assuming a pedal ratio of 3.5). Their simulation model correlates well with the recorded data in power failure mode at room temperature. They further report that the main resistance at room temperature is the HCU.

In their work they studied the behavior of the braking system for only one single braking maneuver. The presented modeling approach correlates well only for room temperature with the measurements. Antanaitis, Riefe and Sanford [2] focused in their work on the brake hose fluid consumption. They stated that the brake hoses have a significant effect on corner- and system-level on the fluid consumption and the brake system response time. They quantified the fluid consumption of brake hoses for a vast number of brake hoses. In total they measured 33 rubber hoses from eight different manufacturers. The section lengths ranged from 102 mm up to 670 mm. Their research included ambient temperature fluid consumption testing, but also various temperatures have been studied.

The resolution of the used fluid displacement transducer on the dynamometer was 0.01 cm^3 . The studied temperature varied between room temperature and 125°C . No temperatures below room temperature were studied. The rubber hose fluid consumption was more than doubled between room temperature and 50°C . They noticed that most of the temperature related changes in mechanical properties of the hoses are activated between 50 – 75°C .

The two introduced studies on hydraulic brake systems focus on two different aspects of the brake system itself. Pang and Agnew [1] focused purely on the resistance in the brake system with regard to pressure build-up. They ignored the behavior of the brake hoses and assumed all pipes and hoses to be rigid. Antanaitis, Riefe and Sanford [2] looked in their study mainly on the behavior of the rubber brake hose itself. Both effects, the resistances and the volume losses, seem to play an important role for brake system performance. It is not yet understood how important which effect is for different kind of braking maneuvers or driving cycles.

In the following section the utilized brake test bench is introduced. Key figures for the further analysis and rating of the systems overall performance are introduced. Measurement data for a hydraulic brake system of a small electric passenger car for various temperatures and input velocities is presented and analyzed. Followed by single

component identification measurements and modelling. These single component models are then implemented in the overall brake system model. The effects of resistances and brake hose behavior on the brake system performance are studied in simulation and compared with measurements. The relevance of these effects for different kind of braking maneuvers and the needed degree of detail for the simulation model are discussed.

In this study the two relevant tools used for investigating the brake system characteristics are the brake test bench and the simulation. The test bench is used to generate the measurement data for component identification, as well as for measurement of the complete brake system with and without brake booster.

BRAKE TEST BENCH SETUP

The studied brake system, which is schematically shown in [figure 1](#), is taken from a small sized electric passenger car. This car is equipped with disc brakes at the front axle and drum brakes at the rear axle. The piping and hoses are taken of the original setup. The brake booster, with the master cylinder following, used in this system is an electromechanical brake booster, with an electric motor to amplify the driver foot force. The basic design inside the booster is comparable with that of a vacuum booster, except for the replacement of the valves and a boost body driven by the electric motor surrounding the valve body of the booster. The hydraulic unit in this system is a third party product and can only be used in passive mode. Meaning the fluid is just streaming through the normally open valves of the component without undergoing any active changes through switching valves or boosting by the hydraulic-units pump. Therefore also no electronic brake distribution (EBD) is taking place, which would for example limit the pressure at the rear brakes to avoid wheel locking.

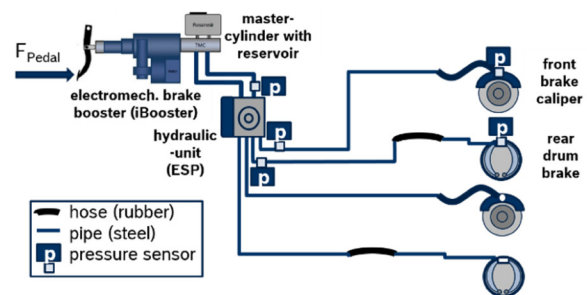


Figure 1. Brake system layout with electromechanical brake booster

The brake test bench, pictured in [figure 2](#), is a system test bench, on which the complete brake system is mounted and tested under varying conditions. Unlike an inertia dynamometer this type of test bench does not have a rotating mass or brake disc. It is mainly used to study the overall system performance under varying ambient temperature conditions, including the vehicle systems relevant operating range from -40°C to 120°C . Therefore the brake system, including the booster and all hydraulic parts, is located inside a temperature chamber, which allows to investigate these operating points in a reproducible manner.

For the actuation of the brake system a brake robot consisting of a combination of an AC-Servomotor with a linear gearbox is used. The system pushes directly inline on the input rod of the booster or in case of a setup without booster on the plunger of the master-cylinder. Omitting the brake pedal, is beneficial for using an inline setup for the actuation of the system. Therefore the lever of the brake pedal is taken into account, when converting the pedal travel and force to the corresponding input rod travel and force. The high force potential of the servomotor makes it possible to cover these high forces easily. The maximum velocity the pushrod can be driven with by the brake robot is 250 mm/s, which covers almost all realistic actuation cases. The needed travel range is small, and has a maximum for the here used master cylinder of around 40 mm. Special adapters are mounted with the system, equipped with a force transducer located before the input rod and at the booster pushrod. By that, the input force and the amplified force by the booster can be measured.

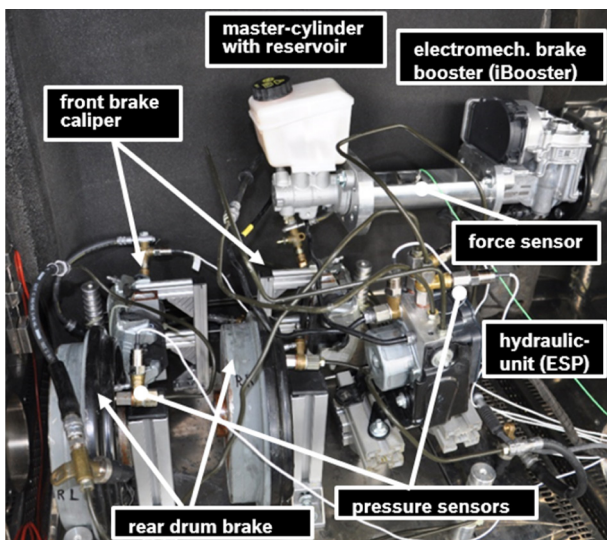


Figure 2. Brake test bench setup inside the temperature chamber

For the acquisition of the measurement data, there are two relevant sources. One is the data received from the CAN-bus of the servomotor, this includes the travel and speed data. The travel signal of the servomotor is generated by using the integrated encoder of the motor. Therefore it has a high accuracy and a resolution of 0.01 mm. Further the system is equipped with two aforementioned force transducers and five pressure sensors. This data is recorded by a National Instruments CompactRIO. The analog signals, like force and pressure, are recorded using the FPGA of the CompactRIO. The CAN-Data is recorded by a CAN-Card with the Real Time System. It should be mentioned that the pressure sensors are read in with 1 kHz. Even more important is that the analog digital converter used for transforming the analog signal only needs 8 micro seconds for one channel. This and the fact that they are connected without any other signals in between makes sure that the maximum delay between two pressure signals is < 0.4 ms. Due to this setup it is possible to use the pressure signals also to study changes in pressure dynamics for example during pressure build-up.

KEY FIGURES

The totality of requirements for brake systems are very large. Many of them are hard to quantify. To help developers in evaluating brake system components and architectures, here two types of key figures are studied in detail. By analyzing the load collective for brake systems described in AK-Bremse [3], which is used for developing new brake systems, one can conclude that the brake system has to cover very few high dynamic high pressure cases as well as a vast amount of low pressure low dynamic events. In the following the already in use key figure to evaluate the dynamics of a system and several here newly introduced key figures to rate latency and efficiency of a brake system are described in detail.

Dynamics

For the evaluation of brake system dynamics the time to reach the pressure for locking the wheel is used. This figure is also known as time to lock (TTL). The TTL is defined as the time which is needed from braking initiation until locking pressure is reached. The here used definition for the t_{TTL} is the difference of the point in time when the input rod of the brake booster device begins to move $t_{inputrod,Booster}$ and the time needed to reach locking pressure at the wheel $t_{lock,Wheel}$ therefore

$$t_{TTL} = t_{lock,Wheel} - t_{inputrod,Booster} \quad (1)$$

The TTL is typically a value between 150-500 ms for state of the art brake systems. A certain TTL has to be achieved with the brake system to reach the existing NCAP regulations. These regulations demand for increasing system dynamics. With regard to the centralized hydraulic architecture the TTL basically rates the performance of the brake system booster, the actuation device. It should also be mentioned that for a good high dynamic braking not only the actuation dynamics is important, but also the modulation of the wheel pressures to maximize the deceleration. Here also due to limitations of the test bench this property is not studied. In this study it is further interesting how much additional time is needed for this pressure generated at the master-cylinder to reach the wheel brakes, therefore a new key figure delta TTL is introduced and defined as the difference in time between reaching locking pressure at the studied wheel and master-cylinder

$$\Delta t_{TTL} = t_{lock,Wheel} - t_{lock,MC} \quad (2)$$

Efficiency

As high critical high dynamic emergency braking events make up only a very small part of the overall load collective in a brake systems life an additional key figure is needed. Especially with regard to the shift in the brake load collective due to the electrification of the powertrain and the possibility of recuperative braking. A new key figure which makes it possible to evaluate the efficiency of the brake system during pressure build up is introduced. The new proposal in this study is to abstract the brake system to a hydraulic pump, and study how efficient this pump is in building up pressure. To study the influence of single components and parameters on the overall

systems efficiency can be studied with this new key figure. This can help developers in choosing, designing and optimizing brake systems for future vehicle concepts.

The efficiency is defined as the relationship of the effective work, which is for this case defined as the hydraulic power over time, and the input power to the system. The conventional understanding of efficiency is that, for example for an electric motor, it directly describes the relationship between input and output work. By using the mean efficiency to describe a whole pressure build up process the dependency on response speed of the system is becoming relevant. The effective work is the sum of the hydraulic power, which is defined as the product of wheel pressure $p_{n,i}$ and the flow rate $Q_{n,i}$ for all four wheel brakes ($n=4$). The current flow rate is calculated by using the wheel brakes pressure over volume characteristic and the available pressure signal. With the measured pressure values for two time steps the corresponding volume is known and can be used to calculate the mean volume flow rate over each time interval Δt_n . As this derivative signal is very noisy for the measurements a model based filter like a Kalman filter is used. In simulations though the volume flow rate is directly known. The input power for a servo brake system is the sum of the driver foot force on the pedal F_{pedal} and the absorbed power by the booster $P_{booster}$ over time, therefore for the efficiency for pressure build-up follows

$$\eta_{tot} = \frac{W_{OUT}}{W_{IN}} = \frac{\sum_{i=1}^N \sum_{n=1}^4 p_{n,i} Q_{n,i} \Delta t_n}{\sum_{n=1}^N F_{Ped,n} i_{Ped} \Delta s_n + \sum_{n=1}^N P_{Booster,n} \Delta t_n} \quad (3)$$

The energy is numerically integrated, with the time step Δt_n and the corresponding travel of the pedal, with the pedal ratio i_{ped} the input rod Δs_n , for the time interval until the target pressure is reached N . This allows to study the efficiency during pressure build-up as well as determining a mean efficiency value for the maneuver. With the wheel pressure $p_{n,i}$ in the numerator, one can already conclude that the efficiency is dependent on the response speed of the system in building up pressure. The overall efficiency η_{tot} can further be split into a mechanical part η_{mech} and a hydraulic part η_{hydr} . The hydraulic part describes the efficiency of the hydraulic network η_{hydr} in transmitting energy as pressure and volume to the wheel brakes. For this analysis only the input to the hydraulic network is taken into account

$$\eta_{hydr} = \frac{\sum_{i=1}^N \sum_{n=1}^4 p_{n,i} Q_{n,i} \Delta t_n}{\sum_{n=1}^N A_{MC} v_{MC} P_{MC} \Delta t_n} \quad (4)$$

which is the hydraulic power generated by the master-cylinder (MC).

MEASUREMENT RESULTS FOR THE HYDRAULIC BRAKE SYSTEM WITHOUT BRAKE BOOSTER

Due to the high force potential of the combination of the servomotor with a linear actuator it is also possible to push directly onto the master-cylinder without a brake booster mounted in between, as can be seen in [figure 3](#). This allows the usage of certain excitation signals, like for example constant speed profiles, which would be deflected by the brake boosters amplifying characteristic if present. With these profiles defined operation points of the brake system can be studied and the behavior at that point can be classified. Starting with a quasi-static measurement of the pressure over volume consumption of the system with minimal speed 0.1 mm/s. Measurement rows in a range from 0.1-200 mm/s are taken. As the brake fluid properties are very sensitive to temperature measurement results for -20 °C, 23 °C (room temperature) and 80 °C are included in this study.

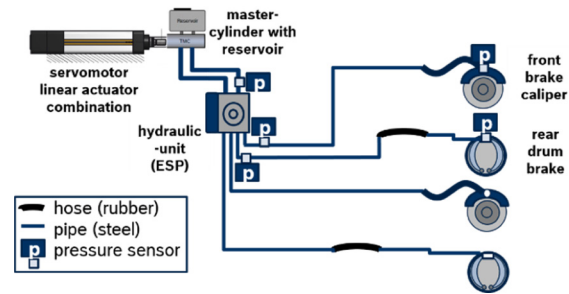


Figure 3. Direct actuation through servo linear actuator combination without brake booster

It should be mentioned here that even though the input speed at the master-cylinder is $v_{inputspeed}$, the effectively produced volume flow of each of the master-cylinders chambers equals to

$$Q_{Circuit} = \frac{v_{inputspeed}}{2} A_{MC} \quad (5)$$

This is due to the fact that the first chambers piston is moved with $v_{inputspeed}$ but at the same time the second piston is enlarging the first chambers volume by half of the input speed. Therefore the velocity volume is pushed into the circuits with is $v_{inputspeed}/2$.

In [figure 4](#), the pressure over time profiles for four different input velocities are plotted. Measurements are taken up to the typical locking pressure of 100 bar. The quasi-static case is shown in [figure 4a](#)), in this case the system has enough time to reach state of equilibrium during pressure build-up. The pressure difference dp is defined as

$$dp = p_0 - p_1 \quad (6)$$

which reflects the difference in pressure over the studied brake systems hydraulic network, between the generation spot master-cylinder 0 and the target spot wheel 1. A first noticeable pressure difference can be seen starting at input velocities 0.1 mm/s (depicted in [figure 4b](#)). With increasing input velocity the pressure