Non-linear Control and Observer Design for the Self-energizing Electro-Hydraulic Brake

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ABSTRACT

The paper presents the Self-energizing Electro-Hydraulic Brake (SEHB) with a non-linear controller, with focus on the design of the observer. The objective for designing a non-linear control algorithm for the innovative Self-energizing Electro-Hydraulic Brake (SEHB) is to obtain a controller that is insensitive concerning possible parameter changes of the control path and especially the friction coefficient. The designed autarkic brake control unit on which the non-linear closed-loop control will be implemented, is located close-by the brake actuator and is to be operated by simple open loop commands from a superior control unit. The paper shortly explains the brake principle of the SEHB and a simplified model of the brake system. The design of the linearizing feedback and the observer is presented. The controller shows good disturbance rejection in simulative studies using a sophisticated, previously verified simulation model of the brake system.

INTRODUCTION

Hydraulic servo systems generally have high demands on the control algorithm. Linear control methods are conventionally valid for a limited operating range. In order to facilitate a simple control strategy and to fulfill the requirements of servo systems, non-linear controllers may be utilized. Many non-linear control algorithms are based on the state space representation.

For the case of the SEHB, the input-output linearization method has been used to linearize the non-linear control path. With this approach, a simple full state feedback control can be used. In a previous publication, a state feedback controller was presented for the SEHB.

A disadvantage of full state feedback control in practice is that all state variables have to be measured. A cost effective solution to reduce instrumentation is the use of an observer, which estimates the states. Sometimes this is the only viable approach as many dynamic variables are hard to measure.

For the SEHB a full state feedback is used with an observer. First of all, a functional overview of the SEHB is given in the next chapter.

PRINCIPLE OF THE SEHB

There are numerous ways to realize a hydraulic self-energizing brake. The brake system layout of the system under study is shown in Fig. 1. The brake system is dedicated for railway applications and mainly consists of the brake pads, a brake caliper, and a brake actuator. The brake actuator clamps the brake hydraulically. In contrast to conventional brakes, the SEHB features an additional supporting cylinder. The functionality shall be described at the example of a regular braking process:

To start the braking process, the left control valve is opened from neutral position positively. Pressurized oil flows towards the piston face side chamber of the brake actuator. The stroke of the brake actuator causes the brake pads to be pressed onto the brake disc. Thereby, a normal force is applied and a friction force develops. The brake caliper is suspended movable and due to the friction force it advances in tangential direction. By the movement the brake caliper exerts a force on the supporting cylinder. In doing so, pressure builds up in the chamber of the supporting cylinder. As the control valve is still open in positive direction, the pressurized oil is continued to be fed to the piston face side chamber of the brake actuator, increasing the clamping force. In this way the loop of self-energization is closed.
As a result, the brake draws its actuation energy out of the kinetic energy of the vehicle and only necessitates low electric power for the control valve. By measuring the pressure in the supporting cylinder, the brake torque is controlled. This is a major benefit of the SEHB when compared to conventional brakes.

In order to design a model based controller for the SEHB, the next chapter presents the system model. For the sake of simplicity the brake force build-up is described only.

MATHEMATICAL SYSTEM DESCRIPTION

MATHEMATICAL MODEL – During the brake force build-up the ring side chamber of the brake actuator is connected to the low pressure line by the fully opened control valve on the right hand side in Fig. 1. As only one chamber is pressurized the brake actuator shall be modeled as a plunger cylinder. The supporting cylinder is also a plunger cylinder. For the control of the piston side chamber of the brake actuator only one control edge is necessary. Therefore, a 2/2-way proportional control valve is used for the mathematical model. Furthermore, if the valve is opened, as is the case for brake force build-up, there is a single hydraulic capacity in the system. These simplifications lead to the system illustrated in Fig. 2.

This simplified model consists of a plunger brake actuator, a plunger supporting cylinder, a mechanical connection between the cylinders, and a 2/2-way proportional valve to control the oil flow. The hydraulic capacity is concentrated in the chamber of the brake actuator.

The pressure build-up equation of the brake actuator can be formulated by

$$\dot{p}_{BA} = \frac{1}{C_H} \left( B \sqrt{p_{SC} - p_{BA}} \cdot x_V - A_{BA} \cdot \dot{x}_{BA} \right) \quad (1)$$

If friction, damping, and spring forces are neglected, \( \dot{x}_{BA} \) can be expressed by

$$\dot{x}_{BA} = \frac{A_{BA}}{c_{cal}} \cdot \dot{p}_{BA} \quad (2)$$

By taking the mechanical connection of brake actuator and supporting cylinder into account, the pressure of the supporting cylinder can be expressed depending on the brake actuator pressure (3).

$$p_{SC} = \frac{2 \cdot \mu \cdot i \cdot p_{BA} \cdot A_{BA}}{A_{SC}} \quad (3)$$

Inserting (2), (3) into (1) and solving for \( \dot{p}_{BA} \) yields (4).

$$\dot{p}_{BA} = \frac{B \cdot \left( \frac{2 \cdot \mu \cdot i \cdot A_{BA}}{A_{SC}} - 1 \right) \cdot p_{BA} \cdot x_V}{C_H + \frac{A_{BA}^2}{c_{cal}}} \quad (4)$$

The dynamics of the control servo valve is approximated as a second order lag element\(^6\). Then the acceleration and the velocity of the valve spool are expressed by (5) and (6).

$$\ddot{v}_V = \omega^2_V \cdot K_V \cdot u_V - 2 \cdot D_V \cdot \omega_V \cdot v_V - \omega^2_V \cdot x_V \quad (5)$$

$$\dot{x}_V = v_V \quad (6)$$
The equations (4), (5), and (6) constitute the three-dimensional state space representation of the SEHB, which forms the basis for the non-linear controller synthesis. A fixed friction coefficient is assumed in the mathematical model.

POSSIBLE IMPROVEMENTS TO THE MATHEMATICAL MODEL – In order to model the SEHB system more precisely, the underlying mathematical model can be enhanced. With the more sophisticated modeling of the SEHB described in this chapter the controller is designed.

Stiffness - The greatest inaccuracy of the simplified model described above is the combined stiffness of the brake pads and the brake caliper $c_{cal}$. As is known from measurements, this stiffness is not constant like assumed above. Instead, the stiffness $c_{cal}$ depends on the position of the brake pads with respect to the brake disc. The position of the brake pads is directly related to the position of the brake actuator $x_{BA}$. The relation between the stiffness $c_{cal}$ and the position of the brake actuator has been determined by measurements and is a polynomial equation, in which $x_{BA}$ is unknown. With the balance of forces of the brake actuator $c_{cal}$ is expressed dependent solely on the brake actuator pressure.

Friction and spring forces - Another major refinement can be applied to the mathematical model by considering the inner friction of the two cylinders and the spring force of the brake actuator. Then the balance of forces for the brake actuator is given by (7).

$$F_{BA} = p_{BA} \cdot A_{BA} - F_{Fr,BA} + F_{Sp,BA} = c_{cal} \cdot x_{BA}$$

The balance of forces for the supporting cylinder is then described by (8).

$$F_{SC} = p_{SC} \cdot A_{SC} - F_{Fr,SC}$$

Solving equation (7) for $x_{BA}$ and differentiating it yields (9).

$$\dot{x}_{BA} = \frac{\dot{p}_{BA} \cdot A_{BA} - \dot{F}_{Fr,BA} + \dot{F}_{Sp,BA}}{c_{cal}}$$

Neglecting the derivatives with respect to time of the friction and spring force, and inserting (11) and (10) into the pressure build-up equation of the brake actuator (4), leads to equation (12).

$$\dot{p}_{BA} = \frac{B \cdot x_{v} \cdot \left[ \frac{2 \cdot \mu \cdot i \cdot p_{BA} \cdot A_{BA} - F_{Fr,BA} + F_{Sp,BA}}{A_{SC}} + \frac{F_{Fr,SC}}{C_{ii} + \frac{A_{BA}}{c_{cal}}} - p_{BA} \right]}{C_{ii} + \frac{A_{BA}}{c_{cal}}}$$

Equation (10) is the substitution of equation (4).

Determination of the frictional forces - When examining equation (10), three new parameters are identified. The spring force of the brake actuator is known as a product of the spring constant and the position of the brake actuator. The other two parameters are the inner friction forces of the cylinders. In order to determine them they were both measured. For the brake actuator a dependency of the direction of travel was identified, (11) and (12).

$$F_{Fr,BA} = 1800 \text{ N} + \dot{x}_{BA} \cdot 1 \text{ N/s mm} \cdot x_{BA} \geq 0$$

$$F_{Fr,BA} = 1300 \text{ N} - \dot{x}_{BA} \cdot 1 \text{ N/s mm} \cdot x_{BA} < 0$$

The inner frictional force of the supporting cylinder is lower because of smaller diameters and described by equation (13).

$$F_{Fr,SC} = \text{sign} (\dot{x}_{SC}) \cdot \left( 175 \text{ N} + \dot{x}_{SC} \cdot 20 \frac{\text{N s}}{\text{mm}} \right)$$

Cylinder velocities in the equations (11), (12), and (13) are obtained by equation (9) and the mechanical connection.

The state space description constituted by equations (10), (5), and (6) is the basis for control development in the next chapter.

LINEARIZING CONTROL

To obtain a linear system for the SEHB an input-output linearizing controller is designed. The controller design shall not be discussed further in this paper since it is given in a previous article. Fig. 3 gives an overview of the input-output linearizing controller.

$$F_{Fr,SC} = \text{sign} (\dot{x}_{SC}) \cdot \left( 175 \text{ N} + \dot{x}_{SC} \cdot 20 \frac{\text{N s}}{\text{mm}} \right)$$

The control path of the SEHB is non-linear due to the orifice formula of the control valve. Therefore, the relation of the system’s output, the supporting cylinder pressure $p_{SC}$, and the system’s input, the control valve voltage $u_{v}$, is also non-linear. With the feedback designed by the input-output linearization a linear relation of the new system input $v$ and the output $p_{BA}$ is...
achieved. A coordinate transformation from $x$ to $z$ coordinates is crucial to design the linearizing feedback.

**OBSERVER DESIGN**

STATE SPACE CONTROLLER – The linearized control path with the new input $v$ and the output $p_{BA}$ is illustrated in Fig. 4. To control the system a state space controller is applied. The state space controller creates the output $v$, which is the new input of the control path. Inputs of the state space controller are the reference input $w$ ($p_{BASE}$) and the vector $z$, consisting of the three variables brake actuator pressure $p_{BA}$, control valve spool velocity $v_v$, and control valve spool position $x_v$. The state feedback can then be formulated by

$$v = p_{BASE} \cdot S - k_1 \cdot p_{BA} - k_2 \cdot z_2 - k_3 \cdot z_3$$

(14)

**Fig. 4: State space controller**

![State Space Controller Diagram](image)

For the case of the SEHB the system matrices $B$ and $C$ are column matrices $b$ and $c$ (SISO system). Moreover, the controller has a row matrix $k$. For the input of the reference signal, a transfer factor $S$ is required. The differential equation for the transformed SEHB system in state space representation is given by (15).

$$\dot{z} = A \cdot z + b \cdot v$$

(15)

The system's output $y$ is expressed by (16).

$$y = c^T \cdot z$$

(16)

The state space controller is designed with the pole assignment method. The poles of the open loop SEHB control path are established by the matrix $A^T$. They are determined by solving the characteristic equation (17).

$$\det(s \cdot I - A) = \det \left[ \begin{array}{ccc} s & 0 & 0 \\ 0 & s & 0 \\ 0 & 0 & s \end{array} \right] - \left[ \begin{array}{ccc} 0 & 0 & 1 \\ 0 & 0 & 1 \\ 0 & 0 & 0 \end{array} \right] = s^3 = 0$$

(17)

It can be clearly seen that all three poles of the control path, $s_1$, $s_2$, and $s_3$, are zero. Aim of the pole assignment is to place all poles of the closed loop left of those of the control path$^{1)}$. With the knowledge of the control path poles it is possible to allocate the poles of the closed loop system. These poles are designated by the matrix $A_k$ (18).

$$A_k = A - b \cdot k^T = \begin{pmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & 0 & 0 \end{pmatrix} - \begin{pmatrix} k_1 & k_2 & k_3 \end{pmatrix}$$

(18)

By setting up the characteristic polynomial of the closed loop in the factored form with $s_1$ roots of the polynomial

$$(s - s_{c1}) \cdot (s - s_{c2}) \cdot (s - s_{c3}) = 0$$

(19)

and comparing its coefficients in the expanded form with $\det(s \cdot I - A_k) = 0$, the equations for the elements of the controller vector $k$ are expressed (20).

$$k = \begin{pmatrix} k_1 \\ k_2 \\ k_3 \end{pmatrix} = \begin{pmatrix} -s_{c1} \cdot s_{c2} \cdot s_{c3} \\ s_{c1} \cdot s_{c2} + s_{c2} \cdot s_{c3} + s_{c1} \cdot s_{c3} \\ -s_{c1} \cdot s_{c2} - s_{c3} \end{pmatrix}$$

(20)

Equation (20), also referred to as Ackermann Formula$^3$), allows the calculation of the controller coefficients depending on the poles of the closed loop SEHB system which can be chosen by the designer. All three real values of the controller poles are set at -20 1/s, so that the controller vector $k$ calculates to

$$k = \begin{pmatrix} k_1 \\ k_2 \\ k_3 \end{pmatrix} = \begin{pmatrix} 8000 \frac{1}{s^3} \\ 1200 \frac{1}{s^2} \\ 60 \frac{1}{s} \end{pmatrix}$$

The transfer factor $S$ is calculated so that the system's output equals the reference input $w$ under steady state conditions$^3$, for which (21) holds.

$$\dot{z} = A_k \cdot z + b \cdot S \cdot w = 0$$

(21)

This leads to equation (22) for the determination of $S^\top$.

$$S = - (c^T \cdot A_k^{-1} \cdot b)^\top$$

(22)

With

$$c^T = \begin{pmatrix} 1 & 0 & 0 \end{pmatrix}$$

(23)
and
\[
A_k^{-1} = \begin{pmatrix}
-\frac{k_2}{k_1} & -\frac{k_3}{k_1} & -\frac{1}{k_1} \\
1 & 0 & 0 \\
0 & 1 & 0
\end{pmatrix},
\tag{24}
\]

Therefore \( S \) is calculated to
\[
S = k_1 = 8000 \frac{1}{s^3}.
\tag{25}
\]

The poles of the state space controller are placed left of the open loop poles\(^1\).

A problem of state space controllers often is that not all system states are accessible. For the railway application of the SEHB it is intended to solitary measure the pressure of the supporting cylinder \( p_{SC} \). By the mechanical connection, see equation (3), the brake actuator pressure \( p_{BA} \) is related to \( p_{SC} \). The state variables of the control valve, \( v_v \) and \( x_v \), are hardly accessible through measurement and are thus unknown. Taking the application of the controller into account, only two out of the three state variables of the state vector \( z \) are known in practice.

LUENBERGER OBSERVER – To overcome this problem, an observer is designed. Instead of measuring the unknown two state variables, they are estimated. The observer uses a model of the control path in parallel and estimates the state vector \( z \), which is then termed \( \hat{z} \), Fig. 5.

Fig. 5: SEHB state space controller with observer

For the control problem of the SEHB, a Luenberger observer approximates the state vector. It uses the input of the control path \( v \) and the available output, which is the brake actuator pressure \( p_{BA} \). In addition to the parallel control path model, the Luenberger observer utilizes a feedback of the difference between the system output \( p_{BA} \) and the estimated observer output \( \hat{p}_{BA} \). In doing so, the observational error of the SEHB system is minimized.

The observational error is defined as (26).
\[
e = z - \hat{z}
\tag{26}
\]

Its differential equation is given by (27).
\[
\dot{e} = (A - l \cdot c) \cdot (z - \hat{z}) = (A - l \cdot c) \cdot e
\tag{27}
\]

According to Luenberger\(^5\), it is necessary that the eigenvalues of the observer are negative, which causes the state of the observer to converge to the state of the original system. If all eigenvalues of the matrix \( (A - l \cdot c) \) have negative real parts, equation (28) holds for the observational error\(^1\).
\[
\lim_{t \to \infty} \|e(t)\| = 0
\tag{28}
\]

Lunze\(^1\) proposes to set the eigenvalues of the observer significantly left of the eigenvalues of the control path respectively the closed loop (Fig. 6).

Fig. 6: Observer poles are left of closed loop poles

He furthermore recommends that the absolute value of the real parts should be two to six times as large as the absolute values of the dominating eigenvalues.

The eigenvalues of the observer are determined by characteristic equation (29).
\[
\det(A - l \cdot c) = 0
\tag{29}
\]
\[
\det \left( s \cdot I - A + l \cdot c^T \right)
\Leftrightarrow \det \left( \begin{pmatrix}
s & 0 & 0 \\
0 & s & 0 \\
0 & 0 & s
\end{pmatrix} - \begin{pmatrix}
0 & 1 & 0 \\
0 & 0 & 1 \\
0 & 0 & 0
\end{pmatrix} + \begin{pmatrix}
l_1 \\
l_2 \\
l_3
\end{pmatrix} \cdot (1 & 0 & 0) \right)
\tag{30}
\]

If the calculations are carried out, (30) leads to (31).
\[
\det \left( s \cdot I - A + l \cdot c^T \right) = s^3 + s^2 \cdot l_1 + s \cdot l_2 + l_3 = 0
\tag{31}
\]
By pole assignment of the observer eigenvalues $s_{o1}$, $s_{o2}$, and $s_{o3}$ and a comparison of coefficients with equation (31) follows the Ackermann Formula (32).

$$l_1 = -s_{o1} - s_{o2} - s_{o3}$$
$$l_2 = s_{o1} \cdot s_{o2} + s_{o2} \cdot s_{o3} + s_{o1} \cdot s_{o3}$$
$$l_3 = -s_{o1} \cdot s_{o2} \cdot s_{o3}$$

(32)

If the poles of the control path of the SEHB are known, the coefficients of the Luenberger matrix are determined by pole placement of the observer eigenvalues. They are set to -60 1/s, so that the Luenberger matrix calculates to

$$l_1 = 180 \frac{1}{s}$$
$$l_2 = 10800 \frac{1}{s^2}$$
$$l_3 = 216000 \frac{1}{s^3}$$

The differential equation of the observer yields (33).

$$\dot{z} = A \cdot \dot{z} + b \cdot v + l \cdot (y - \hat{y})$$
$$\dot{z} = \begin{pmatrix} \dot{z}_2 \\ \dot{z}_3 \\ 0 \end{pmatrix} + \begin{pmatrix} 0 \\ 0 \\ v \end{pmatrix} \cdot \begin{pmatrix} l_1 \cdot (z_1 - \dot{z}_1) \\ l_2 \cdot (z_1 - \dot{z}_1) \\ l_3 \cdot (z_1 - \dot{z}_1) \end{pmatrix}$$

(33)

With the given coefficients it is

$$\dot{z} = \begin{pmatrix} \dot{z}_2 \\ \dot{z}_3 \\ 0 \end{pmatrix} + \begin{pmatrix} 180 \frac{1}{s} \cdot (z_1 - \dot{z}_1) \\ 10800 \frac{1}{s^2} \cdot (z_1 - \dot{z}_1) \\ 216000 \frac{1}{s^3} \cdot (z_1 - \dot{z}_1) \end{pmatrix}$$

It can be seen that the estimated state vector uses only one measurable input $z_1$, which is the set point of the brake actuator pressure. The absolute value of the real parts of the observer poles are placed three times left of the closed loop poles.

### SIMULATION

**CONTROLLER PERFORMANCE** – The controller with the observer is implemented in a simulation environment together with a model of the whole SEHB system. The simulation model is built using the system simulation tool DSHplus. The parameters for the simulation designated “Reference” are shown in Table 1.

<table>
<thead>
<tr>
<th>Simulation Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Valve Eigenfrequency $f$</td>
<td>55 Hz</td>
</tr>
<tr>
<td>Friction Coefficient $\mu$</td>
<td>0.35</td>
</tr>
<tr>
<td>Nominal Valve Flow @ 35 bar $Q_N$</td>
<td>20 l/min</td>
</tr>
<tr>
<td>Valve Damping Coefficient $D$</td>
<td>0.7</td>
</tr>
<tr>
<td>Real Values of Controller Poles $s_{ci}$</td>
<td>-20 1/s</td>
</tr>
<tr>
<td>Real Values of Observer Poles $s_{oi}$</td>
<td>-60 1/s</td>
</tr>
</tbody>
</table>

All variations are completed by changing one parameter at a time, whilst the others are kept at the values listed in Table 1. Most importantly, the controller and observer poles, respectively the assigned poles, are not changed. The brake force decrease is also controlled by the non-linear controller.

The simulation model of the brake was validated by test rig measurement data of the SEHB so that simulation and prototype behavior fit very well (Fig. 7). The results were obtained with a switching controller. With the verified simulation, the non-linear control is investigated by simulation.

**Fig. 7: Measurement and simulation of SEHB system**

**Fig. 8** shows the step response of the simulation model controlled by the described state space controller in combination with the Luenberger observer. The figure illustrates the set point of the brake force and the simulated brake force with the non-linear controller. Besides the set point for the brake force, the solitary input of the controller is the pressure of the supporting cylinder $p_{SC}$, from which the brake actuator pressure necessary for the controller feedback calculations is obtained by solving equation (3) for $p_{BA}$.
Fig. 8: Simulation result of controller with observer

Fig. 8 also shows the step response of a well tuned P-controller for steps of the set point signal from 10000 N to 20000 N and 20000 N to 30000 N. The P-controller is tuned for the first step response. At the first step response, the P-controller reaches 90% of the brake force after 431 ms and the non-linear controller after 479 ms. At the second step response the time to reach 90% of the brake force is 210 ms for the P-controller, whereas the non-linear controller needs 409 ms. This means that the dynamics of the non-linear controller are more constant within the working range of the SEHB, which results in similar time constants. Fig. 8 is only an extract of the working range. For lower brake forces the P-controller is slower than the non-linear controller. This is not intended for the control of the SEHB as the time constants should be similar within the working range to meet the comfort requirements during braking of the train. The results confirm that the SEHB can be controlled by a simple state space controller once the system is linearized. Time limitations for the brake force build-up of a railway brake are typically at about 2 s.

ROBUSTNESS - During operation of the brake various parameters can change. For instance, the swiftness of the servo valve increases for smaller openings. Hence, investigations are carried out to verify the performance of the designed control algorithm with a significantly slower and faster servo valve (Fig. 9).

Fig. 9: Sensitivity concerning valve frequency

A rapid reaction servo valve does not affect the controllers step response adversely. The slow servo valve is parameterized at a frequency of 35 Hz and the non-linear controller shows constant dynamics.

In a second test, the loop gain of the SEHB control path is considered. Two parameters, $\mu$ and $Q_n$, are representative for the control path loop gain. Fig. 10 illustrates changes of the friction coefficient $\mu$ between the brake pads and the brake disc.

Fig. 10: Step response for different friction coefficients

A decrease of the friction coefficient is equivalent to a reduced loop gain. Therefore, the step response is protracted. An increased friction coefficient, on the other hand, shortens the time for the brake force build-up. To improve the performance, future work will focus on the automatic prediction of the friction coefficient.

A variation of the nominal valve flow influences the flow rate amplification and thereby the loop gain of the SEHB. A variation of the valves nominal flow at 35 bar pressure drop per control edge is shown in Fig. 11.

Fig. 11: Nominal valve flow

As the nominal valve flow is well-known, a variation of ±10% is carried out. Once again a reduced amplification, represented by the nominal valve flow, prolongs the time necessary for brake force build-up (90%) to 843 ms. On the contrary, examining the step response with a control valve of raised nominal flow, a fast response can be observed. For the case of a nominal flow of 22 l/min, the brake force overshoots the set point by 1.78% and has a peak time of 689 ms.

CONCLUSION AND OUTLOOK

This paper shows the design of a full state feedback control for the SEHB. In a first step a mathematical model of third order with one pressure build-up equation is described. The yielded model is input-output...
linearized and thereafter a state space controller is
designed. The calculation for the controller coefficients
on basis of the Ackermann Formula is presented.

As only one of the three variables of the state vector, the
brake actuator pressure $p_{BA}$, is available through
measurement, an observer is required. Therefore, a
Luenberger observer to estimate the unknown states of
the control valve is designed. The determination of the
Luenberger matrix coefficients is shown so as to assign
the observer poles left of the controller poles.

The performance of the non-linear controller with
observer is verified by simulation. The used simulation
model was validated by test rig measurements. Step
responses of the non-linear controller confirm a more
constant dynamic behavior when compared to a
proportional controller. Furthermore, the non-linear
controller performance for changing parameters of the
control path is successful.

The simulation results of the non-linear controller are
promising and future work will focus on the validation of
the controller by measurements with the SEHB
prototype. Moreover, an automatic prediction of the
friction coefficient is in need.

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**REFERENCES**

1. Lunze, J., “Regelungstechnik 2”, 5th ed., Springer,
3. Kühnlein, M., Liermann, M., Murrenhoff, H.,
   “Simplified Modelling of a Self-energising Hydraulic
   Brake”, Proceedings of the 6th FPNI – PhD
4. Ogata, K., “Modern Control Engineering”, 5th ed.,
5. Luenberger, D.G., “An Introduction to Observers”,
   IEEE Transactions on Automatic Control, Vol. AC-
   Hydraulische Antriebe”, 3rd ed., Shaker, Aachen,
7. Abel, D., “Regelungstechnik”, 33rd ed., Mainz,
8. N.N., “DIN EN 13452-1: Railway applications -
   Braking - Mass transit brake systems - Part 1:
   Performance requirements”, German-European
    Systems”, IEEE Transactions on Automatic Control,

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**ACRONYMS**

BA: Brake Actuator
NL: Non-Linear
SC: Supporting Cylinder
SEHB: Self-energizing Electro Hydraulic Brake