Observer based feedforward/feedback control of electro-pneumatic clutch systems

Theses of Ph.D. dissertation

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1 Motivation and Aim

Recently, there has been a growing need to further increase the dynamics of the clutch of commercial vehicles. Besides the cost reduction of the system has vital importance as well. Hence these make the clutch control systems an important part of the innovation of medium and heavy duty trucks. The requirements of the clutch control function are determined by the controllability of the torque, transmitted by the clutch. The transmitted torque is a function of the piston position of the clutch actuator realized by a disc spring, therefore the position control can provide the clutch torque control function.

In the last decade, several papers have been published on the topic of the position control of electro-pneumatic clutch actuators. These actuators are driven by proportional- or on/off valves, which yield that the control signals can be continuous and have to be quantized respectively.

One of the proposed control methods are the PID type controls, extended with self-tuning ability using parallel feedforward compensator, neural network or fuzzy PSD methods [1, 2, 3]. These controls are designed for clutch actuators with proportional valves. Other published methods are the switching controls, in which a control Lyapunov function is defined to achieve exponential stability in the operation domain using quantized control input, where the quantization came from applying on/off valves instead of proportional ones [4, 5, 6, 7]. Explicit model predictive control techniques are also used, where the minimization of a cost function in a finite horizon is performed off-line [8, 9, 10].

In the recent years, the flow cross section of the on/off solenoid valves, applied in clutch actuators, are increased to achieve increased performance. Obviously, this performance requirement is dictated by the additional clutch control functions to further improve the safety, the fuel efficiency and the maintenance period of the trucks. Although, increasing the cross section of the valves allows fast dynamics, but causes difficulties to the control, since the increased throughput of the valves changes the opening and closing dynamics and consequently reduces the potential of the fine application of the compressed air and through this the fine application of the torque transmitted by the clutch. Hence, there is an obvious development opportunity to improve the performance of the electro-pneumatic clutch control. This was the initial motivation for the research studies of the author.

Based on these facts, the aim of the research work summarized in the dissertation was to develop an appropriate control for electro-pneumatic clutch systems satisfying a prescribed control specification.

In order to reach the above described research aim, the dynamic model of the investigated electro-pneumatic clutch system [11, 12] has to be developed. The model can be set up from thermodynamical, mechanical, electronic and
magnetic first engineering principles including suitable constitutive relations by a systematic modelling procedure. The layout of the electro-pneumatic clutch system is shown in Fig. 1 (Chapter 2).

Figure 1: The layout of the electro-pneumatic clutch (EPC) system

Dynamic system models derived from first engineering principles are sometimes too detailed and complex for the given purposes. Therefore, a systematically derived model should often be simplified. The developed nonlinear dynamic hybrid model of the electro-pneumatic clutch system has been considered for simplification to obtain lower order models, on one hand for simulation and on the other hand for control design purposes. For this purpose a systematic simplification method has been applied for the detailed model of the electro-pneumatic clutch system (Chapter 3).

The state vector of the simplified model for control design purposes is obtained as:

\[ \mathbf{x}_{\mathcal{M}_2} = \begin{bmatrix} p_{ch} & v_{pst} & x_{pst} \end{bmatrix}^T, \]  

where \( p_{ch} \) is the pressure of the chamber, \( v_{pst} \) is the velocity of the piston and \( x_{pst} \) is the stroke of the piston.

The resulted control input vector contains the total mass flow rate of the valves:

\[ \mathbf{u}_{\mathcal{M}_2} = \sigma_v. \]
The disturbance input vector is written as:

\[ \mathbf{d}_{M_2} = [p_{amb} \ T_{amb}]^T, \]  

(3)

where \( p_{amb} \) and \( T_{amb} \) are the ambient pressure and temperature, respectively.

The output vector is obtained as follows:

\[ \mathbf{y}_{M_2} = [p_{ch} \ x_{pst}]^T. \]  

(4)

The state-space description of the model for control design purposes is written as follows:

\[
\frac{d\mathbf{x}_{M_2}}{dt} = \mathbf{f}_{M_2}(\mathbf{x}_{M_2}, \mathbf{d}_{M_2}) + \mathbf{g}_{M_2}(\mathbf{x}_{M_2}, \mathbf{d}_{M_2})u.
\]  

(5)

The coordinate functions are written as follows:

\[
f_{1,M_2} = \frac{-p_{ch} v_{pst} A_{pst}}{V_{ch}^d + x_{pst} A_{pst}},
\]  

(6)

\[
f_{2,M_2} = \left( \frac{(p_{ch} - p_{amb}) A_{pst} - v_{pst} k_{pst} - F_l(x_{pst})}{m_{pst}} \right),
\]  

(7)

\[
f_{3,M_2} = v_{pst},
\]  

(8)

\[
g_{1,M_2} = \frac{R T_{amb}}{V_{ch}^d + x_{pst} A_{pst}},
\]  

(9)

\[
g_{2,M_2} = 0,
\]  

(10)

\[
g_{3,M_2} = 0.
\]  

(11)

The measured output is written as the following linear equation:

\[ \mathbf{y}_{M_2} = \mathbf{C}_{M_2} \mathbf{x}_{M_2}, \]  

(12)

where

\[ \mathbf{C}_{M_2} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 0 & 1 \end{bmatrix}. \]  

(13)

The performance output is generated from the measured output by the following simple equation:

\[ \mathbf{z}_{M_2} = [0 \ 1] \mathbf{y}_{M_2}. \]  

(14)

The parameters can be found in Tab. 1.

The dynamic analysis of the control oriented model has been executed and the dynamic properties related to control design are examined (Chapter 4).

Based on the control aims and the input signal constraints an observer based feedforward/feedback controller has been developed. For the feedforward and
Table 1: List of parameters

<table>
<thead>
<tr>
<th>Parameter name</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
<th>Confidence</th>
</tr>
</thead>
<tbody>
<tr>
<td>Specific gas constant</td>
<td>R</td>
<td>287.14</td>
<td>J/kgK</td>
<td>K</td>
</tr>
<tr>
<td>Area of piston</td>
<td>A_{pst}</td>
<td>0.0227</td>
<td>m²</td>
<td>PK</td>
</tr>
<tr>
<td>Dead volume of chamber</td>
<td>V_{ch}</td>
<td>5.5982 · 10^{-4}</td>
<td>m³</td>
<td>PK</td>
</tr>
<tr>
<td>Lumped mass</td>
<td>m_{pst}</td>
<td>9.3922</td>
<td>kg</td>
<td>PK</td>
</tr>
<tr>
<td>Damping coefficient of piston</td>
<td>k_{pst}</td>
<td>2251.3825</td>
<td>Ns/m</td>
<td>UK</td>
</tr>
</tbody>
</table>

the feedback controllers several conceptions have been considered, such as the static- and dynamic mass flow rate decomposition approach, linear quadratic (LQ), robust $H_\infty$ and sliding mode control (SMC) approach. The obtained closed loops are investigated by extensive simulation, bench- and vehicle tests to verify the properties of the different control concepts (Chapter 5).

2 Methods and Tools

The model building of the electro-pneumatic clutch system has been carried out by executing a systematic modelling procedure [13]. First of all, the objective and the aim of modelling have to be defined, which highly influence the final form of the model. For dynamic models differential equations are required, which can be obtained from conservation balances, while they have to be supplemented with algebraic equations to obtain a solvable set of equations. The modelling assumptions have to be taken into consideration in a consistent way throughout the whole model development procedure. Since the modelled system exhibits discrete-continuous or hybrid behavior, the developed model has similar properties as well.

In order to obtain a state space model, the possibility of substituting the algebraic equations into the differential equations has to be investigated. If all algebraic equations can be substituted into the differential equations, then the final result of the modelling is a set of differential equations, which can be transformed into nonlinear state space form.

Information related to model validation can be obtained from laboratory measurements. For this purpose a concentric type electro-pneumatic clutch actuator from ZF SACHS has been installed on a test bench at the Knorr-Bremse R&D Center Budapest. The data acquisition system provides the measurements of the dynamic transient processes. The measurable quantities for the validation have been the electric current and terminal voltage of the solenoid magnet valves, chamber pressure, piston position, supply voltage and supply air pressure. The

\[^1\text{K: known, PK: partially known, UK: unknown}\]
validation of the dynamic nonlinear hybrid model was performed by measurements of the open loop system.

A model for control design purposes should retain all major dynamic characteristics of the real plant (such as its stability and main time constants, which are needed to be invariant under the simplification process) but omit all details that are weakly represented in the state variables and not related to the control aims.

There are several methods proposed in the literature for performing model simplification and reduction in different ways to obtain a model with suitable size and complexity. These methods can be classified based on the underlying engineering knowledge used during model simplification. The applied simplification method uses engineering insight and operation experience to leave out state variables based on the dynamics of the original state variables with physical meaning [14, 15].

The dynamic analysis of the prepared model has primary importance in control design. Dynamic analysis means checking of basic dynamic properties of the system: such as controllability, observability and stability. The proof of the stability of the zero dynamics is essential for output tracking. Moreover, the sensitivity of the state variables regarding the model parameter uncertainty plays an important role in the control design as well.

The controllability and observability properties have been investigated by differential geometric approaches [16]. The asymptotic stability is proved by Lyapunov’s method and the invariant set theorem [17, 18, 19]. The zero dynamics is determined via the relative degree and the sensitivity is computed using the partial derivatives of the coordinate functions regarding to the model parameters [20].

Based on the control aims and the input constraints, the controller structure is selected as an observer based feedforward/feedback type controller (see Fig. 2). It includes three blocks in form of a state observer supplying the unmeasurable states, a feedforward controller unit producing the mass flow rate control of the valves, and a model based feedback controller unit that provides piston position control. The main tasks of the controller design are the determination of the mentioned three blocks. For these blocks several conceptions are considered and these conceptions are compared with each other.

The complete modelling and controller design procedure has been carried out in the MATLAB/Simulink environment. For measurement and control hardware dSpace Autobox with DS1005 baseboard and DS2211 multi I/O measurement card have been used.
3 New Scientific Results

1. Thesis Nonlinear dynamic hybrid model of electro-pneumatic clutch systems (Chapter 2) ([P1])

The nonlinear dynamic hybrid model of electro-pneumatic clutch systems ($\mathcal{M}_0$) considered as a mixed thermodynamical, mechanical and electro-magnetic system has been built and verified using a systematic modeling methodology. It has been shown that the model exhibits the following special properties:

i. The nonlinear dynamic hybrid model of electro-pneumatic clutch systems is given by a set of nonlinear ordinary differential-algebraic equations.

ii. It has been shown that the 16 state equations of the nonlinear hybrid dynamic model can be written into hybrid nonlinear state-space form.

$$\frac{d x^{(k)}_{\mathcal{M}_0}}{dt} = f^{(k)}_{\mathcal{M}_0} \left( x^{(k)}_{\mathcal{M}_0}, u^{(k)}_{\mathcal{M}_0}, d^{(k)}_{\mathcal{M}_0} \right),$$

where $k : \mathbb{R}^n \to \mathbb{N}$ is a piece-wise constant switching function mapping from the state-space to the finite integer set $\mathbb{N} = \{1, 2, \ldots 746496\}$. The coordinate function depends also on the state vector $x_{\mathcal{M}_0}$, the control input vector $u_{\mathcal{M}_0}$ and the disturbance vector $d_{\mathcal{M}_0}$.

iii. The model output is linear with respect to the state and disturbance vectors as:

$$y_{\mathcal{M}_0} = C_{\mathcal{M}_0} x_{\mathcal{M}_0} + E_{\mathcal{M}_0} d_{\mathcal{M}_0},$$

where $C_{\mathcal{M}_0}$ and $E_{\mathcal{M}_0}$ are constant matrices.

iv. It has been shown that the developed model structure is valid for clutch systems applied with concentric and forked lever type electro-pneumatic...
clutch actuators as well, only the values of model parameters differ from each other.

2. Thesis  Simplification of the electro-pneumatic clutch model (Chapter 3)([P2])

A systematic model simplification method has been applied to the nonlinear dynamic hybrid model of electro-pneumatic clutch systems to obtain a lower order model suitable for control design purposes ($M_2$).

The obtained simplified model for control design purposes has the following properties:

i. The dimension of the state vector has been reduced from 16 to 3. The dimension of the control input vector has been reduced from 4 to 1. The dimension of the disturbance vector has been cut to 2 from the original 5, while the output vector has been reduced from 8 to 2. The number of the parameters has been reduced from 83 to 8.

ii. All retained system variable entries have preserved their physical meaning.

iii. The discrete switching terms have been eliminated completely.

iv. The control oriented model has been rewritten into standard input affine state-space form as:

$$\frac{dx_{M_2}}{dt} = f_{M_2}(x_{M_2}, d_{M_2}) + g_{M_2}(x_{M_2}, d_{M_2})u.$$ 

The coordinate functions depend on the state vector $x_{M_2}$ and the disturbance vector $d_{M_2}$.

v. The retained disturbance variables are not measured thus the output equation is written as follow:

$$y_{M_2} = C_{M_2}x_{M_2},$$

where $C_{M_2}$ is a constant matrix.

3. Thesis  Dynamic properties of the control oriented electro-pneumatic clutch model (Chapter 4)([P3])

The dynamic properties of the control oriented electro-pneumatic clutch model has been analyzed. It has been shown that the model exhibits the following dynamic properties:
i. The control oriented model is jointly controllable and observable with the selected input and output, hence the model is minimal.

ii. The zero dynamics is asymptotically stable, since the model has a maximum relative degree \( r = n \).

iii. The control oriented model is uniformly globally asymptotically stable. The stability criterion is obtained as follows:

\[
\frac{d}{dx_{pst}} F_l(x_{pst}^*) + \frac{p_{ch} A_{pst}^2}{V_{ch} + A_{pst} x_{pst}^*} > 0.
\]

iv. The model parameters and disturbance inputs can be arranged to the following list according to the sensitivity of the states variables, regarding the change of these model components: \( A_{pst}, V_{ch}, p_{mab}, T_{amb}, m_{pst} \) and \( k_{pst} \).

4. **Thesis** Specification of the clutch control problem and design of an observer based cascaded feedforward/feedback controller for electro-pneumatic clutch systems (Chapter 5)[P4],[P5],[P6],[P7],[P8],[P9],[Pa],[Pb])

The control requirements and constraints have been defined for the state of the art clutch systems. Based on the system model an observer based cascaded mass flow rate / piston position controller structure has been developed, where the mass flow rate control part provides I/O linearization for the applied on/off solenoid magnet valves and the position control part ensures the position tracking, stabilizing and disturbance rejection task. For the estimation of the unmeasured state a high-gain observer has been designed to feed both of the position control and the flow rate control parts. The properties of the closed loop systems investigated by simulation, bench- and vehicle tests have lead to the following observations:

i. A nonlinear type sliding mode controller has been designed and tuned for the nonlinear dynamic control oriented model of electro-pneumatic clutch systems. In order to decrease the load of the computing device the reduction of the equivalent control part to zero \( (u_{eq} = 0) \) is executed. Based on simulation results it has been found that the simplified sliding mode controller fulfills the dynamic requirements.

ii. The cascaded dynamic mass flow rate / simplified sliding mode position control can fulfill the requirements in case of simulation with the detailed nonlinear dynamic hybrid model of electro-pneumatic clutch systems.
iii. For the bench tests the cascaded dynamic mass flow rate / simplified sliding mode position control is further extended with the designed high-gain observer to get the observer based feedforward/feedback control structure and it has been shown that the requirements have been fulfilled in real environment as well.

iv. Vehicle tests have been executed with the designed high-gain observer based cascaded dynamic mass flow rate / simplified sliding mode controller and it has been found that the system can fulfill the requirements and ensures good driver comfort.

4 Publications Directly Related to the Thesis


5 Submitted patents


6 Publications Partially Related to the Thesis


7 Application Aspects

The modelling and controller design methods used for the electro-pneumatic clutch system can be applied to other electro-pneumatic systems. Moreover, the presented methods can be used to investigate the enhancement possibilities of an existing electro-pneumatic clutch actuator to improve the control performance. The nonlinear dynamic hybrid model of the electro-pneumatic clutch system provides an effective way to improve the mechanic or electric design of the actuator.

The developed observer based feedforward/feedback control structure, the mass flow rate control, the piston position control and the high-gain observer have been applied to electro-pneumatic gear box actuator unit as well [21, 22].

References


