Computational and Experimental Investigation on Dynamics of Electric Braking Systems

Viktor SZENTE, János VAD
Department of Fluid Mechanics
Budapest University of Technology and Economics
Bertalan Lajos u. 4 - 6., H - 1111 Budapest, Hungary
Email: szente@simba.ara.bme.hu, vad@simba.ara.bme.hu

Gábor LÓRÁNT, Ansgar FRIES
KNORR-BREMSE Systems for Commercial Vehicles Ltd.
Research and Development Center
Major u. 69., H - 1119 Budapest, Hungary
Email: gabor.lorant@knorr-bremse.com

Abstract
Electro-pneumatic (EP) components are frequently used in brake systems of commercial vehicles. The simulation of EP brake systems is of great importance in order to understand their dynamics for developing a control logic being robust but fulfilling the modern functional demands. On the other hand, the simulation aids the design of EP components being able to execute the commands of a precision control. The paper presents a flexible computational simulation tool being applied in industrial research and development related to complex mechatronics in brake systems for commercial vehicles. The Electric Braking System (EBS) case study presented herein comprises an air supply unit, an EBS modulator, piping, a diaphragm brake chamber, and the connected brake mechanism. The simulation environment is AMESim® 3.0. Considering the complexity of the EP components and the related phenomena, special models have been elaborated for the solenoid valves, piping, and the diaphragm brake chamber. The simulation results show good agreement with measurement data. The comparative numerical and experimental study confirmed that the simulation tool can be effectively used in design, research and development of EP brake systems.

Keywords: AMESim, brake test bench, dynamic simulation, EBS, solenoid valve
1 Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>$A$</td>
<td>orifice cross-section</td>
<td>$[m^2]$</td>
</tr>
<tr>
<td>$C_m$</td>
<td>orifice flow parameter</td>
<td>[-]</td>
</tr>
<tr>
<td>$C_q$</td>
<td>orifice flow coefficient</td>
<td>$[\sqrt{K/(m/s)}]$</td>
</tr>
<tr>
<td>$m$</td>
<td>mass flow rate</td>
<td>$[kg/s]$</td>
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<tr>
<td>$p$</td>
<td>relative pressure</td>
<td>$[bar]$</td>
</tr>
<tr>
<td>$t$</td>
<td>time</td>
<td>$[s]$</td>
</tr>
<tr>
<td>$T$</td>
<td>temperature</td>
<td>$[K]$</td>
</tr>
</tbody>
</table>

Subscripts

- $0$: reference value
- $ch$: brake chamber
- $out$: EBS modulator output port
- $supply$: air supply
- $up$: upstream

2 Introduction

Pneumatic Electric Braking Systems (EBS) are getting wide-spread in commercial vehicles. Such systems ensure a fast-response, intelligently controlled braking process, fulfilling several demands related to brake and vehicle dynamics [1]. Considering that the EBS is controlled at a sampling frequency in the order of magnitude of 100 Hz, dynamic behavior of the electro-pneumatic (EP) components must be taken into account in design of EBS system composition and control if appropriate brake operation is to be guaranteed. This is emphasized by the fact that the characteristic time scale of dynamic phenomena appearing in EBS operation is in the order of magnitude of milliseconds. Examples for such time scales are: switch-on and switch-off times of solenoid valves [2] controlling relay valves; response time and hysteresis of relay valves [3]; time necessary for fluid mechanical wave phenomena (shock waves) to travel along pneumatic pipes [4].

For development of competent, fast-response EP brake systems satisfying increasingly complex and strict demands, dynamic effects and unsteady physical phenomena must be taken into account. For this purpose, it is of great significance to elaborate a flexible simulation tool to predict the dynamic behavior of EBS. Such computational simulation tool can be extensively used concerning EBS applied in automotive industry: establishment and test of new operational concepts; design optimization of hardware and control parameters; hardware-in-the-loop simulation; prediction of EBS behavior under extreme operational circumstances; detailed study on complex physical phenomena which are difficult to survey by theoretical or experimental means.

The paper presents a computational simulation tool used for prediction of dynamic behavior of EP brake systems. The capability of the simulation tool is demonstrated for a simplified case study. The comparative evaluation of simulation results and experimental data confirms that the presented simulation tool can be reliably applied in design and R&D in the field of electro-pneumatic brake systems.

3 Case study set-up

The scheme of EBS case study set-up presented herein is shown in Figure 1. It comprises an air supply unit, an EBS modulator, piping, a diaphragm brake chamber, and the connected brake mechanism. A realistic EBS system applied in modern motor vehicles is rather complex including also other components, larger number of elements,
and the related complex control system. Contrarily, this simplified case study was considered reasonable to illustrate the capabilities of the simulation tool for resolution of EBS subsystem dynamics. Theoretically, the entire EBS system and the related vehicle dynamics can be modeled with use of the simulation software presented herein.

**Figure 1:** EBS case study set-up for simulation and experiments

**Figure 2** represents a functional scheme of the EBS modulator. The modulator comprises in simulation a relay valve; a filter; a silencer; LOAD and EXHAUST solenoid valves; an in-built pressure sensor measuring the pressure on the modulator output port $p_{out}$ and acting as a signal encoder in the pressure control loop; piping and bores connecting the modulator components; and pneumatic ports (air supply port SUPPLY, output port OUT, and a port towards the atmosphere through the silencer, ATM in Fig. 2.). A realistic EBS modulator also comprises a backup solenoid valve. In absence of solenoid excitation (i.e. in case of failure of control electronics), the backup valve is opened and connects the brake pedal directly to the relay valve via a pneumatic contact. If the control electronics operate properly, the backup valve is continuously energized and stays in the closed state. In this normal operating condition, a braking action is controlled by means of the LOAD and EXHAUST solenoid valves. Given that the backup valve has no role for the test cases presented here, it is neither considered in simulation nor represented in Fig. 2.

The air supply unit and the diaphragm brake chamber are connected to the EBS modulator on the SUPPLY and OUT ports, respectively, via flexible piping.

The aim of the EBS modulator is to provide a controlled pressure $p_{out}$ (see Fig. 1) on the OUT port. $p_{out}$ appears on the output port of the relay valve, which is supplied with compressed air from the SUPPLY port. The input pressure signal for the relay valve is provided by the LOAD and EXHAUST solenoid valves. Normally an electronic...
pressure control system supplies the valve commands. For the case study presented herein, the pressure control loop is not operated but direct LOAD and EXHAUST solenoid valve commands are applied. Such simplification has been carried out because this paper focuses on investigation of the pneumatic behavior of the system.

The valve commands appear as DC excitation voltage signals applied to the solenoids. If the LOAD solenoid valve is excited, an ordering pressure signal is applied to the relay valve from the SUPPLY port, and the air from the SUPPLY port is loaded by the relay valve to the pressure output port increasing $p_{out}$. From the OUT port, the air is loaded to the diaphragm brake chamber via a piping. The brake actuator operates the brake mechanism, and the brake force increases.

If the EXHAUST solenoid valve is excited, the ordering pressure signal applied to the relay valve decreases (the control air from above of the relay valve piston is released). Accordingly, the air from the diaphragm brake chamber is exhausted towards the atmosphere ATM through the silencer. As a consequence, the brake force decreases.

If none of these valves is excited, the instantaneous pressure conditions are retained and the air is held back in the brake chamber (HOLD state).

A controlled braking process is realized through rapidly repeated, controlled LOAD, HOLD and EXHAUST commands.

![Figure 2: Functional representation of the EBS modulator](image)
4 Simulation tool and modeling

According to the need on the behalf of EP design and R&D, the following requirements have been formulated for the simulation tool:

- System approach, guaranteeing a physically appropriate coupling of electro-
  dynamical, mechanical, and fluid dynamical subsystems and the related control;

- Modularity, making possible the consideration of EBS components as independent
  modules, thus ensuring the applicability of the component models in other modeling
  topology (e.g. model extension for consideration of vehicle dynamics);

- Possibility for modeling on different levels of complexity, making possible a
  phenomenological description of certain sub-elements for modeling simplification,
  and offering studies on significance of certain physical effects;

- Flexible variability of model topology, making possible the extension and refinement
  of the model by including new components in the EBS development phase and by
  neglecting sub-models proven to be insignificant from the viewpoint of EBS
  operation;

- Flexible variability of model parameters for parameter optimization and sensitivity
  studies.

For fulfilment of these demands, the simulation software AMESim® (Advanced
Modeling Environment for Simulations of engineering systems), version 3.0 has been
used. Including a number of ready-made sub-model elements structured in libraries, this
simulation environment makes possible a convenient and effective modification,
extension, and improvement of the EBS simulation software. This software proved its
appropriateness in simulation of systems related to automotive industry [5][6][7].

The simulation tool has been elaborated at the Department of Fluid Mechanics (DFM),
Budapest University of Technology and Economics. **Figure 3** represents the AMESim®
model of the test case presented herein, in a topology similar to that of Fig. 1. Most sub-
models have been built up using the commercially available AMESim® sub-
models taken from the mechanical, pneumatic, control, and hydraulic model libraries.
Sub-models of certain system components of especially complex behavior had to be
elaborated by the DFM research team using the model editing tool AMESet®:

- Solenoid valve model, receiving DC excitation voltage, upstream and downstream
  total pressures and total temperatures as input variables, and supplying mass flow
  rate and enthalpy flow rate as output variables. The experimentally verified model of
  coupled electro-dynamic and mechanical subsystems is based on [8]. It considers a
  varying magnetic resistance and inductance of the electro-dynamic subsystem,
  depending on the valve body position.

- Gas dynamical models for pneumatic pipes [9], modeling pipe flow in the entire
  physically possible Mach number range, and resolving wave phenomena such as
  shock waves.

- Experimentally verified diaphragm brake chamber model, representing the nonlinear
  push rod position - output force characteristics. In order to resolve such behavior, a
  complex model had to be elaborated for the diaphragm. i/ The central (undeformed)
  part of the diaphragm was modeled as a traditional brake cylinder ("piston effect" in
The nonlinear brake chamber characteristics are related to the following (ii/ and iii/) sub-models: ii/ The diaphragm is made of fiber-enforced rubber composite material. Its conical part performs significant deformation, depending on chamber pressure and push rod position. The pressure force forwarded by the conical part to the push rod depend on the actual diaphragm shape. For consideration of this force, a special shape function and pressure force model was elaborated ("diaphragm shape effect"). iii/ According to its flexible deformation, the diaphragm acts also as a cup spring ("cup spring effect"). The diaphragm was modeled as a parallel connection of models i/, ii/ and iii/. The input variable of the diaphragm brake chamber is the chamber pressure, and the outputs are the output force and push rod position.

The parameters of the EBS case study set-up have been precisely determined by means of basic geometrical, volume, mass, force, and electro-dynamic measurements. Some parameters have been specified on the basis of technical documentation. Sensitivity studies have been carried out on parameters estimated with higher uncertainty, such as Coulomb friction forces, or flexibility module of the diaphragm in the brake chamber. It was found that changing these parameters within the presumed uncertainty ranges influence the simulated pressure values concerning the following aspects:

- Temporal shift, usually one order of magnitude less than the sampling time applied in EBS control,
- Change in magnitude of unsteady pressure, usually less than 3 percent of reference pressure $p_0$ (see Chapter 6).

Such changes are within the variance of experimental results derived from repeated measurements for each test case. Therefore, setting of these parameters to the mid of the presumed uncertainty ranges was considered as an appropriate way of parametrization.

5 Experimental technique

For experimental verification of the simulation results, the case study set-up (see Fig. 1) has been realized on a brake test bench in the Pneumatic Laboratory at the Budapest Research and Development Center of KNORR-BREMSE Systems for Commercial Vehicles Ltd. It must be emphasized that the pressure is the most important quantity from the viewpoint of brake system operation. Accordingly, the experimental verification of simulation results was focussed on pressure measurements. Among its several capabilities, the following features of the test bench and the related components have been utilized for the experiments presented herein:

- Air supply unit. In other tests, a controlled, stabilized pressure is maintained in the reservoir of the air supply unit from a local compressed air network, according to air consumption. At the beginning of each test reported in the paper, the air supply unit has been loaded to a maximum initial pressure $p_0$ and the valve toward the air network has been closed. With this condition, the air supply unit was modeled in simulation simply as a single pneumatic reservoir. $p_0$ has been precisely measured by means of a computer-controlled pressure transducer. The accurate values of $p_0$, the atmospheric pressure, the air temperature in the air supply unit and the ambient temperature were used as initial and boundary conditions for simulation. These values were kept at identical level for each test case.
- Flexible pipe connecting the air supply unit to the SUPPLY port of the EBS modulator. For each test, this piping is loaded to $p_0$ in the initial state.

- EBS modulator. The pressure control loop of the modulator has been opened.

- Computer-controlled solenoid valve drive unit, maintaining a continuous excitation to the backup solenoid valve (thus detaching the foot brake pedal pneumatic subsystem from the set-up) and supplying direct LOAD, HOLD and EXHAUST commands to the solenoid valves in the open pressure control loop.

- Fast-response pressure sensors and the related computer-controlled data acquisition system for measurement of air supply pressure ($p_{\text{supply}}$, not presented in the diagrams) and pressure in the brake chamber ($p_{\text{ch}}$).

- The in-built pressure sensor of the EBS modulator and the related computer-controlled data acquisition system for measurement of pressure at the outlet port of the EBS modulator ($p_{\text{out}}$). The investigation of this pressure is especially important given that this pressure signal is utilized in the pressure control loop.

- Flexible pipe connecting the EBS modulator OUT port to the diaphragm brake chamber.

- Diaphragm brake chamber.

- Compression spring mechanism connected to the push rod of the diaphragm brake chamber, modeling the brake mechanism. The compression spring is of linear characteristics. At the end of the stroke, the threads of the spring contact each other, thus representing a spring of very high stiffness. Such progressive spring characteristics model suitably a realistic brake mechanism.

6 Test cases, test results and discussion

In order to test the capabilities of the simulation tool for resolution of EBS dynamics, four test cases are presented herein. These tests approach well the EBS operation in practice and are considered as a representative survey on unsteady electro-dynamic, mechanical and fluid dynamic processes. The test cases are as follows:

- LOAD – EXHAUST: loading and exhausting the brake chamber by means of single LOAD and EXHAUST commands,

- GRADUAL LOAD – EXHAUST: loading the brake chamber gradually by means of rapidly repeated LOAD and HOLD commands, and exhausting the brake chamber by means of a single EXHAUST command,

- LOAD – GRADUAL EXHAUST: loading the brake chamber by means of a single LOAD command, and exhausting the brake chamber by means of rapidly repeated EXHAUST and HOLD commands,

- PULSED: rapidly repeated LOAD and EXHAUST commands with no intermediate HOLD commands.
Figure 3: AMESim model of the EBS case study
Figures 4 – 7 show the comparative diagrams of nondimensionalized time functions for the simulated and measured $p_{\text{out}}$ and $p_{\text{ch}}$ relative pressures for the LOAD – EXHAUST, GRADUAL LOAD – EXHAUST, LOAD – GRADUAL EXHAUST and PULSED test cases, respectively. For a lifelike representation of test results, the relationship between the variables is presented in dimensionless form. The time scale has been nondimensionalized by a reference time period $t_0$, in which the pressures in the air supply reservoir and in the brake chamber are equalized during a single LOAD process. The relative pressures are nondimensionalized by the initial relative pressure in the air supply reservoir $p_0$. The LOAD and EXHAUST solenoid valve commands (appearing as DC voltage signals applied to the solenoids) are indicated at the top of the figures using black and grey bars, respectively, with intermediate HOLD states as appropriate.

The agreement between the simulated and measured time-pressure functions is generally good, from qualitative as well as quantitative points of view. The simulation follows $p_{\text{out}}$ related to single LOAD and EXHAUST commands with especially high accuracy (Fig. 4, second part of Fig. 5, first part of Fig. 6). The departure is below the reproduction limit of experiments. For the gradual LOAD and EXHAUST actions (first part of Fig. 5, second part of Fig. 6) as well as for the PULSED test (Fig. 7), some parts of simulated and measured $p_{\text{out}}$ functions show a higher departure. This observation is commented in Subchapter 7.2.

A higher departure can be generally observed between simulated and measured $p_{\text{ch}}$ values. This is probably due to the approximate manner of diaphragm modeling.

The R&D activity for EBS systems regards the EBS modulator and the related control. Accordingly, the good agreement in $p_{\text{out}}$ considered to be the most important verification of the simulation.

7 Notes on solenoid valve modeling - fluid mechanical aspects

Sensitivity studies on the system parameters pointed out that the dynamics of the solenoid valves influence critically the dynamic behavior of the entire EBS system. For this reason, it is worthwhile to comment the simplified fluid dynamical solenoid valve sub-models in a more detailed manner. The discussion on modeling solenoid valve dynamics is divided into two parts: flow transmission characteristics and valve body dynamics.

7.1 Flow transmission characteristics

The through-flow area of both solenoid valve was modeled as a pneumatic orifice with a geometrical cross-section controlled by the valve body position. The kinetic energy of the upstream flow was neglected. The flow through the solenoid valve was approached as series of short-term stationary sub-processes. The mass flow rate $m$ through the orifice is a function of upstream absolute pressure $p_{\text{up}}$, upstream temperature $T_{\text{up}}$, orifice cross-section $A$, flow coefficient $C_q$ and mass flow parameter $C_m$ [10]:

$$m = A C_q C_m \frac{p_{\text{up}}}{\sqrt{T_{\text{up}}}}$$

(1)
Figure 4: Time function of relative pressures: LOAD – EXHAUST test

Figure 5: Time function of relative pressures: GRADUAL LOAD – EXHAUST test
Figure 6: Time function of relative pressures: LOAD – GRADUAL EXHAUST test

Figure 7: Time function of relative pressures: PULSED test
The pressure ratio (ratio of downstream and upstream absolute pressures) was generally estimated to be sub-critical at opening of the valves, so it was assumed that after opening the valves, a throttled expansion occurs ($C_m$ has a maximum value) and a sonic flow develops at the narrowest cross-section. According to this assumption, the narrowest available cross-section of solenoid valve orifice was considered as $A$ in each simulation step as the actual cross-section. At the beginning of valve body stroke, the cylindrical surface at the edge of the valve body was the narrowest cross-section. As the valve body was displaced further on, the cylindrical bore in the axis of the valve settle became the narrowest cross-section. As a first approach, the through-flow cross-sections were modeled geometrically as sharp edged orifices. Accordingly, the flow coefficient $C_q$ was determined as a function of pressure ratio using the Perry polynomial [10]. The $C_q$ coefficient determined this way has been multiplied by a shape factor higher than unity considering that the realistic through-flow geometry is not a sharp-edged orifice but ensures a limited flow contraction. As the diagrams show, such phenomenological model is acceptable for modeling the flow transmission especially if the valve body is in a steady state (good agreement in steepness of simulated and measured $p_{out}$ curves for single LOAD and EXHAUST commands).

7.2 Valve body dynamics

In the simulation presented herein, only the mechanical and magneto-dynamic forces acting on the valve body were considered, using the model described in [8]. The effect of fluid mechanical forces acting on the valve body, such as unsteady momentum forces, pressure forces, and jet momentum forces, is completely neglected at the present state of investigation. For the gradual LOAD and EXHAUST actions and the PULSED test, the departure between simulated and measured $p_{out}$ time functions may be partly related to this simplification (and also to $C_q$ influenced by unsteady flow phenomena).

7.3 Future developments

For a more accurate and comprehensive modeling of solenoid valve action in EBS systems, the following efforts are planned in the future, based on three-dimensional, unsteady Computational Fluid Dynamics studies on the solenoid valves, and also involving advanced measurement techniques:

- Consideration of actual flow cross-sections during valve body displacement, precisely following the transition from sonic to subsonic flow (not only the narrowest cross-section is to be considered during the entire through-flow process),
- Refinements in modeling flow coefficient $C_q$, also including unsteady flow phenomena,
- Consideration of fluid mechanical forces acting on the valve body (unsteady momentum forces, pressure forces and jet momentum forces).

8 Conclusions

A flexible computational simulation tool has been presented herein, being capable for prediction on dynamic behavior of complex Electric Brake Systems applied in commercial motor vehicles. The appropriateness of the simulation tool has been confirmed through dynamic pressure measurements in an experimental setup with
topology identical to that of the simulated case. The simulation results showed generally good agreement with measurements. It has been concluded that the simulation tool can be favorably used in design, research and development of electro-pneumatic brake systems, regarding the hardware elements as well as the related control system.

The simplified phenomenological model applied to predict the flow through the solenoid valves was considered as a reasonable compromise to model the EBS dynamics relatively simply but properly. For an in-depth understanding of solenoid valve physics, these models must be refined from the viewpoints of flow transmission characteristics as well as valve body dynamics.

Acknowledgement

Authors acknowledge the support by the Hungarian Ministry of Education within the framework of Applied Research and Development Project No. AKFP ALK-00026/98.

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