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GAS DYNAMIC PIPE FLOW EFFECTS IN CONTROLLED PNEUMATIC SYSTEMS – A SIMULATION STUDY

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Abstract

Electro-pneumatic modulators (EPMs) are widely used in controlled pneumatic systems. Their aim is to ensure a controlled operating pressure modulated at a high accuracy and high temporal resolution. This paper reports a computational case study representing the influence of gas dynamic pipe flow effects on the operation of a pneumatic system to be controlled by means of an EPM. The simulation has been carried out with use of simulation software AMESim version 3.01. Given that no standard gas dynamic pipe model and electro-dynamically-relevant solenoid valve model are included in the present version of the simulation software, such models had to be elaborated by the authors. The simulation studies reveal that the self-developed gas dynamic pipe model resolves properly the wave effects and flow fluctuations expected from a realistic pneumatic pipe performing high velocity flow. The simulation must accurately resolve such phenomena if the computational study aims to provide an aid to the design of a proper pressure control loop. It has been concluded that a gas dynamic pipe model, covering reliably the physically possible entire Mach number range, is essential in simulation of pneumatic systems.

Keywords: electro-pneumatic modulator, pneumatic fluid power, dynamic simulation, gas dynamic pipe model, solenoid valve.

1. Introduction

Electro-pneumatic modulators (EPMs) are widely used in controlled pneumatic systems, e.g. in pneumatic brake systems of commercial vehicles [1] [2]. Their aim is to ensure a controlled operating pressure in an actuator chamber, modulated at a high accuracy and high temporal resolution, according to the application demands. In a usual set-up, the EPM maintains a controlled pressure in an actuator chamber (e.g. working chamber of a pneumatic cylinder) connected to the output port of the modulator via a pneumatic pipe. The modulator is supplied with compressed air at its input port by an air supply unit, considered as a pressure source in the study presented herein. The pressure is controlled by solenoid valves integrated in the EPM.

An example for application of EPM is a pneumatic braking system used in commercial vehicles. In such systems the actuator can be found next to the braking device on the wheel body (e.g. a disc brake), and the modulator is located on the chassis. The modulator is connected to the actuator with a flexible pneumatic pipe, which is long enough that the wheel movement does not stress the pipe walls too much.

In a simplified layout, the EPM consists of the following components, as illustrated in *Fig. 1*:

- A small-scale internal chamber (CH) connected to the output port (OUT) of the EPM. The output port (OUT) is connected to the actuator chamber via a pneumatic pipe (see also *Fig. 2* later).
- A LOAD solenoid valve connecting the air supply (SUPPLY) and the internal chamber (CH). The LOAD valve offers a potential for loading the actuator chamber, thus increasing the actuator pressure if prescribed by the control.
- An EXHAUST solenoid valve connecting the internal chamber (CH) and the atmosphere (ATM). The EXHAUST valve offers a potential for exhausting the actuator chamber towards the atmosphere, thus reducing the actuator pressure if prescribed by the control.
- A pressure sensor (PS) measuring the pressure in the internal chamber (CH). The measured pressure signal serves as a feedback signal for the pressure control loop electronics supplying the solenoids with appropriate commands.

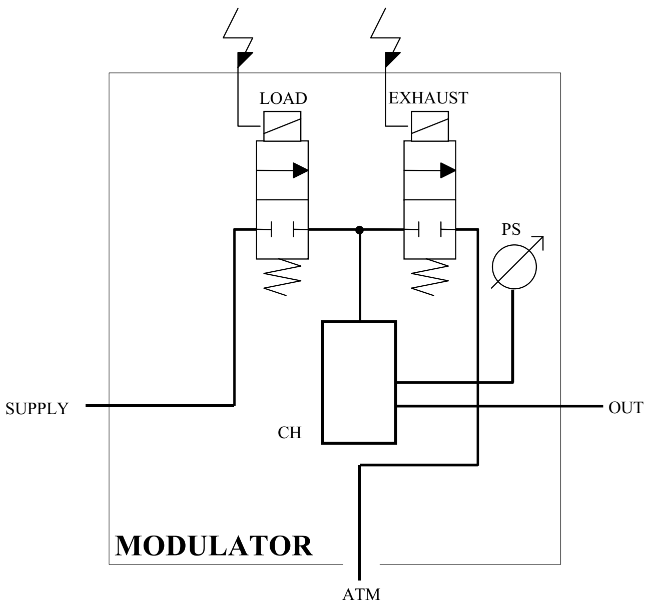

Fig. 1. Functional representation of the electro-pneumatic modulator

The electro-pneumatic components such as the solenoid valves (LOAD, EXHAUST) and the pressure sensor (PS) are physically integrated in the same EPM unit and are connected to the same integrated circuit panel included in the EPM casing. This is the reason why the pressure measured in the internal chamber (CH) is used as feedback signal, although the pressure control aims to realize a controlled pressure in the actuator chamber. This compromise necessitates the consideration of dynamics of the pipe connecting the modulator output port (OUT) and the actuator chamber in design of the pressure control loop.

In the past years, a number of investigations has been carried out on the dynamic behaviour and modelling of solenoid valves [3] [4], including also their control aspect [5]. The gas dynamic behavior of pneumatic pipes is well-understood [6]. Strategies have been elaborated for consideration of gas dynamic pipe flow effects in pressure control [7]. However, no detailed pressure control models are presented incorporating realistic solenoid valve models (resolving mechanical, electro-dynamic and fluid mechanical phenomena) and realistic pneumatic pipe models (resolving gas dynamic flow effects), and simulating the interaction of these elements.

The present paper aims to introduce a novel simulation tool for investigation of EPM dynamics, with special regard to gas dynamic pipe flow effects. The appropriateness of the simulation tool is illustrated in a simplified case study.

2. Case Study Set-Up and Test Cases

The case study set-up presented in the paper is outlined in *Fig. 2*. This setup aims to represent the gas dynamic pipe flow and control problem described in the introductory section. The set-up consists of a pressure source supply, an EPM modulator unit, an actuator chamber of fixed volume, and a pipe connecting the modulator and the actuator chamber. The pressure supply is set to an absolute pressure of 11 bar; all other components are filled in the initial state to the absolute atmospheric pressure of 1 bar. The initial temperature in the pneumatic components as well as the ambient temperature is 293 K. Pipe length and diameter are 3 m and 10 mm, respectively. The pipe wall roughness is as usual for a flexible pneumatic pipe. The modulator internal chamber (CH) and actuator chamber volumes are 10 cm^3 and 1000 cm^3, respectively. The other parameters are based on [8].

The test cases presented herein for representation of fluid dynamic behavior of the system are as follows:

- *Test case 1:* loading the actuator chamber using a step-like LOAD solenoid command, then exhausting the actuator chamber using a step-like EXHAUST solenoid command,
- *Test case 2:* loading the actuator chamber using a step-like LOAD solenoid command, then applying a short (10 ms) impulse-like EXHAUST command (interrupted before the full exhaust of the actuator chamber),
- *Test case 3:* applying a short (10 ms) impulse-like LOAD command on the fully deflated actuator chamber.

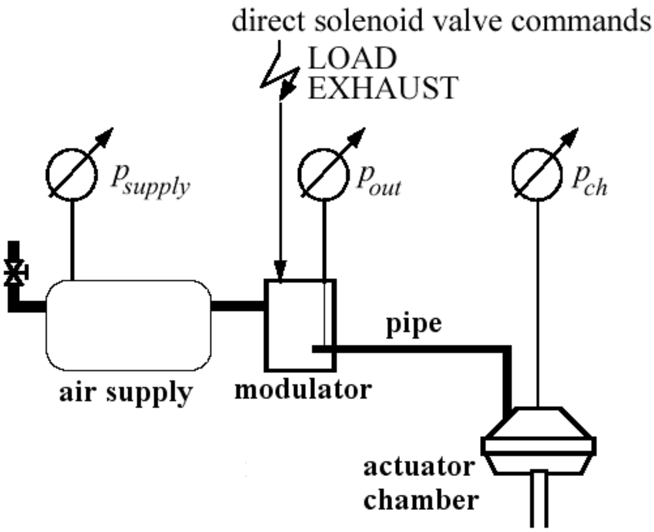

Fig. 2. Scheme of case study set-up

The pressure ratio (ratio between the pressures downstream and upstream of the pipe) takes instantaneously subcritical values for each test case. Therefore, the temporary development of sonic pipe flow is anticipated for the tests, giving an opportunity for comparison of pipe models based on different assumptions on compressibility.

3. Simulation Tool and Modelling

The study of dynamic behavior of the system outlined as above has been carried out using AMESim (Advanced Modeling Environment for Simulations of engineering systems) version 3.01 at the Department of Fluid Mechanics, Budapest University of Technology and Economics. Including a number of ready-made submodel elements structured in libraries, this simulation environment makes possible a convenient and effective modification, extension, and improvement of the case study simulation. Among several application fields, this software proved its appropriateness in simulation of systems related to automotive industry [9], [10], [11].

Fig. 3 represents the AMESim model of the test case presented herein, in a topology similar to that of *Fig. 2*. Most sub-element models have been built up using the commercially available AMESim sub-models taken from the mechanical, pneumatic, control, and hydraulic model libraries.

Given that the cited version of the AMESim software does not contain a realistic solenoid valve model, it had to be developed by the authors [2], [8]. For

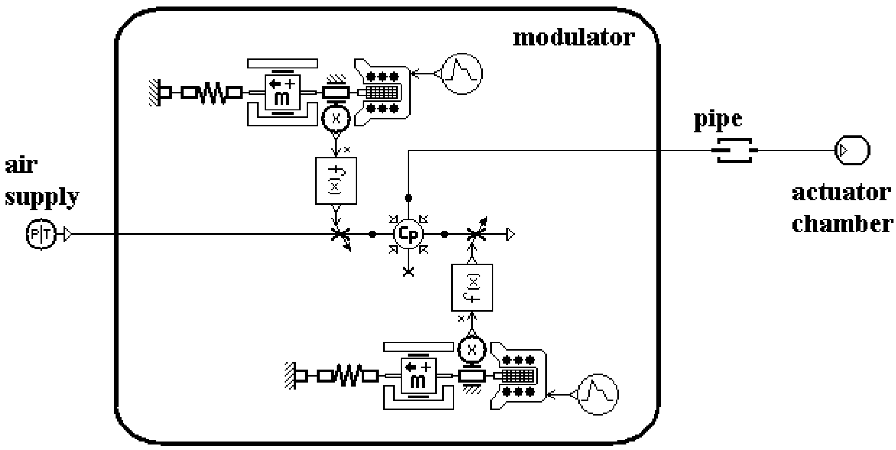

Fig. 3. AMESim model of case study set-up

pneumatic pipes there are several different models included in AMESim, but their documentation explicitly states that they should not be used when the gas velocity is high, i.e. $M > 0.3$ [12]. Given that a pneumatic pipe performs the physically possible entire Mach number range (up to sound speed) above the critical pressure ratio, a novel AMESim gas dynamic pipe model had also to be developed by the authors. The self-developed pipe and solenoid models were irreplaceable in the case study presented herein. Due to their special characteristics, the self-developed AMESim models deserve a more detailed description in the next chapter.

4. Self-Developed AMESim Models

Solenoid Valve Model

Solenoid valves are applied in fast-response pneumatic systems as control valves providing e.g. pressure signal for relay valves. Such miniature valves must provide rapid, pulsed fluid transmission between enclosures of relative pressures in the order of magnitudes of 10 bar and 0 bar within a time period in the order of magnitude of 0.01 s. In absence of solenoid excitation, the valve body is kept at its closed end-position by the return spring. The solenoid is energised by DC voltage. The resultant magnetic force displaces the valve body against the return spring. As a consequence, a flow cross-section develops through the orifice.

Solenoid valve models considering magneto-dynamic and mechanical effects are not available in the 3.01 version of the AMESim environment. The fully self-developed solenoid valve model [2], [8] is capable for realistic consideration of valve body position-dependent inductance, solenoid current, and magnetic force, on the basis of a detailed and accurate modeling of the magnetic circuit.

The complex system of a solenoid valve can be generally decomposed to interacting magneto-dynamic, mechanical, and fluid dynamic subsystems. Without going into detail of the self-developed complex mechatronic solenoid valve model, authors refer to [2] and [8] where the coupled electro-dynamic and mechanical subsystem models and the fluid mechanical aspects of valve operation are presented and verified by experiments.

The following section gives a summary on solenoid valve modelling. The magneto-dynamic subsystem comprises the following elements:

- The solenoid, acting as a magnetic exciter and also representing ohmic resistance.
- The frame, the jacket, the clearance between the valve body and the jacket internal bore, and the valve body. These elements represent together a magnetic circuit with magnetic resistance depending on valve body position. The magnetic field line loops are closed through the frame, the jacket, the air clearance, and the valve body.

The input variables of the magneto-dynamic subsystem are the excitation voltage and the valve body position. The output variables are the magnetic force acting on the valve body and the solenoid current. The magnetic force model is based on the achievement of magnetic energy minimum. This model is capable for resolution of valve body position-dependent magnetic resistance. It considers unsteady electro-dynamic effects in an accurate manner.

The mechanical subsystem comprises the valve body representing a mass, and the return spring. The equation of motion of valve body expresses that the temporal derivative of valve body linear momentum must be equal to the forces acting on the valve body. At the present state of investigation, such forces are: gravity force, magnetic force, return spring force, and forces of collision of the valve body at the end-positions. Resolving the effect of flexible seal and contact surfaces at the end of the valve body, flexible collision of the valve body is suitably modelled. With numerical treatment of the equation of valve body motion, the valve body position is computed.

The valve body position controls the cross-section of the throughflow orifice forming the fluid mechanical subsystem. Fluid mechanical forces acting on the valve body, including jet forces, pressure forces and viscous drag forces, are neglected for the present case study focusing on gas dynamic pipe behavior.

Gas Dynamic Pipe Model

This model is capable for resolution of pipe flow in the entire physically possible Mach number range, including wave phenomena such as shock waves [6] and pipe oscillations between the connected chambers. Heat transfer through the pipe wall and wall deformations are neglected. Pipe friction is based on wall roughness and the local Reynolds number.

The descriptive equations for the pipe model are as follows:
Ideal gas law:

$$\frac{p}{\rho} = R \cdot T \qquad \left(\kappa = \frac{c_p}{c_v} = \text{const.}\right). \tag{1}$$

Continuity:

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho \cdot v)}{\partial x} = 0. \tag{2}$$

Equation of motion:

$$\frac{\partial(\rho \cdot v)}{\partial t} + \frac{\partial(\rho \cdot v^2)}{\partial x} + \frac{\partial p}{\partial x} = -\frac{\rho \cdot \lambda}{2 \cdot d} \cdot v \cdot |v|. \tag{3}$$

Energy equation:

$$\frac{\partial(\rho \cdot e)}{\partial t} + \frac{\partial(\rho \cdot v \cdot h)}{\partial x} = 0, \tag{4}$$

$$e = c_v \cdot T + \frac{v^2}{2}, \tag{5}$$

$$h = c_p \cdot T + \frac{v^2}{2}. \tag{6}$$

Two-step Lax-Wendroff scheme [6] [13] was used to solve the equations. The maximum time step needed for stable run was derived from the Courrant–Friedrich–Lévy criteria [6]. Discontinuities caused by shock waves were treated with the method of artificial viscosity [13]. In this method the local pressure has been increased according to the shock wave pressure ratio. This flattens the numerical oscillations at the cost of slightly decreased accuracy.

5. Test Results

As discussed above, in realistic systems the pressure sensor PS is integrated into the modulator chamber (CH in *Fig. 1*). Thus the modulator chamber pressure is considered to be the most representative measure of this system, also reflecting pipe flow phenomena in a lifelike manner. Therefore, the following diagrams presented for the three test cases contain the time function of modulator chamber (CH) absolute pressure (p_out, since appearing also on the output port of the EPM). For test case 1, the time function of the actuator chamber pressure (p_ch) is also presented in order to illustrate the difference between the development of actuator and modulator chamber pressures. This calls attention again to control aspects given that the aim of the EPM is to ensure a suitably controlled pressure in the actuator chamber.

As a basis of comparison, the simulation tests have also been carried out with use of the most sophisticated wave equation pipe model available in the standard AMESim 3.01 pneumatic library. This distributed parameter pipe model is recommended for the most demanding use resolving wave effects. However, its application is recommended to be restricted to low-velocity pipe flow of $M < 0.3$ (condition of incompressibility). *Figs.* 4, 6 and 7 present the comparative diagrams of p_out functions obtained with use of the self-developed gas dynamic pipe model as well as the most sophisticated standard AMESim pipe model, with subscripts 'gd' and 'wave', respectively.

At the bottom of each graph the solenoid valve commands are indicated (dark bars: excitation of the LOAD valve, grey bars: excitation of the EXHAUST valve).

Fig. 4 presents the simulation results for test case 1. The differences between the standard and self-developed pipe modelling are minor, although it can be seen that these differences concentrate around the wave and throttled expansion effects (at the beginning of the load phase). It is clearly visible in *Fig.* 5 that the larger actuator chamber filters out almost totally the differences caused by the different pipe flow calculation methods. It can be concluded that for such a simple operational state, for which no controlling intervention occurs in the pipe flow, also the standard low flow velocity AMESim pipe model supplies satisfactory results.

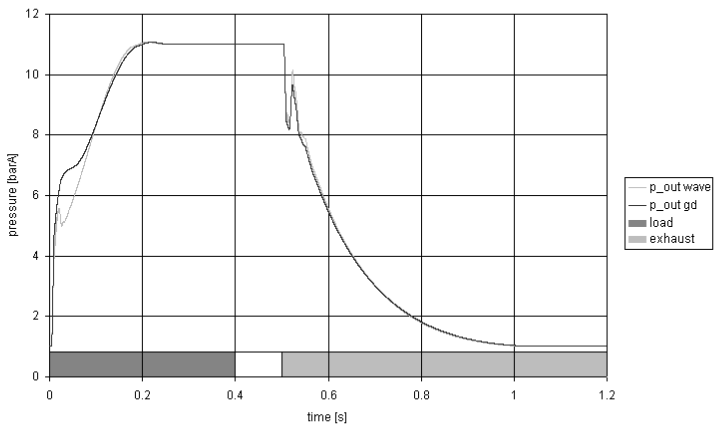

Fig. 4. Modulator chamber pressure, Test case 1

Test cases 2 and 3 are more realistic from the viewpoint of pneumatic control. *Fig.* 6 shows the simulation results for test case 2. In this case the advantages of the self-developed gas dynamic pipe model are obvious. It resolves the same wave effect as the built-in model, but without going into heavy oscillations.

For test case 3 (*Fig.* 7) the gas dynamic pipe submodel proves its advantage over the standard model again. Besides, the standard model does not even finish this test run, as because of the heavy oscillations the absolute pressure becomes negative, which is, of course, physically impossible. The gas dynamic model successfully

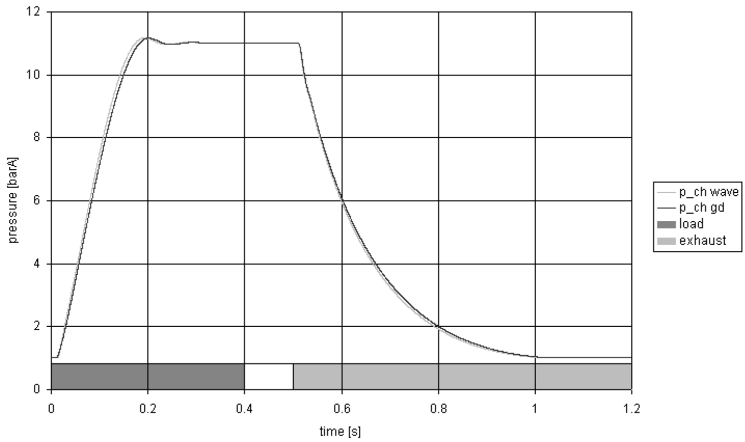

Fig. 5. Actuator chamber pressure, Test case 1

finishes this test as well, although some oscillations can be observed in the first quarter of the graph.

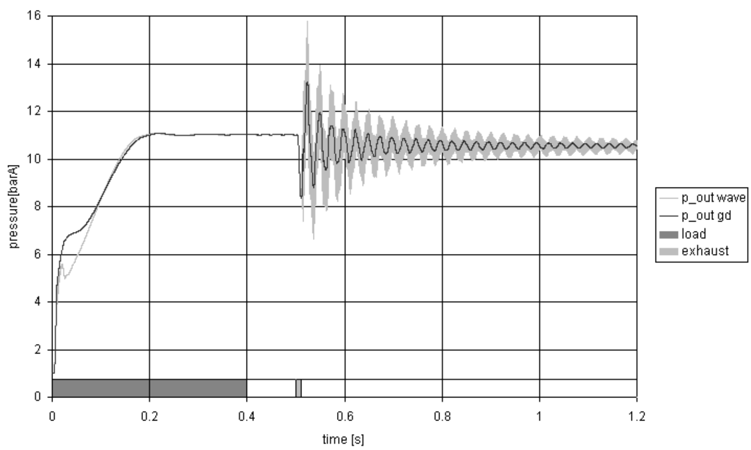

Fig. 6. Modulator chamber pressure, Test case 2

6. Conclusions

The simulation studies reveal that the pneumatic pipe performs wave effects and flow fluctuations between the modulator chamber and the actuator chamber. The gas dynamic pipe model resolves such phenomena in a realistic manner. The results also

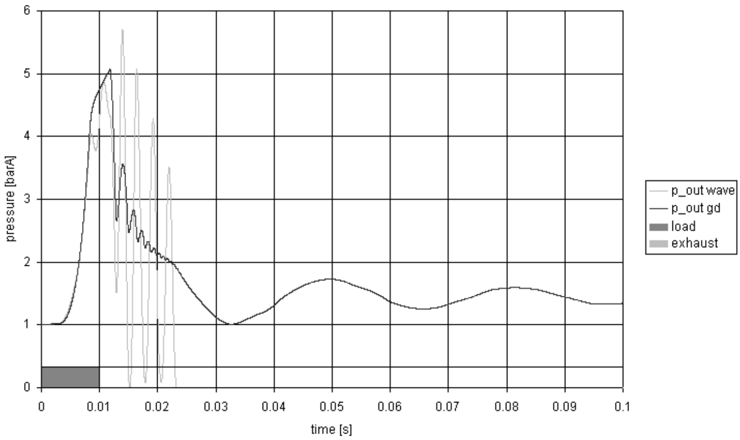

Fig. 7. Modulator chamber pressure, Test case 3

demonstrate that the standard AMESim pipe model performs quite well when the pipe flow velocity does not change drastically, as in test case 1. However, strong numerical oscillations can be observed for test case 2, where the flow velocity change is sudden and drastic. In that case the gas dynamic model performs much better since it resolves the same effect without heavy oscillations. The wave model performed even worse in the pulsed flow of test case 3, as it could not finish the simulation. Results show that because of the oscillation the modulator chamber pressure ran into a physically impossible region with use of the standard AMESim pipe model (negative absolute pressure). The gas dynamic model performs better again, showing similar effects as in case 2, although some numerical oscillations can be observed here as well.

The simulations forecast unsteady pipe flow effects that may lead to a harmful resonance in the pressure control loop. Such resonance manifests itself in oscillation of pressure around the value prescribed by the control, also leading to a considerable reduction of solenoid valve life cycle.

It is concluded that an advanced gas dynamic pipe model is irreplaceable if a reliable pneumatic control is to be designed by numerical simulation for systems comprising pneumatic pipes.

In the near future, experiments will be carried out for verification of simulation results obtained using the pneumatic pipe models presented in the paper.

Nomenclature

c_p constant-pressure specific heat
c_v constant-volume specific heat

d	diameter of pipe
e	total energy
h	total enthalpy
M	Mach number of pipe flow
p	gas pressure
p_ch	actuator chamber pressure
p_out	modulator chamber pressure
R	perfect gas constant
t	time
T	gas temperature
v	gas velocity
λ	pipe friction coefficient
x	longitudinal pipe co-ordinate
κ	specific heat ratio
ρ	gas density

Subscripts

wave	most complex standard AMESim pneumatic pipe model
gd	gas dynamic pipe model developed by the authors

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