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AN ANALYSIS OF TRANSIENT PROCESSES IN PNEUMATIC BRAKE SYSTEM WITH AUTOMATIC REGULATOR OF BRAKE FORCES OF AUTOMOTIVE VEHICLES[1]

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Summary. The method of computer identification of dynamic conductivity of automatic regulator of brake force of automotive vehicle is presented in the paper. The obtained results were used in calculations of brake circuit dynamic of a truck with automatic regulator of brake force. Diagrams of pressure runs in different points of brake circuit with different positions of control lever are presented.

Key words: Pneumatic brake systems, regulator of brake forces of vehicles

INTRODUCTION

In order to obtain a high efficiency of braking and retain the stability of automotive vehicle movements the automatic regulators of brake force are used. An example of such a regulator which is used in the circuit of truck back wheels is 6120 regulator produced in POLMO Prashka [Katalog... 1996] (Fig.1).

It is necessary to know both the pneumatic conductivity of a regulator and its input-output flow characteristic for the synthesis and analysis of pneumatic brake systems.

The aim of the work is the experimental determination of pneumatic conductivity of automated regulator of brake forces and application of the obtained results in a mathematical model of pneumatic brake circuit.

RESEARCH METHOD

The elements used in brake systems, such as: pipes, valves, regulators, filters, connecting pieces are characterised by their internal volume and conductivity. The conduc-

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tivity may be a constant or variable value. In the case of the presented regulator the conductivity is variable in regard to the mutual displacement of movable elements of the regulator in its working mode, consequently passage 3 changes (Fig. 1a). The size of the passage depends on the location of control lever 8 and on pressure value in switchers 1 and 2.

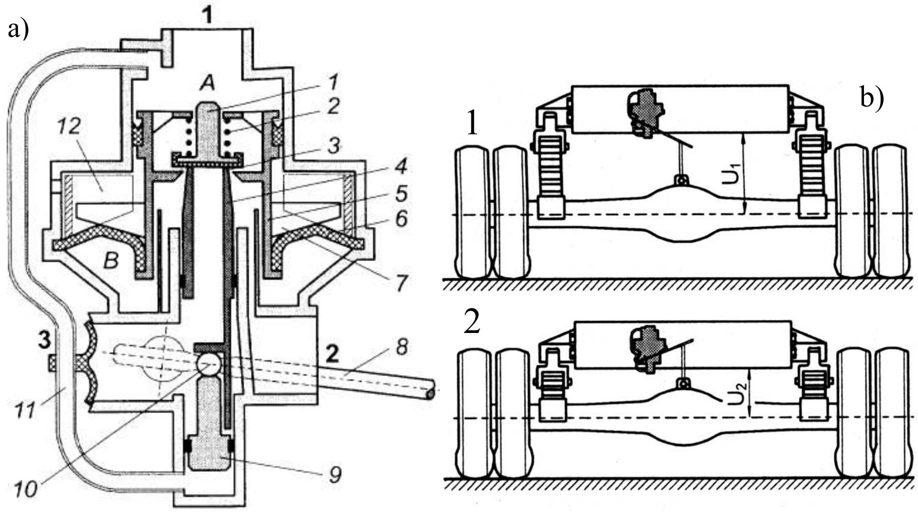

Fig.1. Automatic regulator of brake force [Łomako *et al.* 2002]:

a) The construction: 1 – valve; 2 – spring; 3 – passage; 4 – pusher; 5 – piston; 6 – membrane; 7 – ribs of piston 5; 8 – lever; 9 – piston; 10 – ball pin; 11 – tube; 12 – insert with inner ribs b) Deflexion of suspension for: 1 – unloaded truck; 2 –loaded truck

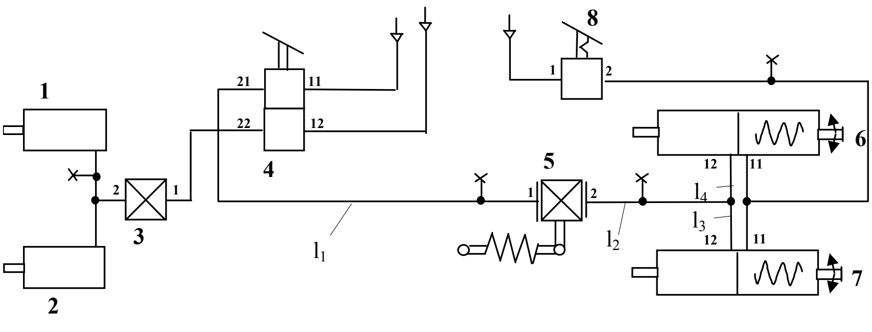

Fig. 2. Scheme of the pneumatic brake system of a truck; 1, 2 – cylinders of front axle; 3 – anti-block system of front axle; 4 – main valve of brakes; 5 – automated regulator of brake force; 6,7 – cylinders of back axle; 8 – manual brake valve

For the regulator exploration the method developed at the Department of Automotive Vehicle of Bialystok University of Technology [Kamiński 2003, Miatljuk *et al.* 2003a, b] is used. The method utilises relation, which describes the mass intensity of air flow through the nozzle [Miatluk 1980]:

$$Q = (\mu A) \cdot v_m \frac{p_2}{R \cdot T} \varphi_{max}(\sigma_{kr}) \cdot \varphi(\sigma) \tag{1}$$

where:

(μA) – conductivity (effective square of flow), m^2;
T – air temperature in front of the resistor, K,
σ – the relation of pressures p_3 behind and p_2 in front of the resistor (Fig. 2),
$\varphi(\sigma)$ – non-dimensional flow function,
R – gas constant, for the air R = 287.14, J/(kg·K),
v_m – the velocity of dispersal of sound in non-movable gas $v_m = \sqrt{\kappa \cdot R \cdot T}$,
κ – adiabatic coefficient,
$\varphi_{max}(\sigma_{kr})$ = 0,578 – maximum value of Saint Venant and Wantsel theoretical function.

In order to simplify the stand, the concept of indirect measurement on the basis of pressure run in the filled receiver and in the tested element is used in the method. The stand scheme is presented in Figure 3.

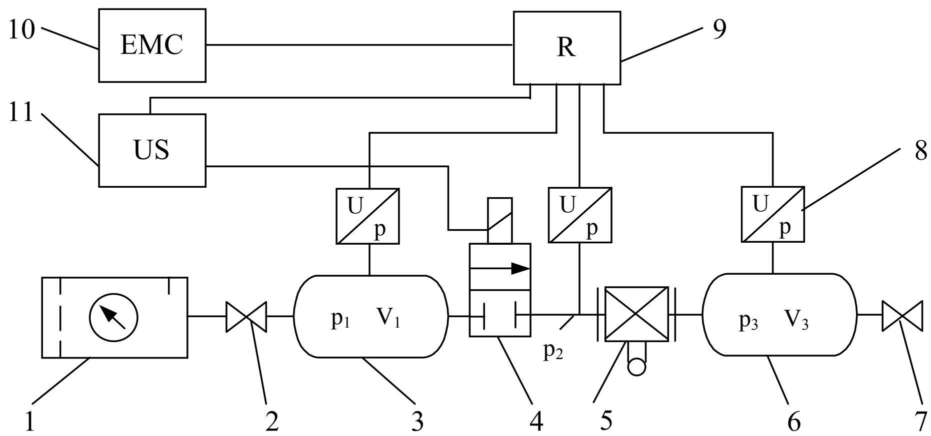

Fig. 3. Scheme of the exploratory stand: 1 – air preparation station; 2, 7 – separating ball valve QH/4, 3 – air receiver (with delay); 4 – electromagnetic valve MV1H-2-11/2; 5 – explored (tested) element, 6 – air receiver (filling), 8 – pressure converter P15RV1/10B, 9 – recorder MC201A, 10 – computer, 11 – control unit

State equations of gas in receiver 6 with constant volume V_1 during the process of its adiabatic filling have the following form:

$$Q = \frac{dm}{dt} = \frac{V_3}{\kappa RT} \cdot \frac{dp_3}{dt} \tag{2}$$

where:

p_3 – pressure in receiver 6.

Using the equations (1) and (2) we obtain the differential equation:

$$\frac{dp_3}{dt} = \frac{\kappa RT}{V_3}\left((\mu A) \cdot v_m \cdot \frac{p_2}{RT} \cdot \varphi_{\max}(\sigma_{kr}) \cdot \varphi\left(\frac{p_3}{p_2}\right)\right) \tag{3}$$

This equation was used in the works [Miatljuk *et al.* 2003a, b] for the computer identification of μA value and parameters of flow function $\varphi(\sigma)$ on the basis of the experimentally detected run of pressures p_{2d} and p_{3d} in the process of air flow from receiver 3 to receiver 6. In the case of the brake force regulator it is also necessary to identify the function which describes the change of the resistance μA (σ,α). The problem can be simplified to a large extent by using theoretically or experimentally detected flow function [Miatljuk *et al.* 2003a, b].

Taking the flow function proposed by Miatluk and Avtushko [1980] in form of:

$$\varphi(\sigma) = 1{,}13\frac{1-\sigma}{1{,}13-\sigma}$$

the equation (3) for theoretically defined pressure p_{3t} run in the receiver V_3 and for experimentally detected pressure p_{2d} in front of the explored regulator turns into the following form:

$$\frac{dp_{3t}}{dt} = \frac{\kappa RT}{V_3}\left(1.13\frac{1-\sigma}{1.13-\sigma}\left(\mu A(\sigma,\alpha)\right) \cdot v_m \cdot \frac{p_{2d}}{RT} \cdot \varphi_{max}(\sigma_{kr})\right) \tag{4}$$

where:

$$\sigma = \frac{p_{3t}}{p_{2d}}$$

THE RESEARCH PROCESS AND RESULTS

The exploration of brake force regulator has been carried out for different angle positions of lever 8 (Fig. 1) ($\alpha = 0$ – horizontal position of the lever). In each position the measurements have been repeated 10 times while registering pressures p_{1d} , p_{2d} and p_{3d} by standard programs of MC201A recorder. The resistance function of $\mu A(\sigma,\alpha)$ run is accepted in the form of:

$$\mu A(\sigma,\alpha) = \begin{cases} \mu A_p\left(1-(\sigma/\sigma_z)^B\right) & \text{dla} \quad 0 \le \sigma < \sigma_z \\ 0 & \text{dla} \quad \sigma_z \le \sigma \end{cases} \tag{5}$$

where:

σ_z – the relation of the pressures behind and in front of regulator at which the flow is closed,

μA_p – conductivity,

B – power index (exponent).

μA_p, σ_z and B parameters were defined by numeric non-gradient method of searching of Hook-Jeeves lines by minimising the expression:

$$\sum_{i=1}^{m} \left(p_{3di} - p_{3ti}\right)^2$$

till reaching the defined computation accuracy. On each iteration step the differential equation (6) was solved by Fehlberg method. For the identification the program worked out by the authors and numeric procedures presented in work [Baron *et al.* 1995] were used.

Exemplary registered pressure runs and theoretical run are presented in Figure 4.

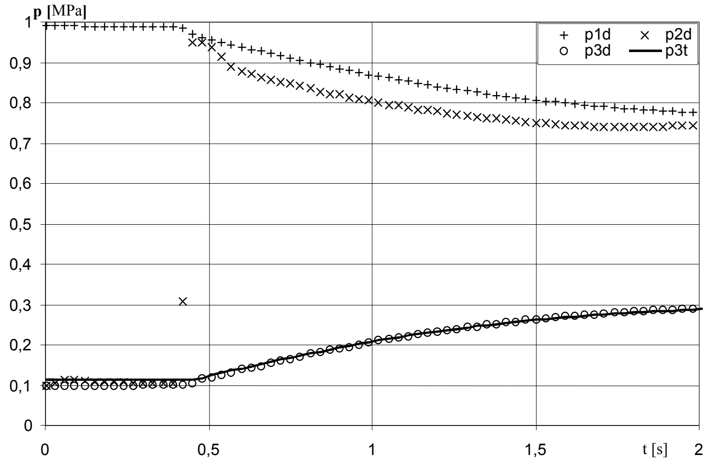

Fig. 4. Registered experimental runs of pressures p_{1d} , p_{2d} , p_{3d} and theoretical run p_{3t} for the lever position α = 20° (presented the each 10th point)

The obtained values for each lever angle position of the regulator were averaged and presented in Table 1.

Table 1. Average values of identified parameters of conductivity function

α [°]	μA_p [x10^{-5}m^2]	σ_z	B
-23	2.7907	0.3506	5.0059
-20	3.0243	0.3583	3.6672
-10	3.0747	0.3989	4.4332
0	3.7268	0.5362	4.4399
10	4.5222	0.7647	4.3672
20	5.2183	1.2369	3.7332
30	5.1037	1.4056	4.2700
37	5.1667	1.9582	3.4695

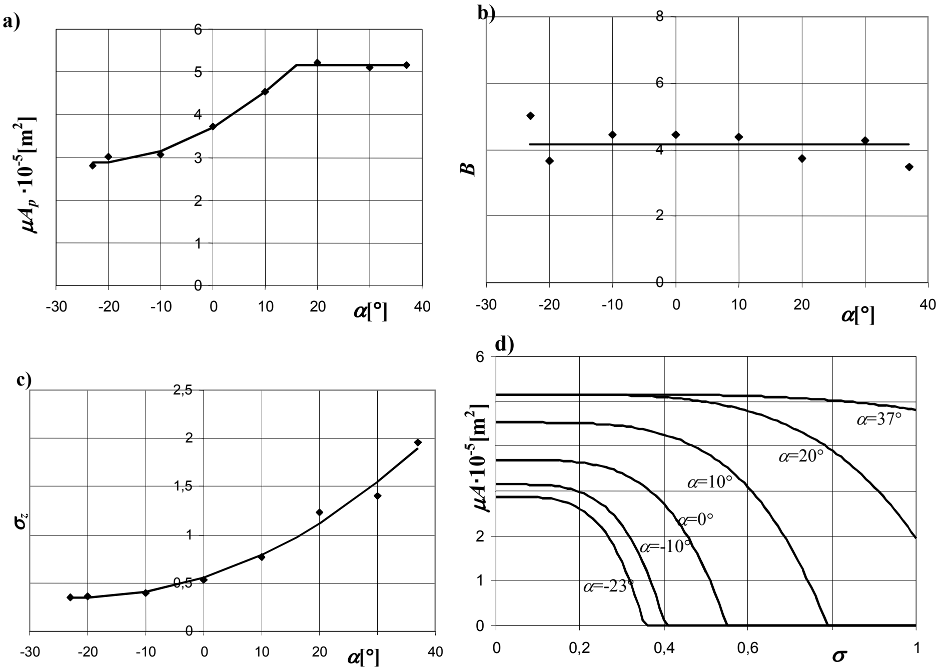

Fig. 5. The results of the exploration of automatic regulator of brake force: a) the diagram of parameter μA_p change in the function of lever position α; b) the diagram of parameter σ_z change in the function of lever position α; c) the diagram of parameter B change in the function of lever position α; d) run of conductivity changes in the function σ for the different lever position α

Next, the dependence of identified parameters from the rotation angle of regulator lever was defined:

$$\mu A_p(\alpha) = \begin{cases} 1.448 \cdot 10^{-3}\alpha^2 + 6.893 \cdot 10^{-2}\alpha + 3.694 & \text{for} \quad -23° \le \alpha < \alpha_n \\ 5.163 & \text{for} \quad \alpha_n \le \alpha \le 37° \quad R^2 = 0.995 \end{cases}$$
$$\sigma_z(\alpha) = 4.659 \cdot 10^{-4}\alpha^2 + 1.925 \cdot 10^{-2}\alpha + 0.5491 \quad \text{for} \quad -23° \le \alpha < 37° \quad R^2 = 0.985 \tag{6}$$
$$B(\alpha) = 4.173 \quad \text{for} \quad -23° \le \alpha < 37°$$

where:

$\alpha_n = 15{,}96°$.

The angle α_n, in which the change of character of conductivity run appears, can be interpreted as an angle of insensitivity of regulator. The dependence of each of the parameters which define the change of conductivity function of regulator and the function run are presented in Figure 5

MODELLING OF TRANSIENT PROCESSES IN THE BRAKING SYSTEM

The obtained resistance function run allows for a modelling of transient processes in the brake circuit of back axle of automotive vehicle (Fig. 2). Computable scheme of the circuit is presented in Fig. 6.

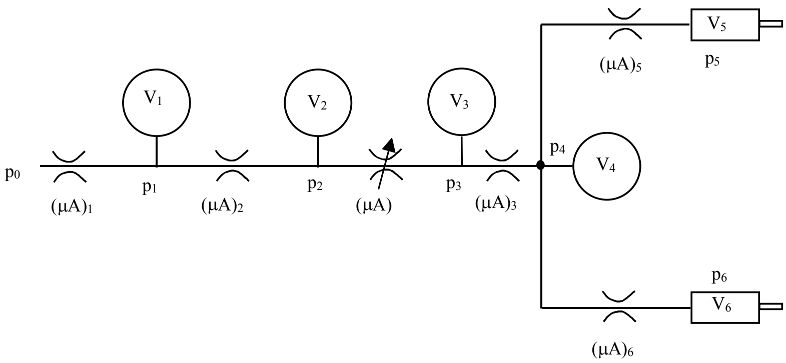

Fig. 6. Computable scheme of the brake circuit of vehicle back axle.

On the basis of nodes method [Miatljuk. 1980] and the equations of mass intensity of flow through local resistors, a mathematical model of transient process was constructed in the form of the following equations (7):

$$\frac{V_1}{\kappa RT}\frac{\mathrm{d}p_1}{\mathrm{d}t} = (\mu A)_1 v_m \frac{p_0}{RT} 0.654 \frac{p_0 - p_1}{1.13 p_0 - p_1} - (\mu A)_2 v_m \frac{p_1}{RT} 0.654 \frac{p_1 - p_2}{1.13 p_1 - p_2}$$

$$\frac{V_2}{\kappa RT}\frac{\mathrm{d}p_2}{\mathrm{d}t} = (\mu A)_2 v_m \frac{p_1}{RT} 0.654 \frac{p_1 - p_2}{1.13 p_1 - p_2} - \mu A\left(p_3 / p_2, \alpha\right) v_m \frac{p_2}{RT} 0.654 \frac{p_2 - p_3}{1.13 p_2 - p_3}$$

$$\frac{V_3}{\kappa RT}\frac{\mathrm{d}p_3}{\mathrm{d}t} = \mu A\left(p_3 / p_2, \alpha\right) v_m \frac{p_2}{RT} 0.654 \frac{p_2 - p_3}{1.13 p_2 - p_3} - (\mu A)_3 v_m \frac{p_3}{RT} 0.654 \frac{p_3 - p_4}{1.13 p_3 - p_4}$$

$$\frac{V_4}{\kappa RT}\frac{\mathrm{d}p_4}{\mathrm{d}t} = (\mu A)_3 v_m \frac{p_3}{RT} 0.654 \frac{p_3 - p_4}{1.13 p_3 - p_4} - (\mu A)_5 v_m \frac{p_4}{RT} 0.654 \frac{p_4 - p_5}{1.13 p_4 - p_5} + \\ - (\mu A)_6 v_m \frac{p_4}{RT} 0.654 \frac{p_4 - p_6}{1.13 p_4 - p_6}$$

$$\frac{V_5}{\kappa RT}\frac{\mathrm{d}p_5}{\mathrm{d}t} = (\mu A)_5 v_m \frac{p_4}{RT} 0.654 \frac{p_4 - p_5}{1.13 p_4 - p_5}$$

(7)

$$\frac{V_6}{\kappa RT}\frac{\mathrm{d}p_6}{\mathrm{d}t} = (\mu A)_6 v_m \frac{p_4}{RT} 0.654 \frac{p_4 - p_6}{1.13 p_4 - p_6}$$

where: p_i – pressures in each node, Pa,

V_i – volumes of system elements, m^3,

$V_1 = V_2 = A_p l_1/2$; $V_3 = A_p l_2/2$; $V_4 = A_p l_2/2 + A_p l_3/2 + A_p l_4/2$;

$V_5 = A_p l_4/2 + V_s$; $V_6 = A_p l_5/2 + V_s$;

V_s – volume of cylinder for 2/3 of the maximum stroke, m^3,

l_i – length of pipes sections of brake system, m (Fig.1c),

A_p – field of pipe cross-section, m^2, $A_p = \pi d_p^2/4$,

d_p – diameter of cross-section, m,

$(\mu A)_i$ – conductivity of element, m^2, in the case of cross-section,

$(\mu A)_i = \mu_i A_p / \sqrt{k(l_i)}$,

$k(l_i)$ – number of one-meter sections of l_i length pipe in marked pipe part of computable scheme,

μ_i – coefficient of expenditures of one-meter pipe section, $\mu_i = 0.2$.

While composing the equation it was accepted, that:

- air temperature is constant and equal to $T = 273$K,
- elements connections are ideally hermetic and pipes are ideally inflexible;
- the commitments volumes of pneumatic cylinders are constant and correspond to 2/3 of maximal stroke,
- the dependence of pressure $p_0(t)$ behind the brake valve is equal to:

$$p_0(t) = \frac{p_m}{t_m} t + p_a \qquad \text{dla } 0 \le t < t_m$$

$$p_0(t) = p_m \qquad \text{dla } t_m \le t$$

where:

p_m – the maximum pressure in brake circuit, $p_m = 7 \cdot 10^5$ Pa,

p_a – atmosphere pressure, $p_a = 1 \cdot 10^5$ Pa,

t_m – time of pressure growing on the output of brake valve, $t_m = 0.2$ s.

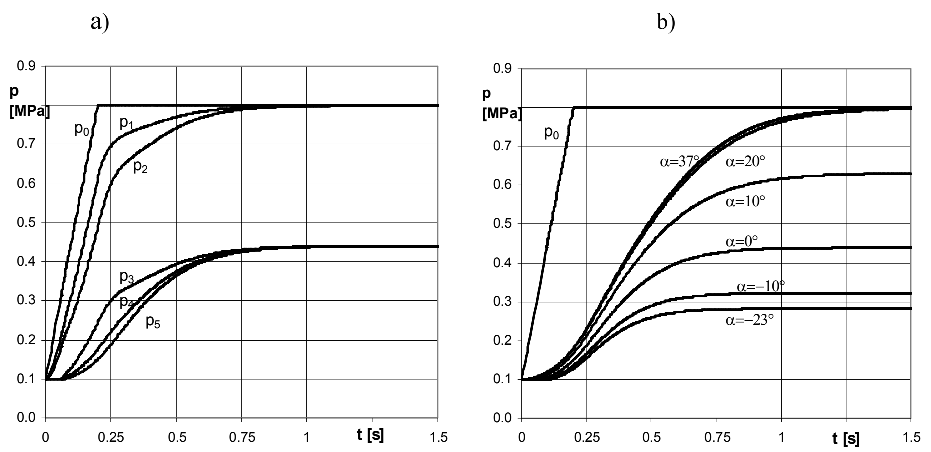

Fig.7. The results of brake circuit of back axle simulation: pressure runs in different points of brake system (α=0°); b) pressure runs in cylinders of back axle for different angle position of regulator lever

The system of differential equations (7) was solved out in Matlab-Simulink system. The following parameters of brake system were accepted: l_1 = 4m, l_2 = 1 m, $l_3 = l_4$ = 2 m, Vs = 0.72 10^{-3} m^3, d_p = 0.013 m. The exemplary pressure runs in each of the circuit nodes with the angle positions of regulator lever being equal to α = 0° are presented in Fig. 7a.

Pressure runs in brake cylinders for other angle positions are presented in Fig. 7b. It is possible to state on the basis of the obtained runs that the pressure value in brake cylinders achieves 75% of its asymptotic value in the time shorter than 0.6 s.

CONCLUSIONS

The exploration method presented in the paper permits to identify the dynamic conductivity of the automatic regulator of brake forces depending on the value of air pressure in the system, and also on the angle position of the control lever. The results obtained allow for the execution of the dynamics of pneumatic braking systems with the regulator of brake force, and, in particular, their optimisation in order to enhance braking efficiency, provide the movement stability and reduce the time of pneumatic cylinders reaction. The presented method can be also applied in the exploration of automotive vehicle regulators and valves of other types.

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