MODELLING OF THE ENERGY SUPPLY EQUIPMENT OF THE AIR BRAKING SYSTEM OF A FARM TRACTOR

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Summary

Farm tractors are provided with pneumatic systems to control and operate air braking systems as well as pneumatic suspension systems, the latter being increasingly often applied to high-capacity trailers. The excessive consumption of compressed air may adversely affect the braking effectiveness of a tractor-trailer unit; therefore, the impact of performance of the air supply equipment of the tractor on the transient processes taking place in the air braking system of the trailer should be taken into consideration as early as at the design stage.

A mathematical model of the air supply equipment, consisting of functional and structural models of individual equipment components such as compressor, pressure regulator, and compressed air reservoir, has been presented in this paper. An example has been included, where the computer model of the supply system, prepared in the Matlab-Simulink program, was used to assess the correctness of compressor selection to the pneumatic system of the Pronar 5110 tractor. The adequacy of the implemented computer model of the air supply equipment, experimentally confirmed, was evaluated with statistical methods, with the use of the Kolmogorov-Smirnov test.

The computer model developed may be used as a tool to assess the correctness of selection of parameters of the air supply equipment within the designing process and as a subsystem in order to evaluate, with the use of simulation methods, the transient processes taking place in the air braking systems of agricultural vehicles.

Keywords: farm tractor, pneumatics, braking system, air supply equipment, modelling

1. Introduction

In most farm tractors, hydraulic systems are used to actuate their braking mechanisms; sometimes, mechanical or pneumatic systems are used as an alternative for this purpose. The pneumatic systems installed in tractors are chiefly used to supply energy to, and to control, air braking systems of trailers and farm machinery coupled with the tractors.

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A typical pneumatic system of a farm tractor consists of two major parts: air supply system and control system. The role of the control system is to provide follow-up control of a single-line or two-line braking system of the trailer in a way that should enable synchronous braking of both the vehicles. The functions of the air supply system are to compress and purify air and to keep adequate air pressure in the tractor and trailer reservoirs so that the required trailer braking effectiveness is ensured. The pneumatic system is also used to supply compressed air to suspension systems of high-capacity agricultural trailers. In such conditions of operation of the pneumatic system, considerable quantities of compressed air are consumed, which may adversely affect the effectiveness of braking of the tractor-trailer unit. To select appropriate system design parameters and to analyse the operation of the pneumatic system of the tractor in transient states with the use of simulation methods, the preparation of a mathematical model of individual system components, including the air supply equipment, was found necessary.

2. Mathematical model

A simplified schematic diagram of a two-line pneumatic system of a farm tractor has been presented in Fig. 1. The air supply equipment includes compressor 1, pressure regulator 2, compressed air reservoir 4, and supply line 5 with a coupling to supply compressed air to the braking system of the trailer. Compressed air is also fed to a control system, which includes valve 6 to control trailer brakes, and control line 7. The air supply equipment may also include other components, not shown in the drawing, such as filter, oil separator, safety valve, or pressure reducing valve indispensable in high-pressure systems.

When the maximum value of the regulated pressure p_{max} in the reservoir is achieved, abrupt operation of the pressure regulator takes place, in result of which the delivery port of the compressor is connected to a vent. When this connection is open, the compressor operates without load and it is gradually cooled. When the pressure in the reservoir drops to the lowest acceptable level p_{min} the regulator abruptly reconnects the compressor

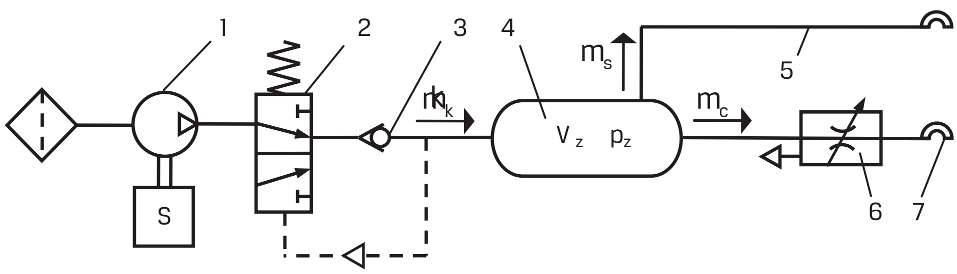

Fig. 1. Computational diagram of the pneumatic system of a farm tractor: 1 – compressor; 2 – pressure regulator; 3 – check valve; 4 – air reservoir; 5 – supply line; 6 – trailer brake control valve; 7 – control line.

with the reservoir. This type of operation of the regulator can be described by a static characteristic curve representing the operation of a bistable relay with a hysteresis loop of width equal to $p_{max} - p_{min}$.

Assuming, to simplify, that the pressure in the compression pressure of the compressor is equal to that in the reservoir, i.e. p_z (short length and small cubic capacity of the parts connecting the compressor with the reservoir), we may describe the volume flow of the air delivered by the compressor as follows [4]:

$$Q_k = \begin{cases} \left.\begin{matrix} \dot{V}_k(n_k, p_z) & \text{gdy } p_z < p_{\max} \\ 0 & \text{gdy } p_z \ge p_{\max} \end{matrix}\right\} \text{gdy } \dfrac{dp_z}{dt} \ge 0 \\ \left.\begin{matrix} 0 & \text{gdy } p_z \ge p_{\min} \\ \dot{V}_k(n_k, p_z) & \text{gdy } p_z < p_{\min} \end{matrix}\right\} \text{gdy } \dfrac{dp_z}{dt} < 0 \end{cases} \tag{1}$$

The volume flow $\dot{V}_k(n_k, p_z)$ depends on compressor shaft speed n_k and pressure p_z in the reservoir:

$$\dot{V}_k(n_k, p_z) = \eta_v \cdot V_s \cdot i_c \frac{n_k}{60} = \eta_v \frac{\pi \cdot D_c^2}{4 \cdot 60} \cdot S \cdot i_c \cdot n_k \quad [\text{m}^3/\text{s}], \tag{2}$$

where: η_v – volumetric efficiency; V_s – displacement volume [m³], D_c – cylinder diameter [m], S – piston stroke [m], i_c – number of cylinders.

The mass flow of the air delivered by the compressor may be calculated by multiplying the volume flow by air density in ambient conditions:

$$\dot{m}_k = \rho_a \cdot Q_k = \frac{p_a}{RT_a} Q_k \quad [\text{kg/s}], \tag{3}$$

where: p_a – atmospheric pressure [Pa], T_a – ambient temperature [K], R – gas constant [J/(kgK)].

The value of volumetric efficiency ηv may be estimated from a theoretical dependence [1], assuming the inlet pressure as being equal to the atmospheric pressure:

$$\eta_v = 1 - \frac{V_{sz}}{V_s}\left[\left(\frac{p_z}{p_a}\right)^{1/n} - 1\right], \tag{4}$$

where: V_{sz} – clearance volume [m³], n – polytropic exponent, n=1,25÷1,4.

The volumetric efficiency η_v may also be determined by the non-linear regression method, based on the compressor delivery performance curve. In general, this coefficient depends on compressor shaft speed n_k and delivery pressure (pressure p_z in the reservoir). According to author's research work, the volumetric efficiency may be described for most piston compressors by a non-linear regression equation in the following form:

$$\eta_v = A_1 + A_2 n_k + A_3 n_k^2 + A_4 p_z + A_5 p_z^2 , \tag{5}$$

where: n_k – compressor shaft speed [rev/min], p_z – delivery pressure [kPa], $A_1 \div A_5$ – coefficients of regression. For the FOS Polmo compressor model 601.23.931 installed in the Pronar 5110 tractor: A_1=0,80698; A_2=0,46902E-4; A_3=-1,47791E-8; A_4=-7,81944E-4; A_5=3,80523E-7 (R^2=99,37%).

The compressor shaft speed may be calculated from the tractor speed. Knowing the total gear ratio of the tractor power transmission system, we may calculate the engine speed and then the compressor shaft speed:

$$n_k(t) = \frac{30 \cdot i_c v}{\pi \cdot i_k r_k (1-s)} \ \text{[revs/min]}, \tag{6}$$

where: v – tractor speed [m/s], i_c – total gear ratio of the tractor power transmission system, i_k – gear ratio of the compressor drive system, r_k – kinematic radius of the tractor wheel [m], s – slip of the driving wheels.

During the braking process, the compressor shaft speed drops from the level corresponding to the engine speed at the beginning of the braking process to that at the engine idling speed:

$$n_k(t) = n_{kh} - a_1 \cdot t \ \text{ dla } n_k > n_{kj} , \tag{7}$$

where: n_{kh} – compressor shaft speed corresponding to the tractor speed at the beginning of the braking process [revs/min]; a_1 – ratio of the compressor shaft speed (engine speed) drop during the braking process, determined experimentally [revs/(min×s)]; n_{kj} – compressor shaft speed at the engine idling speed [revs/min].

The equation of balance of the mass flows in the reservoir has the form:

$$\frac{dm_z}{dt} = \dot{m}_k - \dot{m}_s - \dot{m}_c , \tag{8}$$

where: $\dot{m}_s$ – mass flow entering the supply line [kg/s], $\dot{m}_c$ – mass flow entering the control line [kg/s].

Based on the energy conservation law for an open system and the ideal gas equation of state, we obtain the following dependencies describing the changes of temperature and pressure of air in the reservoir [3]:

$$\frac{dp_z}{dt} = \frac{1}{V_z}\left[(\kappa - 1)\left(\dot{Q} + \dot{H}_k - \dot{H}_s - \dot{H}_c\right)\right]$$

$$\dot{Q} = \alpha_z A_z \left(T_w - T_z\right) \qquad \dot{H}_i = \dot{m}_i c_p T_i \tag{9}$$

$$\frac{dT_z}{dt} = \frac{T_z}{p_z V_z}\left[V_z \frac{dp_z}{dt} - RT_z\left(\dot{m}_k - \dot{m}_s - \dot{m}_c\right)\right] \tag{10}$$

where: H_k – enthalpy of the flow generated by the compressor [J], H_s – enthalpy of the flow entering the supply line [J], H_c – enthalpy of the flow entering the control line [J], Q – heat exchanged with the environment [J], α_z – heat transfer coefficient [W/m²K], T_z – temperature of air in the reservoir [K], A_z – heat transfer area, T_w – temperature of reservoir walls [K].

The temperature of the airflow generated by the compressor may be estimated from the polytropic equation:

$$T_k = T_a \left(\frac{p_z}{p_a}\right)^{\frac{n-1}{n}} \tag{11}$$

3. Example of a model application

The mathematic model of the air supply system was employed to build, in the Matlab-Simulink program, a computer model of the pneumatic system of the Pronar 5110 tractor. The brake control system of this tractor includes a hydraulically operated Haldex brake control valve model 329 020 201. The computer model presented in Fig. 2, in a version intended for the simulation of functioning of the supply system, was used to check the compressor delivery rate at filling a dummy reservoir of 60 dm³ capacity (imitating the trailer braking system) connected to the supply coupling. A test of this type is also normally carried out during validation tests of farm tractors [2].

To assess the simulation curves obtained, experimental curves recorded during tests of the pneumatic system of the tractor were introduced into the computer model in the form of the "From File" component (coloured).

A set of results of the simulation tests, obtained in the form of time histories of the air pressure p_v and temperature T_v in the dummy reservoir and of the volumetric delivery rate Q_k of the compressor have been presented as an example in Fig. 3. In equation (11), from which the temperature of the airflow generated by the compressor was calculated, the polytropic exponent was assumed as n=1,26. A p_{ve} pressure time history obtained from experiments has also been plotted on the graph. The time histories, obtained from the simulation and experimental tests, were used to validate the computer model. Results of the Kolmogorov-Smirnov non-parametric test calculated in the Matlab program for a significance level of

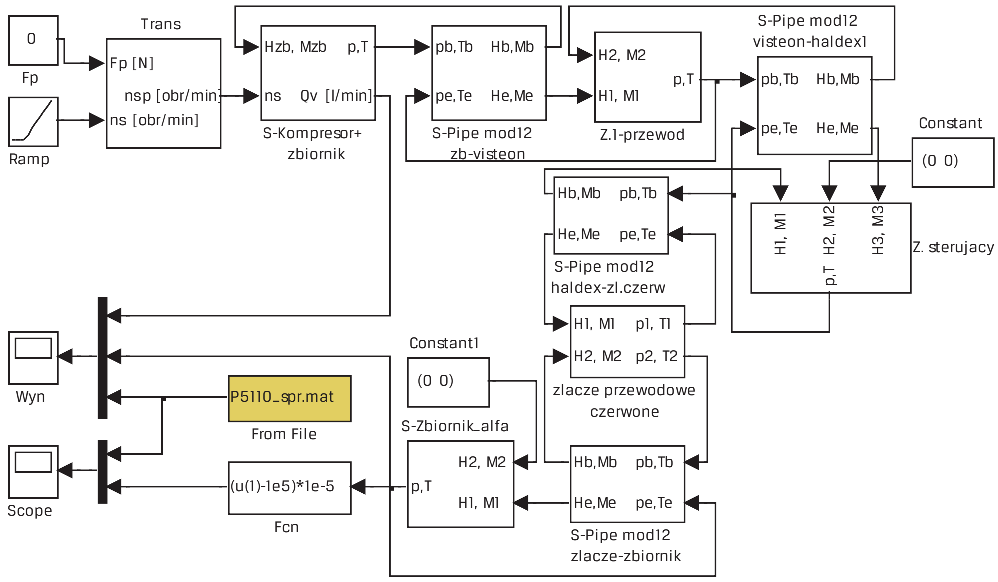

Fig. 2. Block diagram of the pneumatic system of the Pronar 5110 tractor, used for the simulation of functioning of the air supply system (compressor model 601.23.931 manufactured by Polmo Praszka).

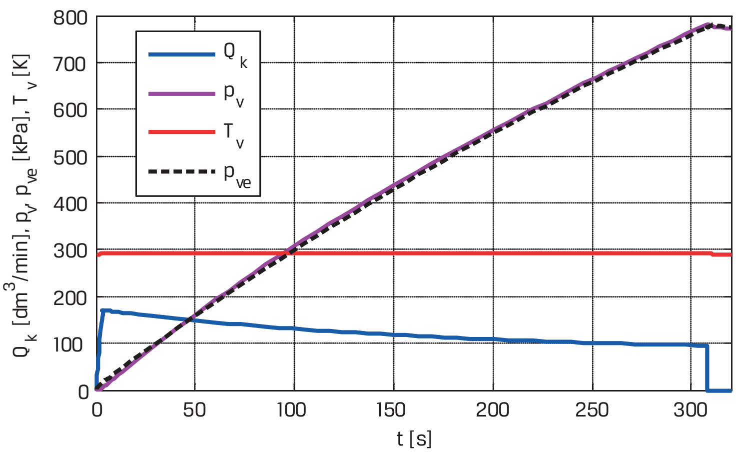

Fig. 3. Results of a simulation test of operation of the air supply part of the pneumatic system of the Pronar 5110 tractor, carried out to check the compressor delivery rate: Q_k – volumetric delivery rate, p_v, T_v – pressure and temperature, p_{ve} – pressure obtained from experiments.

0.05 (k_{s2}=0.0198<0.1923 for 101 points) and the value of the determination index R^2=0.999 confirmed the adequacy of the computer model.

Based on the curves obtained for the model, the time of achieving a minimum acceptable pressure of 6.5 bar in the dummy air reservoir was determined. This time value differed by 2.1 s from the time measured during experiments (245.6 s), which again confirmed the accuracy of the computer model to be satisfactory from the point of view of the modelling purpose.

4. Conclusions

The mathematical model of the air supply system as developed within this study may be used for the construction of a computer model of the pneumatic system of farm tractors and utility vehicles in order to forecast, by simulation methods, the system performance characteristics at the early design stage. The adequacy of the computer model of the air supply system, built in the Matlab-Simulink program, was confirmed by results of the Kolmogorov-Smirnov test where a comparison was made between the experimental and simulation time histories of the system pressure, obtained when verifying the correctness of selection of delivery rate of the compressor to be installed in the Pronar 5110 farm tractor.

References

[1] BLOCH, H.,P.: *A Practical Guide to Compressor Technology.* John Wiley & Sons, Hoboken, New Jersey, 2006.

[2] KAMIŃSKI Z., CZABAN J.: *Propozycja programu badań układów hamulcowych ciągników rolniczych (Proposal of a program of testing braking systems of farm tractors). MOTROL – Motoryzacja i energetyka rolnictwa (MOTROL – Motorization and power industry in agriculture),* Vol. 8 (edited by Eugeniusz Krasowski), Polish Academy of Sciences, Division in Lublin, Commission of Motorization and Power Industry in Agriculture, 2006, pp. 92-100.

[3] KAMIŃSKI Z., KULESZA Z., MIATLUK K., MIATLUK M: *Dynamika pneumatycznych układów napędowych (Dynamics of pneumatic driving systems)*, edited by F. Siemieniako. Rozprawy naukowe No. 145. Wydawnictwo Politechniki Białostockiej, Białystok 2007.

[4] MIATLUK M., AVTUŠKO V.: *Dinamika pnevmatičeskich i gidravličeskich privodov avtomobilej.* Mašinostroenije, Moskva 1980.

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