ISSN: 1231-4005
e-ISSN: 2354-0133
DOI: 10.5604/12314005.1217189

MODEL IDENTIFICATION OF ACTIVE PNEUMATIC VIBRATION REDUCTION OPERATOR'S SEAT OF MOBILE MACHINES

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Abstract

Low-frequency vibrations, generated in mobile machines during their driving, could be reduced only by use of active or semiactive methods. In conditions of low-frequency vibrations, energy dissipation in the machines tires slightly reduces the intensity of the vibration. Unsprung mobile machines are usually equipped with system of vibration isolation, which is located on the way of vibration propagation, between the vibration source and the protected object (the operator of the machine). Generally, controlled seat suspension is used. In the case of the active suspensions, it is necessary to provide external energy, e.g., in the form of compressed air. The compressed air has the advantage that it is generally available in working machines as the working fluid and has its environmentally friendly properties (leaks do not contaminate the environment). This article is the result of the continuation of work on active methods of vibro-activity lowering in mobile machines, which resulted in, among others, elaboration of simulation model of the active operator's seat suspension with controlled pneumatic actuator. Currently aim of the study was experimental verification of the theoretical results; this verification was performed on the laboratory test bench. In the stand tests, special attention was paid on the assumed models of friction and thermodynamic phenomena in pneumatic actuator, as well as on the control system. Experimental tests were carried out under conditions of harmonic excitations, coming from the electromechanical vibration exciter with controllable pitch and frequency. Data acquisition system and control circuit of the proportional directional control valve, supplying compressed air to the actuator were implemented using Matlab-Simulink Real-Time software. Identification of the simulation model allows for getting the right parameters of the seat suspension. In addition, parametric optimization of the seat suspension system and functional optimization of control strategy would be possible in the next step.

Keywords: *unsprung machines, active reduction of vibration, controlled suspension seats, pneumatic actuator*

1. Physical model of the operator's seat in an unsprung mobile machine

The process of object identification involves determining of the parameters of a mathematical model describing the accepted model of the physical object. The complexity of the physical and mathematical model should be adapted to the purpose of research.

In most cases, vibration reduction is effected using controlled seat suspensions. Mobile machines during the ride induce intensive low-frequency vibrations, which cannot be effectively reduced by passive methods.

Energy dissipation in wheel tires reduces the vibration intensity in a minor degree only. Active reduction systems, on the other hand, require an external source of energy, for example compressed air. Fig. 1 shows the model of an active seat suspension in which a controlled pneumatic cylinder exerts the force $F(s)$ upon the, seat platform accommodating the object to be vibro-isolated with the weight G [4, 5].

The absolute motion of the object represented by the coordinate – q is the resultant of the platform motion with respect to the vehicle floor – $y(s)$ and of the floor vibration – $z(t)$. Active seat suspension systems incorporate relief springs to reduce the energy expenditure.

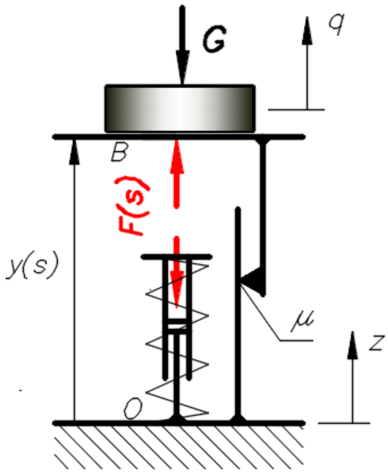

Fig. 1. Model of a dynamic system of the active seat suspension

The environment thus presents a complex of ecologic systems in which life is developed. Transportation, in fact, influences the environment in two ways:

The absolute movement of the seat with the operator is expressed by the equation:

$$q(t) = y[s(t)] + z(t). \tag{1}$$

The relative seat movement – $y(s)$ is determined by the controllable length of the pneumatic cylinder – s. In this case $y(s) = s$ (omission of the kinematic system structure).

Taking into account friction in sliding pairs, vertical vibration of the operator sitting on the seat mounted as in Fig. 1 are governed by the equation:

$$m\frac{\mathrm{d}^2 q}{\mathrm{d}t^2} = F(s) - T_{1,2}\cdot \mathrm{sign}\left(\frac{\mathrm{d}s}{\mathrm{d}t}\right) - G. \tag{2}$$

View of the laboratory stand with physical model of the active suspension of the operator's seat is presented in Fig. 2 and 3. Test stand includes both mechanical and pneumatic system.

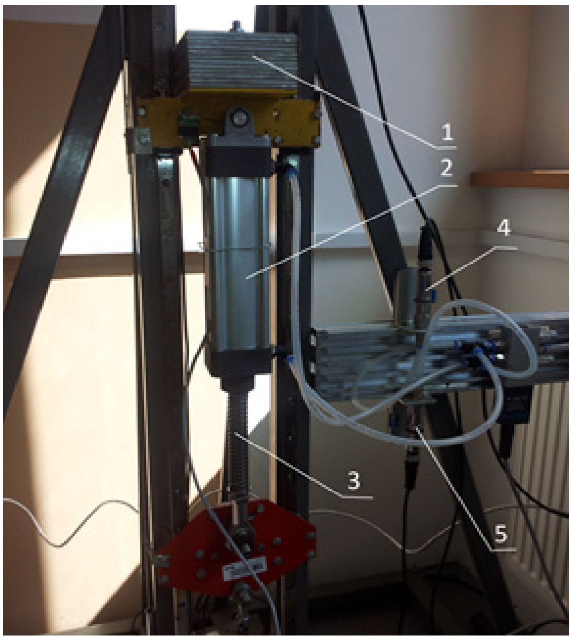

Fig. 2. General view of the test stand; 1 – weight G, 2 – pneumatic cylinder, 3 – spring, 4, 5 – pressure transducer

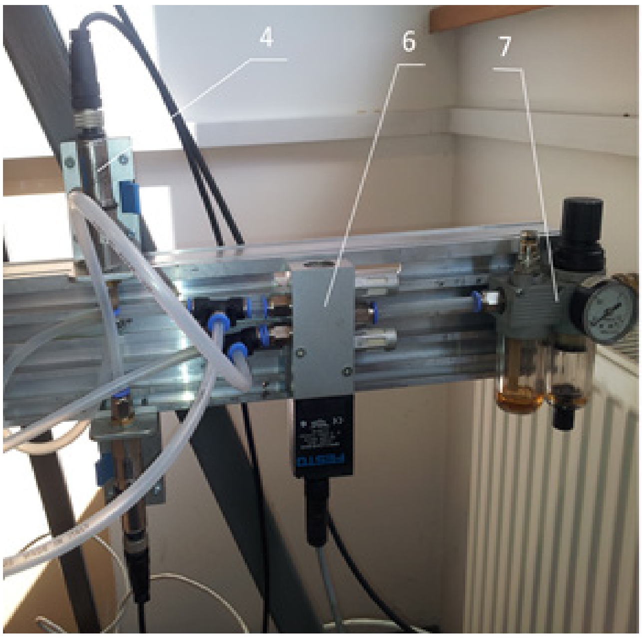

Fig. 3. View of the pneumatic control unit; pneumatic cylinder, 4 – pressure transducer, 6 –proportional directional control valve, 7 – air conditioner

2. Model of a controlled single-acting pneumatic actuator

The total force $F(s)$ determining the seat mount mechanism's motion involves the active component due to air pressure, friction in the cylinder sealing, spring response and viscous friction:

$$F(s) = p_1 A_1 - p_2 A_2 + F_T - c(s - s_{\max}) - k \frac{\mathrm{d}s}{\mathrm{d}t}. \tag{3}$$

Nomenclature used in the description of the mass-free single-acting pneumatic cylinder:
A_1 – surface area of the piston ($A_1 = 2.827 \cdot 10^{-3}$ m^2),
A_2 – surface area of the piston rod ($A_2 = 3.142 \cdot 10^{-4}$ m^2),
p_1 – pressure in the piston chamber in the cylinder,
p_2 – pressure in the piston rod chamber,
p_A – atmospheric pressure ($p_A = 1.01325 \cdot 10^5$ Pa),
p_0 – pressure supplied from the bottle ($p_0 = 5.0 \cdot 10^5$ Pa),
$\dot{M}_d$ – mass flow rate of air supplied to the cylinder,
$\dot{M}_w$ – mass flow rate of air released from the cylinder,
c_p – specific heat under constant pressure,
c_v – specific heat under constant volume,
V – volume of space beneath the piston,
T – air temperature in the in the piston chamber in the cylinder,
T_0 – temperature of supplied air (ahead of the slit $T_0 = 293$ K),
R – gas constant when $R = 287.9$ J/(kg·K).

Recalling the laws of:
– mass conservation,
– energy conservation.

The continuity equation and the energy balance equations are derived for the mass flow rates of air supplied and released from the cylinder [1-4, 6].

Recalling the Clapeyrone's law for ideal gas in the differential form, we get a system of differential equations governing the pressure and temperature changes in the piston space:

$$\frac{\mathrm{d}p}{\mathrm{d}t} = \frac{Rc_p}{Vc_v}\left[(\dot{M}_d T_0 - \dot{M}_w T) + \frac{1}{c_p}\frac{\mathrm{d}Q}{\mathrm{d}t} - \frac{p}{R}\frac{\mathrm{d}V}{\mathrm{d}t}\right], \tag{4}$$

$$\frac{\mathrm{d}T}{\mathrm{d}t} = T\left[\frac{1}{V}\frac{\mathrm{d}V}{\mathrm{d}t} + \frac{1}{p}\frac{\mathrm{d}p}{\mathrm{d}t} - \frac{RT}{pV}(\dot{M}_d - \dot{M}_w)\right]. \tag{5}$$

When the effects of heat exchange between the cylinder walls and the ambience are neglected, the term dQ/dt can be removed from Eq. (4), which implicates the adiabatic process.

$$\beta = \frac{p}{p_0} \quad \rightarrow \quad v = \begin{cases} 0.53^{\frac{1}{\kappa}} \sqrt{\dfrac{2\kappa R T_0}{\kappa - 1}\left(1 - 0.53^{\frac{\kappa-1}{\kappa}}\right)} & \text{if} \quad \beta \le 0.53 \\ \left(\dfrac{p}{p_0}\right)^{\frac{1}{\kappa}} \sqrt{\dfrac{2\kappa R T_0}{\kappa - 1}\left[1 - \left(\dfrac{p}{p_0}\right)^{\frac{\kappa-1}{\kappa}}\right]} & \text{if} \quad \beta > 0.53 \end{cases}. \tag{6}$$

Flow velocity is determined by the ratio of pressure upstream and downstream the throttling slit – β. For the flow delivered to the cylinder, air velocity could be described by the formula (6).

In the adiabatic process the exponent $\kappa = 1.4$. The flow rate is proportional to flow velocity – v, and the flow field – $A_f(U)$, being a function of the command (voltage) signal – U.

$$\dot{M}_d(U,\beta) = \lambda \cdot \varphi \cdot \frac{p_0}{RT_0} \cdot v(\beta) \cdot A_f(U)\,, \tag{7}$$

where:
λ – contraction coefficient ($\lambda = 1$),
φ – velocity coefficient ($\varphi = 0.975$).

The dependence $A_f(U)$ stems from the characteristic of the applied directional control valve and it was determined in experimental studies. Friction in the sealing in the piston and rod is another important issue. Experimental tests reveal that friction forces in sealings in the cylinder have a complex nature and are dependent on the piston velocity relative to the cylinder.

The „stick-slip” phenomenon, i.e. piston creeping with quasi-zero velocity under large loads, causes that only after the stick-slip velocity – Δv_s is exceeded will the friction force decrease significantly [6]. Depending on the type of applied sealing and lubrication, this velocity falls in the range $\Delta v_s \in [0.001-0.01]$ m/s and from that instant the friction of motion (kinetic friction) begins [1-4]. This effect is best captured by the Stribeck's curve and the Karnopp's method provides a concise mathematical formula expressing the static and kinetic friction:

$$F_T = -\begin{cases} \operatorname{sign}[p_1A_1 - p_2A_2 + F_T - c(s - s_{\max})] \cdot \min[F_{FS}, |\, p_1A_1 - p_2A_2 + F_T - c(s - s_{\max})\,|] & \text{if } \left|\dfrac{\mathrm{d}s}{\mathrm{d}t}\right| < \Delta v_s \\ \operatorname{sign}\left(\dfrac{\mathrm{d}s}{\mathrm{d}t}\right) \cdot \left[F_C + k\left(\left|\dfrac{\mathrm{d}s}{\mathrm{d}t}\right| - \Delta v_s\right)\right] & \text{if } \left|\dfrac{\mathrm{d}s}{\mathrm{d}t}\right| \geq \Delta v_s \end{cases}. \tag{8}$$

The forces of static friction F_{FS} and kinetic friction F_{FC} are determined by the difference of pressures acting upon the sealing elements [1, 2, 6].

3. Structure of the pneumatic system

Schematic diagram of the pneumatic system of the active seat suspension is shown in Fig 3. Flow control is effected through throttling of the mass flow rate of air supplied and released from the cylinder.

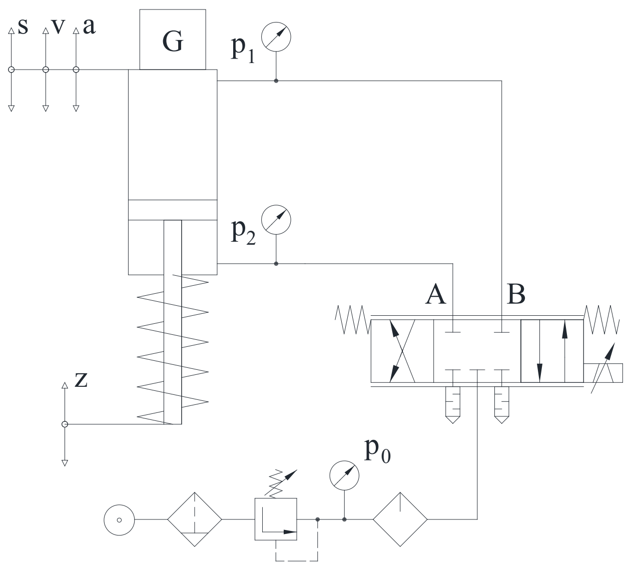

Fig. 4. Schematic diagram of control of the pneumatic cylinder

4. Identification test of the seat suspension system

Due to complexity of the nonlinear mathematical model of the seat suspension active system, the linearization procedure was abandoned. For the purposes of model identification, the weight *G* was attached to the top plate serving as the operator's seat. This eliminates the problem of unilateral constraints. For practical reasons, the total friction force characteristic was determined, including friction in sliding pairs and in pneumatic cylinder. For this purpose, in the absence of kinematic excitations (i.e. $z(t) = 0$), quasi-zero flow is set through the proportional valve, so as to make the effect of „stick-slip”. Exemplary results of this test are presented in Fig. 5.

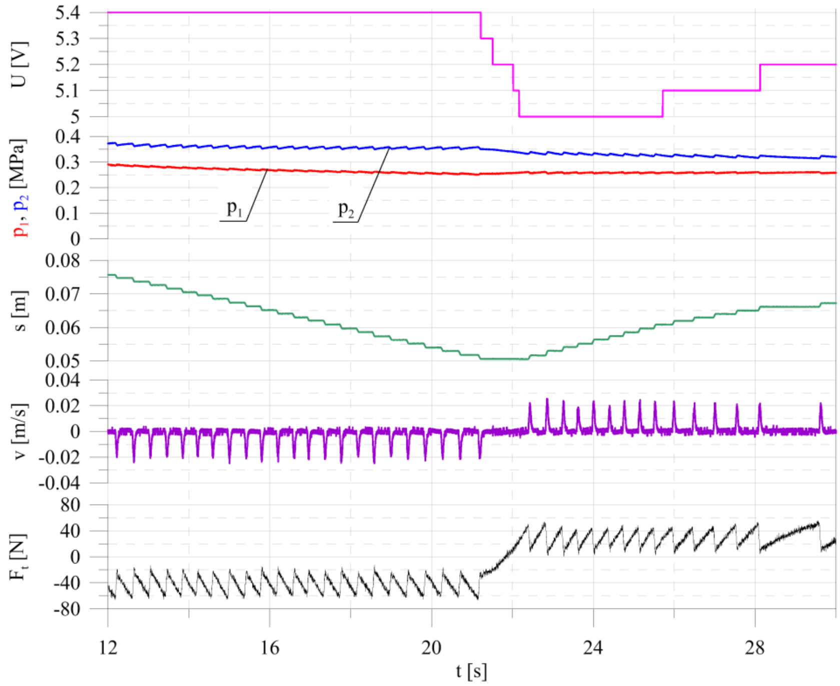

Fig. 5. Exemplary waveforms of system's quantities at low velocity

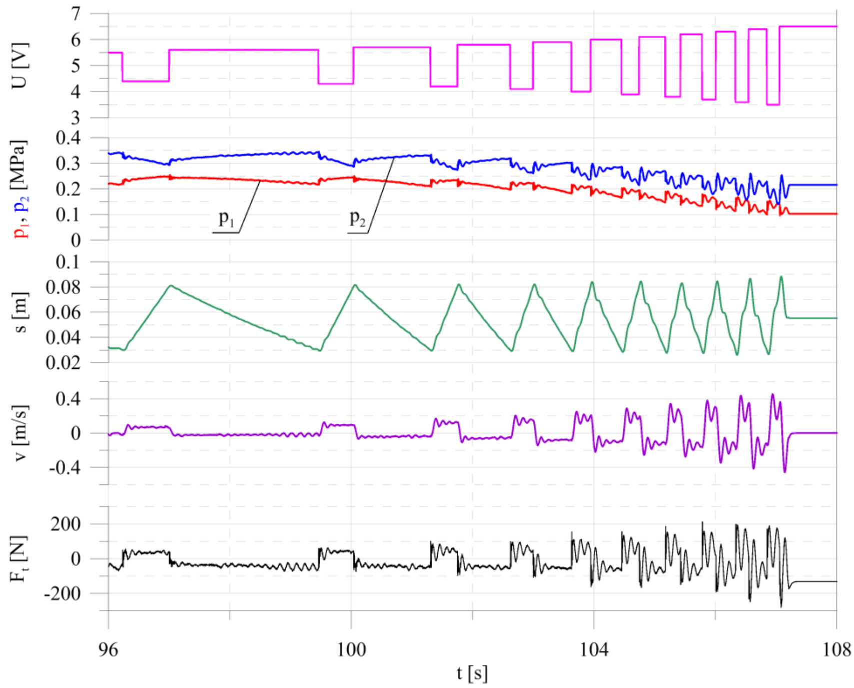

Fig. 6. Exemplary waveforms of system's quantities at rising velocity

Measurement of cylinder piston displacement s, together with the known spring rate c, allows determination of the spring reaction. Whereas by measuring the seat acceleration and pressure p_1 and p_2 in the cylinder chambers (Fig. 4) at a known load G, total friction force can be determined on the basis of equations (2) and (3). The experiment was also carried out for higher values of velocity – Fig. 6, in order to finally get the complete characteristic of the friction force (including static and kinematic phases). Achieved characteristics, presented in Fig. 7 were approximated by linear dependencies; various for cylinder extend and retract movement.

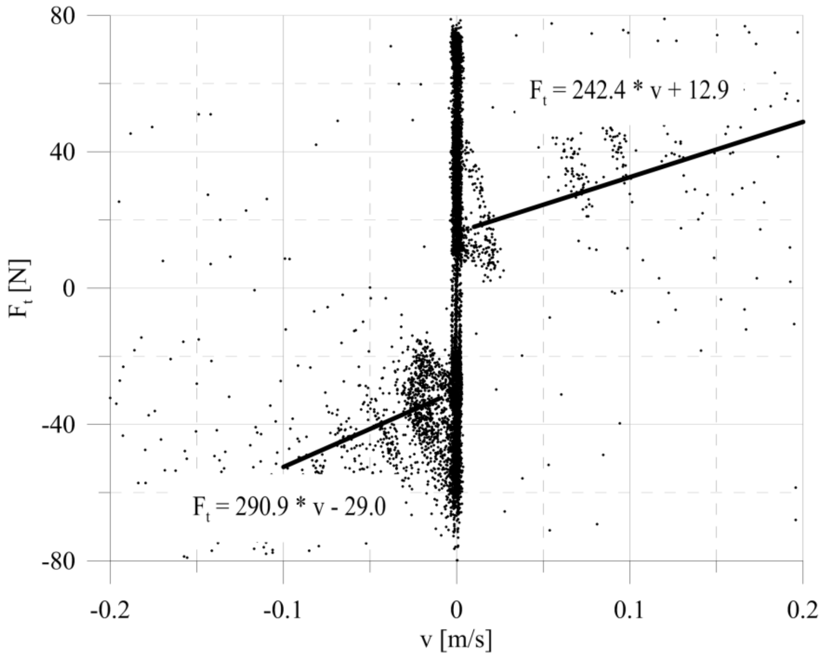

Fig. 7. Characteristic of friction force as function of velocity

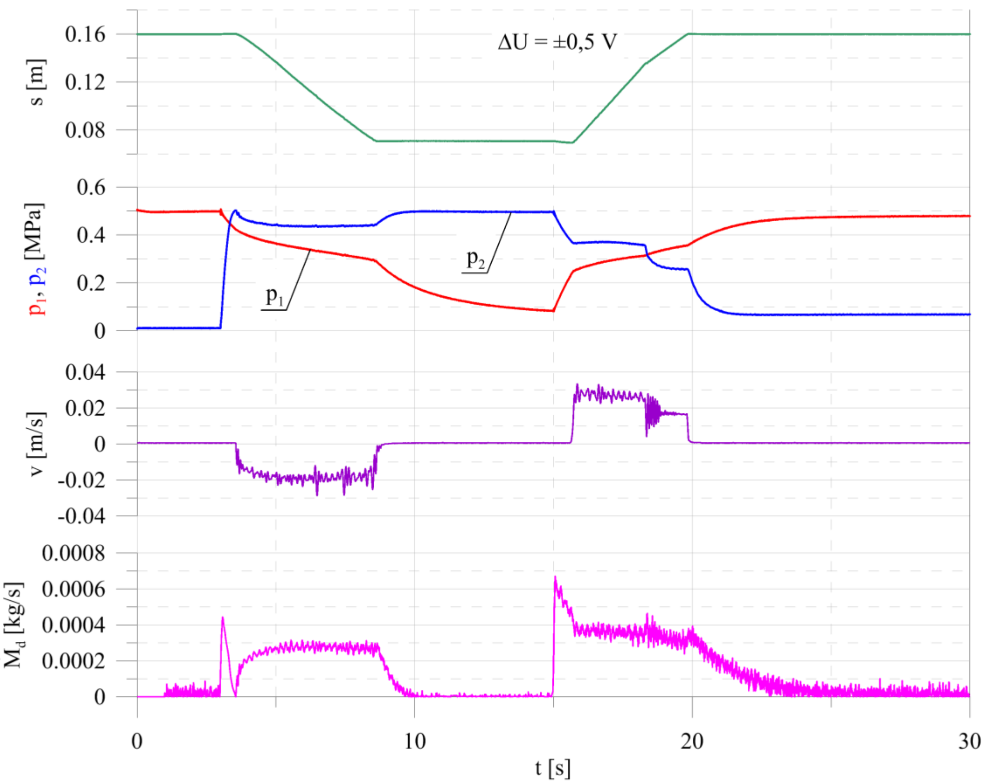

Fig. 8. Exemplary waveforms of system's quantities during mass flow test for voltage ΔU = ±0.5 V

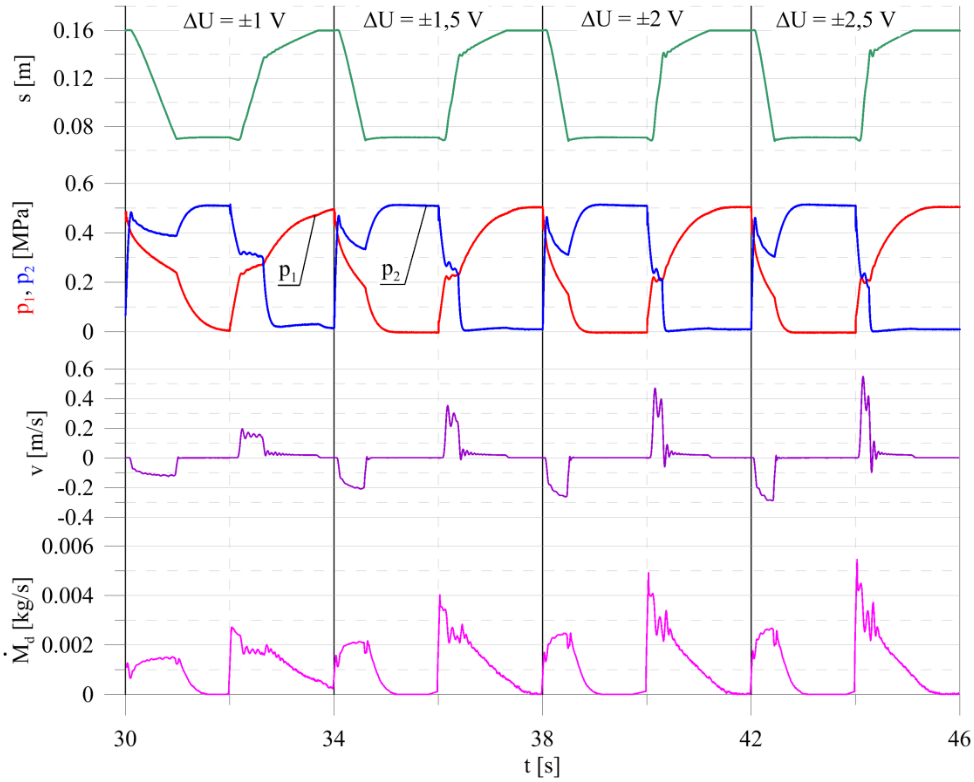

Fig. 8. Exemplary waveforms of system's quantities during mass flow test for voltage ΔU=±0.5 V

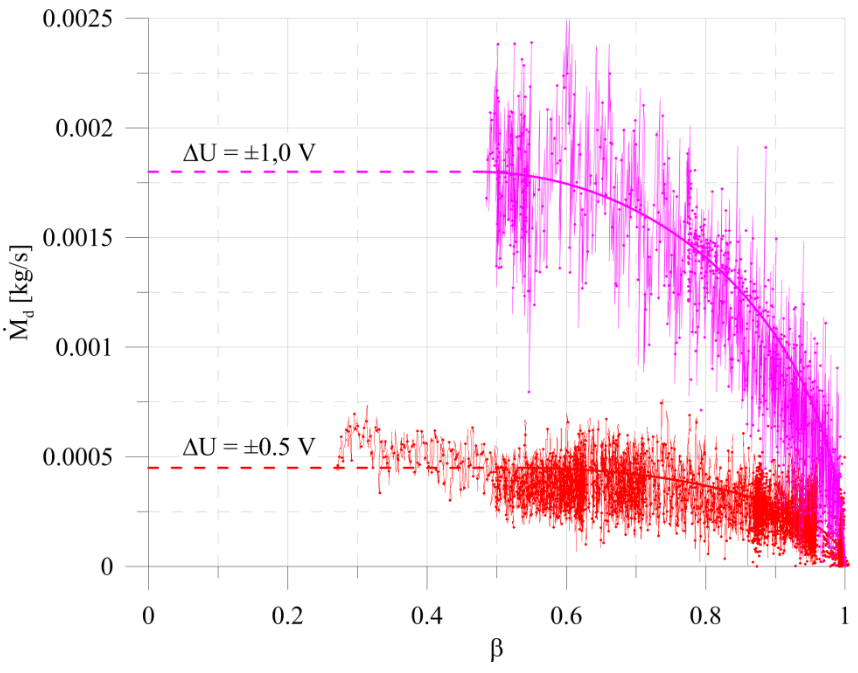

Fig. 10. Mass flow characteristic as function of pressure ratio β

The characteristics of air flow for the proportional directional control valves is not always published by the manufacturers. In the experiment, the mass flow rate was determined from the equation (4), taking into account the measured values such as: the volume of the cylinder chambers V_1, V_2, and pressures p_1, p_2. It was assumed constant supply pressure and a constant supply air temperature T_0 = const. Experimental tests were carried out for different values of the control voltage U. In Fig. 8 and 9 time plots of measured values are presented, while in Fig. 10 characteristics of mass flow rate as a function of pressure ratio β is posted.

5. Summary

The identification process of the seat suspension active system used in the calculation procedure was developed in Mathcad and Matlab-Simulink. The model of the seat suspension active system takes into account thermodynamic processes in the pneumatic actuator as well as friction in the piston and sliding pair's ceilings. For the model parametrization and also to test different control algorithms laboratory stand with physical model of the system was build. In the model identification experiments, three independent transducers of kinematic quantities of pneumatic actuator motion have been used. Cylinder displacement was measured by inductive transducer, velocity – by incremental encoder and acceleration of the actuator accordingly via an accelerometer. In performed tests, a very good agreement of friction characteristics with Karnopp curve was achieved. The characteristics of mass flow rate require further refinement. The system was found to be feasible and implementable with respect to every parameter. Future research efforts will concentrate on development of complex control.

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