FUZZY CONTROLLER TO CONTROL THE ACTIVE AIR SUSPENSION

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Abstract

The paper presents the practical implementation of active seat suspension used in vehicles and machinery. The test object is a passive suspension system, in which the vibro-insulating properties have been improved by using an active system controlled by a pneumatic bellows cylinder. Bellows cylinders in pneumatic drives act as single-acting cylinders - push cylinders. The filled bellows cylinder under constant pressure acts as an air spring. The volume of the air cushion changes under the pressure inside. Proportional pressure valve was used to control the bellows cylinder and non-contact relative displacement laser triangulation sensor was used to measure displacement of the seat. A Proportional-Derivative-Fuzzy Logic Controller (PD-FLC) was used for the control. The use of Fuzzy Logic Controller (FLC) enables the transition from a quantitative description to a qualitative process. The system structure with a designated output control function has been presented. Test results on the seat suspension vibro-insulating properties of a working machine are presented.

Keywords: Fuzzy Logic Controller; air spring; pneumatic actuators; active air suspension

1. Introduction

Pneumatic suspension systems are used in many areas where there is a need to effectively eliminate the effects of vibration. They are used in automotive industry, from seat mounts to suspension systems, improving the comfort, safety and driving conditions of vehicles. Machine operators, drivers are exposed to prolonged exposure to vibrations, which may lead to occupational disease – vibration white finger. Vibrations from machines act on humans locally and are transmitted through the upper limbs and the ground and affect the whole body. People working in a sitting position, i.e.: drivers, machine operators, construction vehicle and tractor drivers are exposed to general vibrations, which are transmitted to the body from the ground through the pelvis and back. People exposed to vibrations experience disorders of the nervous system including decreased concentration and prolonged reaction time, sensory disorders and lesions of the motor organs. Particularly adverse changes in the human body are caused by resonance vibrations of internal organs. The range of these frequencies depends on the individual human structure, position (sitting, standing) and ranges from (2-12) Hz for the internal organs and chest. The resonance frequencies for the head are in

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the range of (20-30) Hz, and for the sight organs (60-90) Hz. As knowledge and technology advances, operators' cages and drivers' cabins are equipped with seats that limit the frequency of transmitted vibrations to the lowest (<20 Hz). However, the vibration white finger in the total number of occupational diseases is still significant and growing. Research has been ongoing for many years to eliminate or reduce these effects through the use of state-of-the art suspension technologies for entire machines and vehicle seats [1, 3, 8]. Pneumatic actuators are used in vibration isolation systems based on electro-pneumatic proportional and servo technology [2]. Its basic components are: bellows pneumatic spring and control electro-pneumatic valve (pressure regulator). Rubber bellows are armed with cross cord and are made in the form of folded, bagged or membrane cylinders [11].

Adjustable shock absorbers allow building a suspension with variable damping. There are several ways in which the shock absorber damping can be changed in real time. Currently there are two main groups on the market. The first of these includes shock absorbers, in which the damping characteristics are changed by using various types of valves, which can be controlled pneumatically or electromagnetically. The second group of shock absorbers uses variable rheological properties of special fluids used in shock absorbers instead of standard oil, depending on the magnetic field strength.

The basic way to counteract the negative effects of vibrations on machine and vehicle operators is to construct seat suspensions with vibro-insulating properties. These are solutions with passive and active vibration isolation. The machinery seats with passive vibration isolation use components with linear elasticity k and non-linear damping c with constant characteristics (Figure 1).

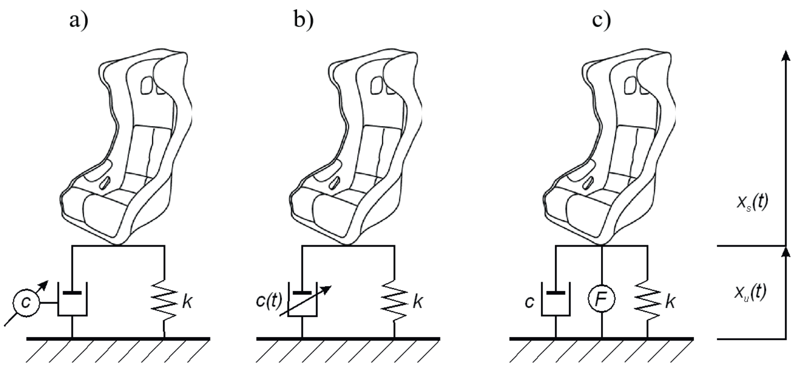

Fig. 1. Block diagram of the active seat suspension control system: a) adjustable, b) semi-active, c) active

In this type of suspension, the damping and elasticity coefficients are adjusted to the prevailing operating conditions [4]. Many constructions use solutions that allow the operator to change the damping coefficient c depending on the individual assessment of operator's weight and operating conditions (Figure 1a). Effective damping can be achieved automatically or manually by setting a default pressure (pneumatic spring) or tension in the magnetorheological damper. A development of this method are semi-active systems, where the damping can be variable in time (Figure 1b). The damping coefficient c changes continuously, so that it can be carried out while the machine is exposed to vibrations caused by road conditions or by the vibrations caused by its equipment itself. The active suspension system (Figure 1c) is equipped with an actuator that generates the force F needed to eliminate vibrations [10]. The well-designed automatic control system for this suspension continuously adapts to the current operating conditions of a working machine [5, 7, 9]

The aim of the research was to develop a fuzzy controller to control the active suspension of the working machine seat. The task of the control system is to generate the control signal so that the error of relative displacement of the operator's seat is minimal. The control signal acts on a proportional pressure valve that generates a force in the air bellows that eliminates seat vibrations.

2. Air Spring Model

Bellows cylinders are used as actuators in pneumatic drive systems of machines and equipment and as air springs (cushions) in vibration isolation of machines, vehicle suspension, etc. [2]. Bellows cylinders in pneumatic drives act as single-acting cylinders - push cylinders. The operation of these cylinders involves filling (applying pressure) and emptying (venting) them. The filled bellows cylinder under constant pressure acts as an air spring (Figure 2). The volume of the air cushion changes under the pressure inside. The change in the bellows shell deformation is calculated according to the elementary triangular deformation (OAA') in relation to the axis of rotation. The total volume of the bellows is defined as the sum of the toroid volume and *R*-radius cylinder volume.

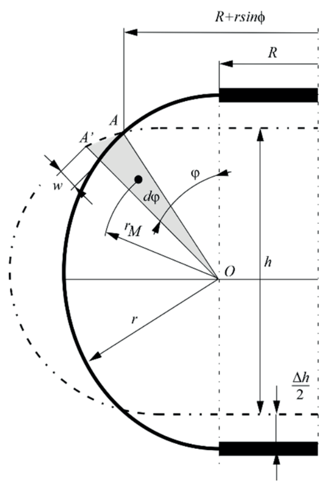

Fig 2. Air cushion computation diagram

Assuming that the distance of the mass centre for an elementary triangle is in 2/3 of its height and applying Pappus's centroid second theorem, the total volume of the air cushion after deflection can be determined according to the formula:

$$V_{as} = 2\pi \int_0^{\pi} \frac{1}{2}(r+w)^2 \cdot (\frac{2}{3} r \sin\phi + R) d\phi + \pi R^2 \cdot 2(r+w) \quad (1)$$

The volume increase at displacement is determined by the formula:

$$\Delta V_{as} = 2\pi \int_0^{\pi} (wr + \frac{1}{2}w^2) \cdot (\frac{2}{3} r \sin\phi + R) d\phi + \pi R^2 \cdot 2w \quad (2)$$

Rounding the displacement can be expressed as follows:

$$w = h \cos(2\phi) \quad (3)$$

By substituting (3) expression for equation (2), a formula for the volume growth of the bellows is obtained:

$$\Delta V_{as} = 2\pi\{[-\frac{4}{9} h r^2 + h^2(\frac{28}{90} r + \frac{\pi}{4} R)] + R^2 h \cos(2\phi)\} \quad (4)$$

The model assumes that only compressed air participates as an elastic solid in assessing the air spring rigidity. In equation (4) R is constant parameter because it is metal connecting element length. The variable h is the most significant, while r is of little importance, therefore a constant value of r was adopted. The energy of gas under pressure in V_{as} volume is $E = \int_{V_{as}} P dV$.

For axial compression of the toroidal bellows the change in internal energy E results from the equation:

$$\partial E = P\partial(\Delta V_{as}) = 2\pi P\{[-\frac{4}{9} r^2 \partial h + 2h\partial h(\frac{28}{90} r + \frac{\pi}{4} R)] + R^2 \cos(2\phi)\partial h\} \quad (5)$$

The force generated by the bellows can be expressed as follows:

$$F_{as} = P\frac{\partial(\Delta V_{as})}{\partial(h)} = 2P\pi\{[-\frac{4}{9} r^2 + 2h(\frac{28}{90} r + \frac{\pi}{4} R)] + R^2 \cos(2\phi)\} \quad (6)$$

The tangent rigidity of the air spring associated with the change in compressive force in equation (5) is calculated by taking into account the relative displacement h of the flanges (Figure 2):

$$K_{as} = \frac{\partial F_{as}}{\partial(h)} = P\frac{\partial^2 \Delta V_{as}}{\partial(h)^2} + \frac{\partial}{\partial(\Delta V_{as})}\left(\frac{\partial \Delta V_{as}}{\partial(h)}\right)^2 \quad (7)$$

Equation (7) shows that the design tangent rigidity of an air spring is:

$$K_{as} = 2P\pi(\frac{28}{90} r + \frac{\pi}{4} R) + \{\pi[-\frac{4}{9} r^2 + 2h(\frac{28}{90} r + \frac{\pi}{4} R)] + \pi R^2 \cos(2\phi)\}^2 \frac{\partial P}{\partial V_{as}} \quad (8)$$

After taking into account the rate of change in pressure P due to the displacement of the bellows, the adiabatic process equation is assumed:

$$\left|\frac{\partial P}{\partial V_{as}}\right| = \frac{\kappa P_0 {V_0}^{\kappa}}{V^{\kappa+1}} \tag{9}$$

where:

V_0 – volume in the initial state,

P_0 – pressure in the initial state,

κ =1.4 - heat capacity ratio for dry air (20oC).

After taking into account (8), the pneumatic rigidity of the bellows is equal:

$$K_{as} = 2P\pi(\frac{28}{90}r + \frac{\pi}{4}R) + \frac{\kappa P_0 {V_0}^{\kappa}}{V^{\kappa+1}}\{\pi[-\frac{4}{9}r^2 + 2h(\frac{28}{90}r + \frac{\pi}{4}R)] + \pi R^2 cos(2\phi)\}^2 \tag{10}$$

Simulation tests were carried out for the proposed air spring model. Figure 3 shows the model response for sinusoidal waveforms for selected frequencies. These frequencies correspond to the frequencies most harmful to human health (internal organs).

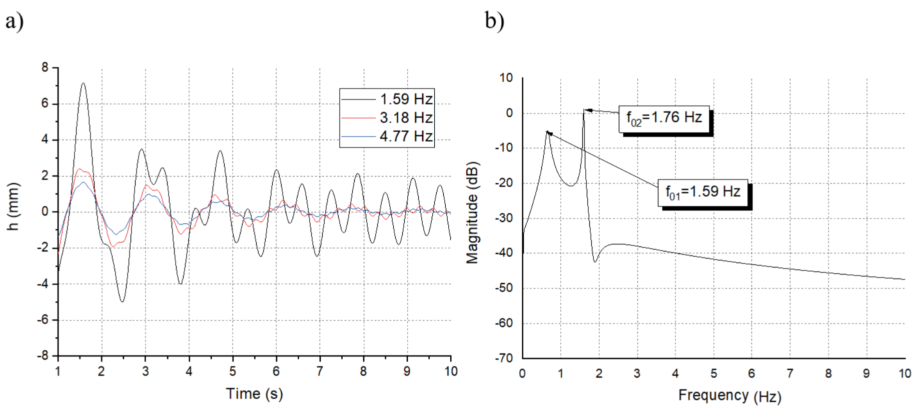

Fig. 3. Impulse response (a) of the air spring and frequency response (b) of the air –spring model: f_{01} - natural frequency, f_{02} –input frequency

In the simulation system of the bellows actuator model, a decrease in amplitude is observed in the natural frequency range f_{01} as well as in the original input frequency range f_{02}. The natural frequency of the mechanical system was determined by the Fast Fourier Transform (FFT) (Figure 3b) from the impulse response (Figure 3a).

For bellows used as air springs, the correct degree of vibration isolation is in the frequency range f_{02} >1.4f_{01}, while a high degree of vibration isolation is in the frequency range f_{02} >>f_{01}.

3. Control System

Problems related to optimal and comfortable damping of vehicle operators' seats have contributed to the search for new control techniques based on artificial intelligence methods – fuzzy logic. The use of Fuzzy Logic Controller (FLC) enables the transition from a quantitative description to a qualitative process. In traditional control systems, control algorithms are designed analytically, based on the mathematical model of the controlled system and on the required objective of the control system. Applying fuzzy logic methods, knowledge gained during process operation and handling, or the knowledge of operators, i.e. intuitive algorithms for controlling control objects, can be saved with verbal logic converted into mathematical operation and used in the control process. Fuzzy logic systems can also be used in processes where there are non-linearities, uncertainties about their parameters or other adverse features of a control object. Fuzzy logic systems are intelligent control methods in which the knowledge encoded in the rule base results from experience and intuition as well as theoretical and practical understanding of process dynamics. The use of fuzzy logic in controllers results from the fact that the knowledge of the process dynamics is not required for the correct tuning of the controller. Fuzzy control has become also popular because the actual control objects are non-linear and therefore require special control techniques, which are usually difficult, laborious and sometimes impossible to design. For such objects, designing fuzzy controllers can be much easier, and they can replace standard or state controllers [6, 11].

The primary objective for seat control is to limit the accelerations to which the driver is subjected (Figure 4). The suspension system control is based on measurements of: seat system displacement with the loading mass (xs) and relative displacement of the operator's seat (x_u - x_s) and floor (x_u).

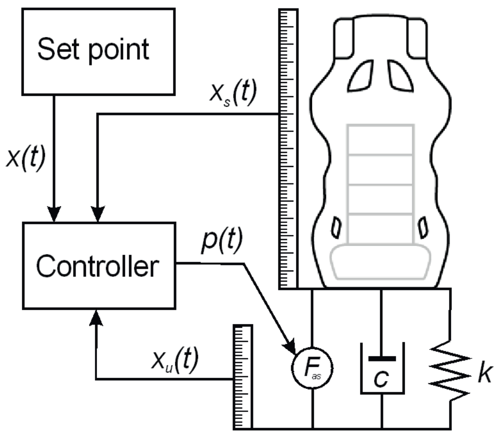

Fig. 4. Block diagram of the active seat suspension control system

The role of the controller is to adjust the pre-set pressure in the bellows actuator to minimize seat acceleration. The initial bellows pressure determination depends on the vibration-insulated mass. The Fas force generated in the active system depends on the control signal or on the current seat deflection x_u-x_s which is the measured value.

The PD-Fuzzy Logic Controller has been designed for the pneumatic control of the bellows cylinder (Figure 5). The controller generates a voltage signal $u_{Fas}(t)$ to control the pressure $p(t)$ in proportional valve based on the relative seat deflection $(x_u$-$x_s)$, which is the feedback of the FLC controller.

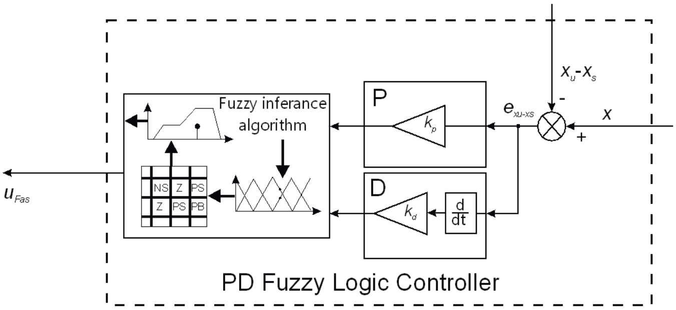

Fig. 5. Diagram of PD Fuzzy Logic Controller

The control law describing a fuzzy controller is presented in the form:

$$u_{Fas}(k) = F[e_{xu-xs}(k), e_{xu-xs}(k-1), \dots, e_{xu-xs}(k-v), u(k-1), u(k-2), \dots, u(k-v)] \quad (11)$$

where:

$u_{Fas}(k)$ – control signal, describing the relationship between the controller input and output,

$e_{xu-xs}(k)=x(k)-(x_u(k)-x_s(k))$ – control error,

$u(k-1), u(k-2), \dots, u(k-v)$ – previous values of the signal

$x_u(k)-x_s(k)$– object output,

$x(k)$– set point,

$k=t/T$– discrete time (sampling moment),

t– continuous time,

T – sampling period,

v – parameter that defines the controller order,

F – a function describing a fuzzy controller with a knowledge base containing rules IF-THEN.

The general relationship of the FLC control in direct version for v=1 can be expressed as follows:

$$u_{Fas}(k) = F[e_{xu-xs}(k), \Delta e_{xu-xs}(k)] \tag{12}$$

The equation (12) describes the relations between the $u_{Fas}(k)$ control signal and the $e_{xu\text{-}xs}(k)$ error and the $\Delta e_{xu\text{-}xs}(k)$ error change. In the fuzzy inference process, the ignition level and the fuzzy implication [11] were determined using the MIN operator, and the individual outputs of all rules were aggregated with the MAX operator. The Centre of Gravity Method (COG) was used in the defuzzification process. The 49 fuzzy Mac Vicar-Whelan rules (Table 1), which form the fuzzy controller processing platform shown in Figure 6, were used as the rule base.

Tab. 1. Base rules

Δe\e	NB	NM	NS	Z	PS	PM	PB
NB	NB	NB	NB	NB	NM	NS	Z
NM	NB	NB	NM	NM	NS	Z	PS
NS	NB	NM	NS	NS	Z	PS	PM
Z	NB	NM	NS	Z	PS	PM	PB
PS	NM	NS	Z	PS	PS	PM	PB
PM	NS	Z	PS	PM	PM	PB	PB
PB	Z	PS	PM	PB	PB	PB	PB

Rule 13

In the FLC, the surface control represents the control strategy of the FLC. The change of the surface control is achieved by modifying the rule base, fuzzy sets, rule antecedent and consequent, shapes of membership function. The following abbreviations are marked in Table 1: NB – negative big, NM – negative medium, NS – negative small, Z – zero, PS – positive small, PM – positive medium, PB – positive big. The rules can be read as follows using the example of rule 13: **IF Δe is NS and e is PS THEN u_{Fas} is Z.**

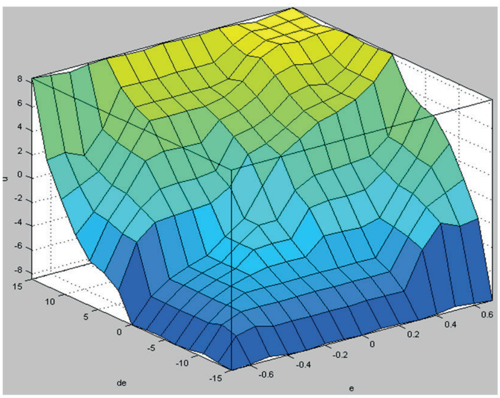

Fig. 6. Surface control of FLC

4. Experimental Stand

Figure 7 shows an experimental stand for tests on vibro-insulating properties of a working machine seat.

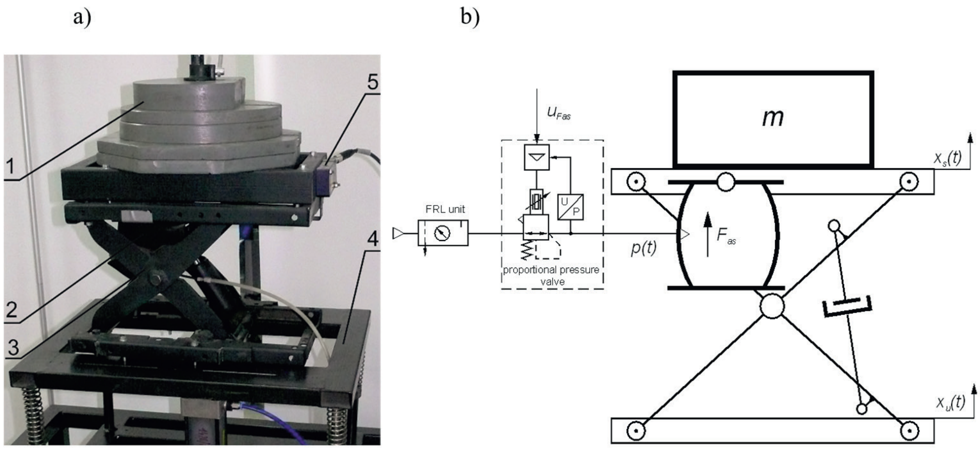

Fig. 7. General view (a) and diagram (b) of experimental stand: 1 – mass load, 2 – bellows cylinder, 3 – scissor-type working machine, 4 – vibrating table, 5 – laser displacement sensor

The basic element of the stand is the vibro-insulating seat base of the scissor-type working machine (3). The lower part of the base is attached to the vibrating table (4). The upper part allows the installation of a mass load (1). The vibrating table is driven by a pneumatic cylinder which is controlled by the Festo pneumatic straight-run servovalve MPYE-5-1/8-HF-010-B. The use of such a solution in the vibrator allows obtaining vertical displacements in the range of (0-220) mm with a frequency up to 5.09 Hz. The passive damping element is a hydraulic damper, while the active element is a pneumatic bellows cylinder (2). Piezoelectric proportional pressure value Hoerbiger Tecno Plus was used to control the bellows cylinder. For non-contact relative displacement measurement, the bench is equipped with BaumerElectric OADM (5) laser triangulation sensor.

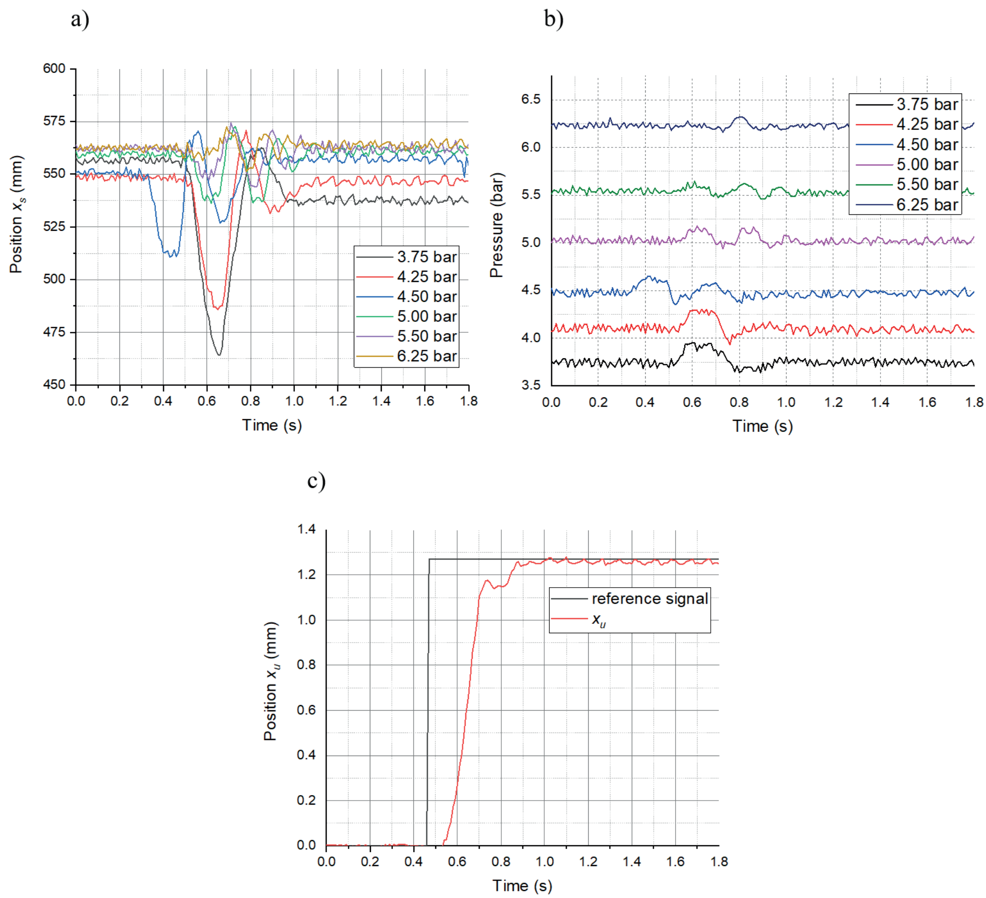

Fig. 8. Dynamic characteristics of displacement of the vibro-insulated mass $x_s(t)$ (a), pressures (b) and floor displacement $x_u(t)$(c)

Figure 8 shows the dynamic characteristics as a function of time of the vibro-insulated seat mass of 72 kg for the step function. The tests were carried out at variable pressure p (Figures 8a and 8b). Analyzing the test results presented in Figure 8b, it appears that the loaded seat is damped the fastest for pressures of 3.75 bar, 4.25 bar and 4.5 bar. However, taking into account the results concerning the displacement of the vibro-insulated mass (Figure 8a), the lowest values are achieved for pressures of 3.75 bar and 4.25 bar. Summing up, the best vibro-insulating properties were achieved by the system for pressures ranging from 3.75 bar to 4.25 bar.

5. Conclusions

Air springs are an important element of air suspensions. During the operation of springs, physical phenomena occur which significantly affect their vibro-insulating properties. There are many dynamic factors that determine the effectiveness of vibro-isolation in oscillating systems. One of them is the force produced by the bellows cylinder. Proper estimation of the value of this force is a basic task of good functioning of semi-active and active suspension systems.

The paper presents the practical implementation of active seat suspension used in vehicles and working machines. The object of the research was a passive suspension system, which used an active force control system in a pneumatic bellows actuator. The structure of the system with the designated output control function was presented. In the experimental tests of the seat suspension system, the values of the dynamic system parameters such as the operator's weight and the stiffness of the air spring were initially determined. The Fas force generated in the active system depends on the control signal or on the current seat deflection. If the actual spring deflection is very small, then even the maximum value of the control signal will not allow obtaining the desired active force. As a result of the use of the PD Fuzzy Logic Controller of the active system with adjustable pressure of the air spring, the deflection value of the vibro-insulated mass decreased.

6. Acknowledgement

The paper submitted for the first time for publication in our periodical will be treated as an original. After the peer reviews are received, all changes to the paper should only be made in the "Track changes" mode.

7. References

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