A pneumatic semi-active seat suspension

Pneumatyczne semiaktywne zawieszenie siedziska maszyny roboczej

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Presented are issues related to improvement of the vibration isolating properties of semi-active seat suspension. It shows the practical implementation of the force control system for pneumatic spring which was mounted in mass-produced seat suspension. Output control function for pressure valve of pneumatic supply system was proposed. The developed solution allowed to carry out the research of air spring applied to a semi-active suspension system.
KEYWORDS: vibro-isolation, semi-active suspension control, bellows actuator

Vibrations of mechanical vibration systems are transmitted to a human and negatively affect the functioning of tissues, blood vessels and organs. In the human body exposed to long-term mechanical vibration, there are a number of disorders that lead to permanent lesions in the form of vibration syndrome [5]. This applies to many professional groups, such as construction machinery operators and vehicle managers, including road users, who work in a sitting position. Negative effects of exposure to resonance frequencies are most severely affected by internal organs and bone system.

The range of these frequencies depends on the individual human model and is 2 to 12 Hz for internal organs [5]. The frequency of vibration of the seat suspension systems of the machines and passive vehicles is in the range of 1 ÷ 3 Hz, so they are less effective in suppressing low frequency vibrations [7].

In the early design of the cushioned driver's seats, passive metal vibration springs and hydraulic dampers were used, followed by bellows pneumatic springs [4]. Popular solutions are the ability to change the attenuation coefficient by the operator depending on the individual assessment of his weight and working conditions. The damping effect is controlled automatically or manually by setting a constant pressure value (pneumatic spring) or voltage (current) in the attenuator with varying damping characteristics (magneto-rheological liquid).

The development of this method are semi-active systems, in which values of attenuation or elasticity coefficients are variable over time and determined by the control system [3]. The low power requirements of external power sources favor the popularization of such systems. They reduce the vibration of the seat with a low frequency of inducing forces (up to 5 Hz), while maintaining good vibro-isolation properties when the frequency of impingement is higher [2].

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Construction of the experimental station

Works upon new seating suspension systems with semiautomatic and active vibration reduction systems are underway in the Department of Mechanical Technology and Metrology at the Faculty of Mechatronics and Machine Building of the Świętokrzyskie University of Technology. A semi-automatic seat suspension with pneumatic bellows spring [8] was constructed. The experimental station for the vibration-proof properties of the working-machine seat is shown in fig. 1. The basic element of this station is the pneumatic-hydraulic vibration isolator of the working seat base (1).

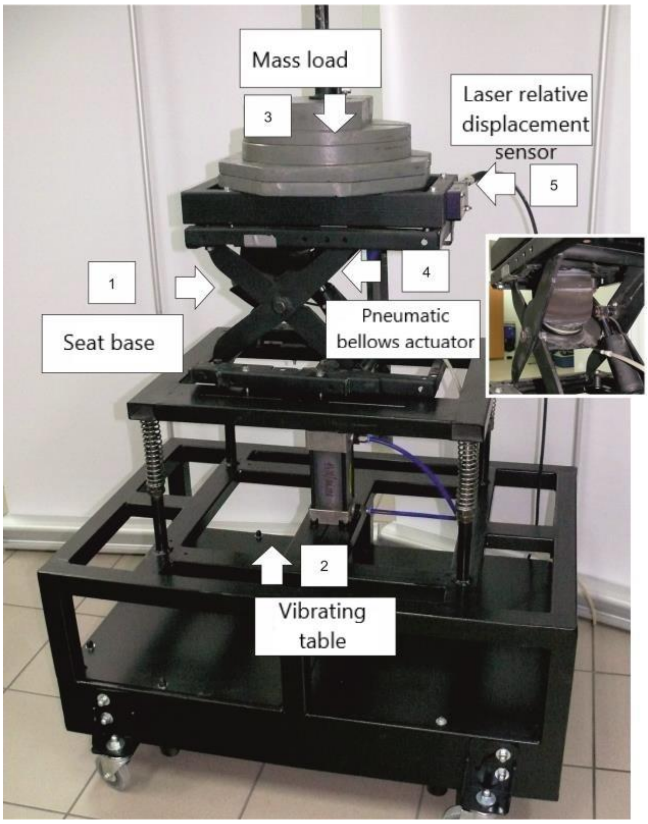


Fig. 1. Experimental position

The vibration isolator is constructed from a two-hinged quadrilateral frame to which the operator's seat is attached. With the quadrillers, pneumatic actuator and hydraulic damper are articulated. The bellows actuator used in the system is a pneumatic drive which allows the system to operate at a deviation from the axis by an angle of 10 ÷ 20°. For testing purposes, the lower part of the vibration isolator is fixed to the vibration table (2); the upper part allows mounting of the mass load (3). The vibrating table is driven by a pneumatic actuator operated by a pneumatic proportional valve [6]. This design enables vertical displacements in the range 0 ÷ 220 mm with a frequency up to 8 Hz. For the non-contact measurement of the relative displacement of the seat, Baumer Electric's OADM (5) triangulation laser transducer [9] was used. The bellows actuator (4) controls the pressure proportional valve SMC type VEP3121-2.

Modeling of the vibration isolation properties of the experimental station

The equation for motion of the vibration-insulated work machine seat can be presented in a generalized form:

$$m\ddot{p} + c\dot{p} + kp - h_1\ddot{p}_g - h_2 f - h_3 u = 0 \tag{1}$$

where: ***m***, ***c*** and ***k*** are respective reduced weight, coefficients of damping and stiffness of the system. Variables $\ddot{p}_g = [\ddot{x}_g\ \ddot{y}_g\ \ddot{z}_g]^T$, $f = [f_1 \dots f_m]^T$ and $u = [u_1 \dots u_n]^T$ are vectors of forces acting on the system: vector of inertial forces, external forces and controls.

When writing a motion equation using generalized modal coordinates, it is achieved:

$$p = \Phi \cdot q \tag{2}$$

where: Φ is a modal matrix, while $q = [x\ y\ z\ \theta_x\ \ \theta_x\ \theta_x]^T$ is a generalized coordinate vector.

Equation (1) can be represented as:

$$M\ddot{q} + C\dot{q} + Kp = H_1\ddot{p}_g + H_2 f + H_3 u \tag{3}$$

where:

$$M = \Phi^T m\Phi,\quad C = \Phi^T c\Phi,\quad K = \Phi^T m\Phi,\quad H_{k=1,3} = \Phi^T m_{k=1,3}$$

Vertical movement of the seat can be analyzed using a simplified model of one degree of freedom. The seat motion equation for a one-dimensional system (i = 1) takes the form:

$$m_i\ddot{q}_i + c_i\dot{q}_i + k_1 q_i = m_i\ddot{q}_{gi} + f_i + u_i \tag{4}$$

In the simplified seat suspension model (fig. 2), the elasticity force of the system derived from the pneumatic spring F_{as}, the damping force F_c of the hydraulic shock absorber, and the force of gravity of the load F_g are taken into consideration.

The movement of the seat suspension system is as follows:

$$m\ddot{x} = F_{as} + F_c + F_g \tag{5}$$

The bellows actuator used in compressed air system can be treated as a pneumatic spring. Cylinders of this type are used where it is important to obtain a high compression force at a relatively small stroke and where the return to the starting position is effected by external forces. The use of the bellows actuator as a drive requires certain conditions - the actuator must work under load and the travel limiter is required.

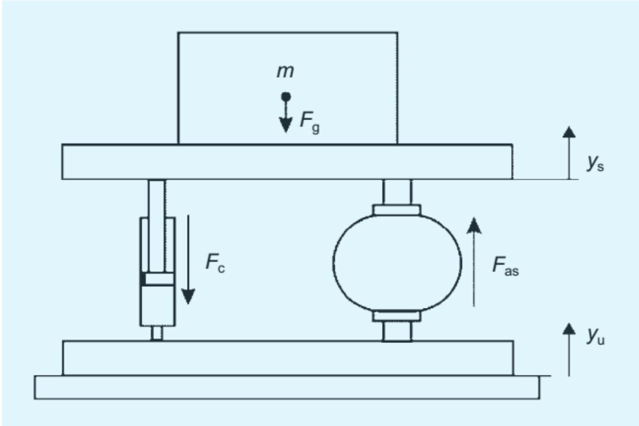


Fig. 2. Scheme of a simplified seat suspension model

If the powered bellows actuator is loaded with mass, a vibration system is obtained (fig. 2). F_{as} force from the pneumatic spring acting vertically on the vibration-insulated object is defined as the product of its effective surface A_{ef} and the relative pressure (p - p_0):

$$F_{as} = A_{ef} \cdot (p - p_0) \tag{6}$$

With the increase in pressure p, and thus the increase in volume V, the height h of the dwell increases and at the same time its effective surface A_{ef} decreases.

The rigidity k of pneumatic bellows is expressed by the ratio of the load F_g to the deflection h: $k = \mathrm{d}F_g / \mathrm{d}h$, and the dependence on the deflection h and pressure p are defined as follows [1]:

$$k(h,p) = p\frac{\mathrm{d}A_{ef}}{\mathrm{d}h} + \frac{A_{ef}^2}{C_g(p)} \tag{7}$$

where: C_g - gas volume of bellows, $C_g = V/K_g$, K_g - modulus of air elasticity, $K_g = \kappa p$, $\kappa = 1{,}4$ - adiabatic process exponent.

If the volume change of the airbag is:

$$V = V_0 - \Delta V = V_0 - A_{ef} \cdot h \tag{8}$$

and $V_0 = A_{ef} h_0$, then the rigidity of the bellows is defined as:

$$k(h) = \frac{\kappa \cdot p_0 \cdot A_{ef}^2}{V_0}\left(\frac{V_0}{V_0 - A_{ef} \cdot h}\right)^{\kappa+1} = \frac{\kappa \cdot p_0 \cdot A_{ef}}{h_0}\left(\frac{V_0}{h_0 - h}\right)^{\kappa+1} \tag{9}$$

where: V_0, p_0 - volume and pressure in the initial state.

As a consequence, the bellows force acting on the suspension system is defined as follows:

$$F_{as}(h) = k(h)(h_0 - h) = \kappa \cdot p_0 \cdot A_{ef}\frac{h_0 - h}{h_0}\left(\frac{h_0}{h_0 - h}\right)^{\kappa+1} \tag{10}$$

The frequency f_0 of the own vibration is:

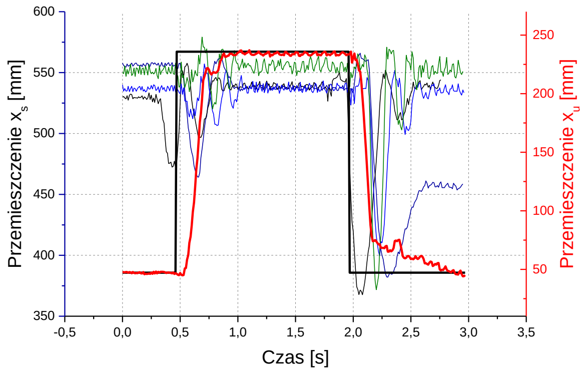
$$f_0 = \frac{1}{2\pi}\sqrt{\frac{\kappa \cdot p_0 \cdot g}{h_0 \cdot p}} \tag{11}$$

The coefficient of vibration amplitude transfer by the bellows actuator is defined as:

$$T_a(f) = \frac{1}{\left(\frac{f}{f_0}\right)^2 - 1} \tag{12}$$

The correct degree of vibration isolation of the bellows (pneumatic spring) is found at frequency $f = 1.4\ f_0$ [1]. Fig. 3 depicts the time course of the vibro-insulated seat mass for rectangular force. The tests were performed at set pressure p.

a)



b)

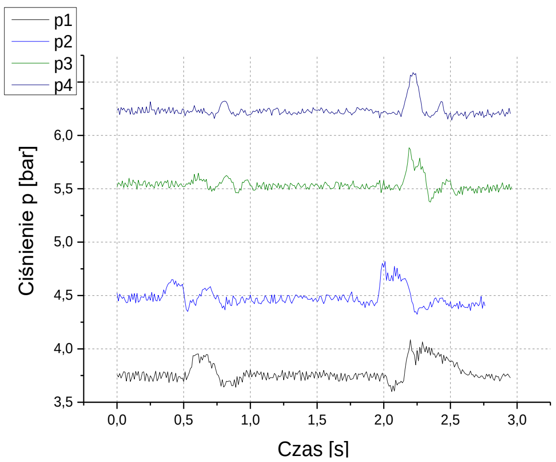
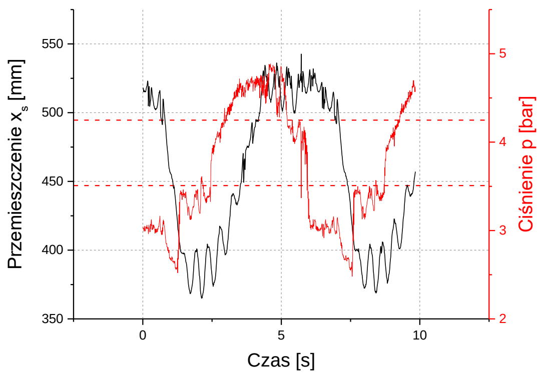


Fig.3. Timelines of: a) displacement of vibration mass x_s and constraint, b) pressure values p in bellows

Figure 4a shows the waveforms of vibro-insulated mass (72 kg) with a forced frequency of 2 Hz and a pressure change in the bellows. The best insulation properties of the bellows have reached the pressure p in the range of 4.44 ÷ 4.90 bar.



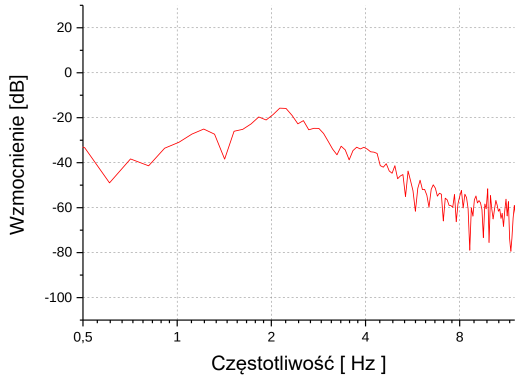


Fig. 4. Timing intervals of displacement of vibration mass under pressure change p (a) and frequency response of system vibration transfer (b)

Structure of the control system

The primary purpose of the seat control is to reduce the acceleration that the driver is subjected to. Suspension control is based on the measurements of the displacement of the working seat with the loading mass and relative displacement of the driver's seat (x_s) and floor (x_u) (fig. 5).

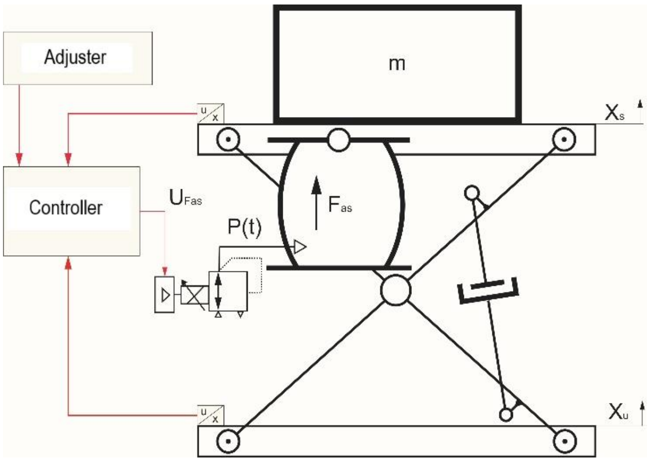


Fig. 5. Block diagram of semi-active seat suspension control

The role of the suspension adjustment system is to adjust the preset pressure in the bellows actuator so as to minimize acceleration of the seat. The initial value of the pressure in the bellows depends on the weight of the vibration isolation m. The force generated in the semi-automatic system depends not only on the command signal but also on the relative seat deflection (x_u - x_s). The acceleration limiting rule can be represented as:

$$\min\left|c\dot{\Delta} + k\Delta\right| \qquad (13)$$

where: $\Delta = x_u - x_s$ - current relative displacement of the suspension, c - damping factor, k - system elasticity coefficient.

The value of the control voltage of the valve is calculated according to the weight function:

$$u_{F\,as} = u_{zad} + w\cdot\left(k_1\cdot u_2(\dot{x}_u - \dot{x}_s) + k_2\cdot u_2(x_u - x_s)\right) \qquad (14)$$

where: $w=\left|\frac{x_u-x_s}{(x_u-x_s)_n}\right|$, $(x_u-x_s)_n$ - nominal value of the suspension system, k_1, k_2 - gain coefficients of the regulator.

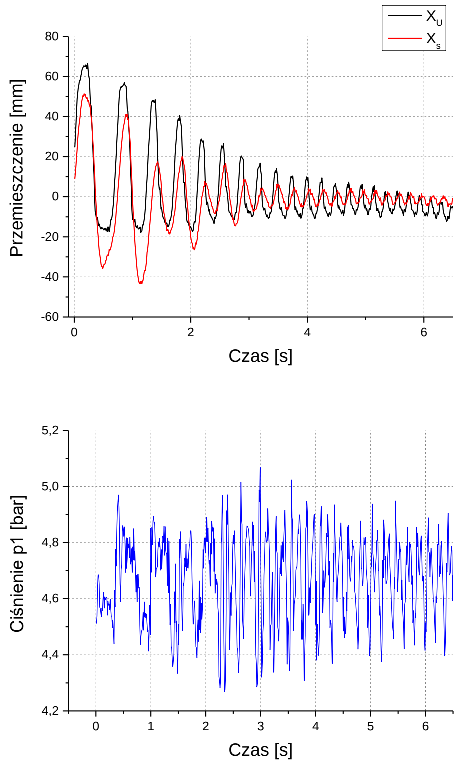


Fig. 6. Timelines of the movements of: a) vibro-insulated mass, b) adjustable pressure *p(t)*

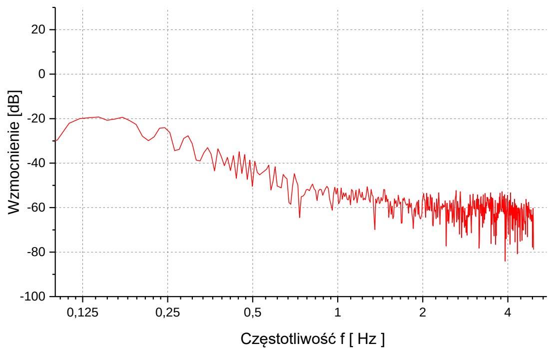


Fig. 7. Characteristics of vibration transfer of semi-active system

Figure 6a shows the time waveforms of vibro-insulated mass x_s and base x_u for forcing using sinusoidal amplitude 80 mm and frequency modulated in the range 0.1 ÷ 6.0 Hz. As a result of the semi-automatic regulation with adjustable stiffness of the pneumatic spring, the amplitude of the response of the vibro-insulated mass decreased. For a mass of 72 kg, bellows pressure of 4.6 bar and vibration frequency of 2.2 Hz, the vibration transmission decreased by approximately 22 dB. At higher frequencies (from 4 Hz), change in the damping properties of the pneumatic spring system is almost imperceptible.

Conclusions

The proposed seat vibration solution is based on the existing design of a passive suspension system commonly used in working machines. Such systems are less effective in reducing vibration in the low frequency range as vibration is amplified in the range of the vibration frequency (1 ÷ 2 Hz). Due to the developed control system, which controls the rigidity of the pneumatic spring, the seat vibration insulation is improved for a specified range of forced frequencies (up to 4 Hz). In this frequency range, the proposed system effectively reduces the relative displacement of the seat, which in the future, may result in improved comfort and safety.

REFERENCES

1. Dindorf R. „Elastyczne aktuatory pneumatyczne". *Monografie, Studia, Rozprawy*. M55. Kielce: Wydawnictwo Politechniki Świętokrzyskiej, 2013, page. 203.
2. Guglielmino E. i in. „*Semi-active Suspension Control: Improved Vehicle Ride and Road Friendliness*". London: Springer-Verlag, 2008.
3. Maciejewski I., Krzyżyński T. "Control design of semi-active seat suspension systems". *Journal of Theoretical and Applied Mechanics*. 49, 4 (2011): pages 1151–1168.
4. Mizuno T., Murashita M., Takasaki M., Ishino Y. "Pneumatic active vibration isolation systems using negative stiffness". *Trans. SICE*. 41, 8 (2005): pages 676–684.
5. Nader M. „*Modelowanie i symulacja oddziaływania drgań pojazdów na organizm człowieka*". Warszawa: Oficyna Wydawnicza Politechniki Warszawskiej, 2001.
6. Takosoglu J.E., Dindorf R., Laski P. "Rapid prototyping of fuzzy controller pneumatic servo-system". *International Journal of Advanced Manufacturing Technology*. 40, 3–4 (2009): pages 349–361.
7. Tora G. "Synthesis of the active cab suspension mechanism". *Advanced Materials in Machine Design*. Trans Tech Publications Ltd. 542 (2013): pages 219–231.
8. Woś P., Dindorf R. "Aktywny układ zawieszenia siedziska maszyny roboczej". *Logistyka.* 6 (2014): pages 11355–11360.
9. Woś P., Dindorf R., Mazur S. „Układ sterowania serwonapędem elektropneumatycznym przy zastosowaniu laserowej metody pomiaru przemieszczenia". *Pomiary Automatyka Kontrola*. 6 (2013): pages 574–577.