Number 5, Volume VI, December 2011

ON THE PROBLEM OF HEIGHT ADJUSTMENT OF THE ACTIVELY CONTROLLED DRIVER‘S SEAT VIBRATION ISOLATION STAND

Libor Kupka[1]

Summary: Every driver's seat vibration isolation stand is supplied with height adjustment. In the case of a standard seat vibration isolation stand (passively suspended), such adjustment is realized by the position mechanical controller. Any change in seat height results also in the change of the position of the working point of the pneumatic spring and the reduced mass or reduced inertia moment of the conducting mechanism are changed as well. This further results in the change of its natural frequency, transmissibility characteristics etc. These dependencies are analysed in the paper using an actively controlled driver's seat vibration isolation stand with scissor mechanism.

Key words: vibration isolation stand, height adjustment, conducting mechanism, pneumatic spring.

INTRODUCTION

If the height adjustment of a driver's seat vibration isolation stand is changed (using the mechanical or electronic position controller), the following quantities are also changed: volume and effective area of the pneumatic spring and reduced inertial moment of the conducting mechanism. As a result of it, the natural frequency is a function of the height adjustment.

This function was identified and analysed using driver's seat vibration isolation stand with a position controller (1) and (2). Actively controlled system with a sophisticated control algorithm (3), (4), (5) and (6) was analysed from the view of sensitivity to the height adjustment.

1 DYNAMICS OF THE SYSTEM

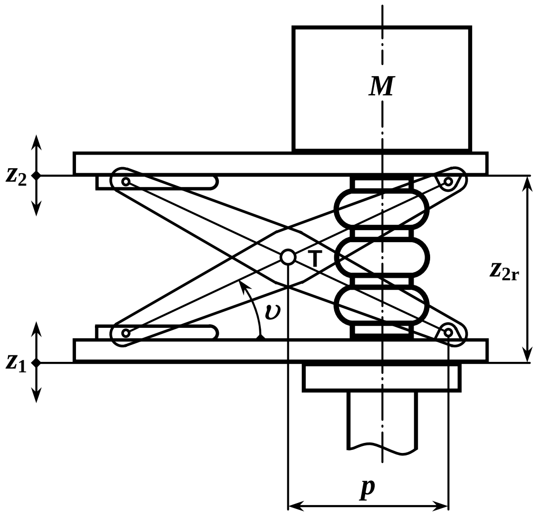
1.1 Basic equations

According to the facts that the used conducting mechanism is a scissors type and that the pneumatic spring suspension is realised between lower base and upper base of the scissors (see fig. 1) – there is a relation deflection the most natural general coordinate. We introduce a following notation:

- absolute deflection of lower base (kinematic excitation) $z_1(t)$,
- absolute deflection of upper base $z_2(t)$,
- relative deflection $z_{2r}(t)$,
- reduced mass of upper base (with reduced part of human body) M,

[1] Ing. Libor Kupka, Ph.D., University of Pardubice, Faculty of Electrical Engineering and Informatics, Department of Process Control, náměstí Čs. legií 565, 532 10 Pardubice, Tel.: +420 466 037 504, Fax: +420 466 036 241, E-mail: libor.kupka@upce.cz

- angle of arm $\upsilon(t)$,
- length of arm l_R,
- mass of scissors arm m_R,
- inertial moment of scissors arm I_R,
- distance of the point T from the point of arms attachment $p(t)$.



Source: Author

Fig. 1 – Scheme of the described vibration isolation stand

Following two equations were derived in (7) by means of Lagrangian mechanics on the basis of fig. 1. The kinetic energy of the described vibration isolation stand with scissor mechanism and pneumatic spring is

$$K = \frac{1}{2} M (\dot{z}_{2r} + \dot{z}_1)^2 + 4 \left\{ \frac{1}{2} m_R \left[\dot{p}^2 + \left(\frac{\dot{z}_{2r}}{2} + \dot{z}_1 \right)^2 \right] + \frac{1}{2} I_R \left(\frac{d\upsilon}{dz_{2r}} \right)^2 \dot{z}_{2r}^2 \right\} \quad (1)$$

and potential energy is

$$P = M\,g\,(z_{2r} + z_1) + 4\,m_R\,g \left(\frac{z_{2r}}{2} + z_1 \right) . \quad (2)$$

In the scissors conducting mechanism there is applied a three-wave pneumatic spring. According to the assumption that its effective area S_{ef} and its volume V_p depend only on its length (8) force of the spring is

$$F_p = S_{ef}(z_{2r})[p_2 - p_a] \ , \quad (3)$$

where p_2 is overpressure inside the spring and p_a is atmospheric pressure. For these reasons, we can also write

$$V_p = -\int_{h_0}^{h} S_{ef}(z_{2r})\, dz_{2r} + V_{p0} \, , \quad (4)$$

where V_{p0} is volume by $z_{2r}(t) = z_{2r}(0) = z_{2r0}$.

If we designate

$$z_{2r} = l_R \sin\upsilon \, , \quad \upsilon = \arcsin\frac{z_{2r}}{l_R} \, ,$$

the first and the second derivation of angle $\upsilon(t)$ are

$$\frac{\mathrm{d}\upsilon}{\mathrm{d}z_{2r}} = (l_R^2 - z_{2r}^2)^{-\frac{1}{2}}, \tag{5}$$

$$\frac{\mathrm{d}^2\upsilon}{\mathrm{d}z_{2r}} = z_{2r}\,(l_R^2 - z_{2r}^2)^{-\frac{3}{2}}. \tag{6}$$

Using general Lagrange equations of the second kind

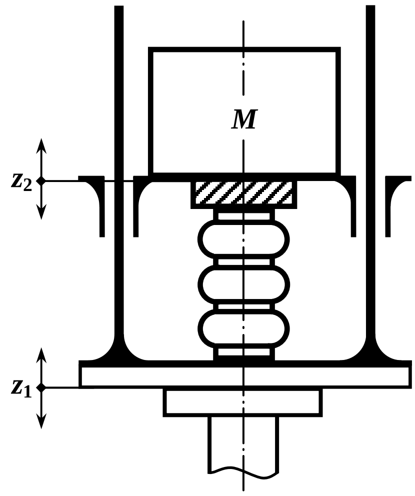
$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\partial K}{\partial \dot{z}_{2r}}\right) - \frac{\partial K}{\partial z_{2r}} + \frac{\partial P}{\partial z_{2r}} = F_p + F_{rp} + F_s \tag{7}$$

a nonlinear second order equation is obtained

$$\left[M + m_R + 4 I_R \left(\frac{\mathrm{d}\upsilon}{\mathrm{d}z_{2r}}\right)^2\right]\ddot{z}_{2r} + 4 I_R \frac{\mathrm{d}\upsilon}{\mathrm{d}z_{2r}} \frac{\mathrm{d}^2\upsilon}{\mathrm{d}z_{2r}^2}\dot{z}_{2r}^2 - S_{ef}(p_2 - p_a) - F_{rp} - F_s = \\ = (M + 2\,m_R)\,(g + \ddot{z}_1) \tag{8}$$

where F_{rp} is a general force of the passive resistance.

We denote that the derived equation (8) can be used also for a mechanism on the fig. 2 if $m_R = 0$ and $I_R = 0$.



Source: Author

Fig. 2 – Vibration isolation stand with straight-line conducting mechanism

The conducting mechanism is provided with rubber stop members. The threshold force can be described by

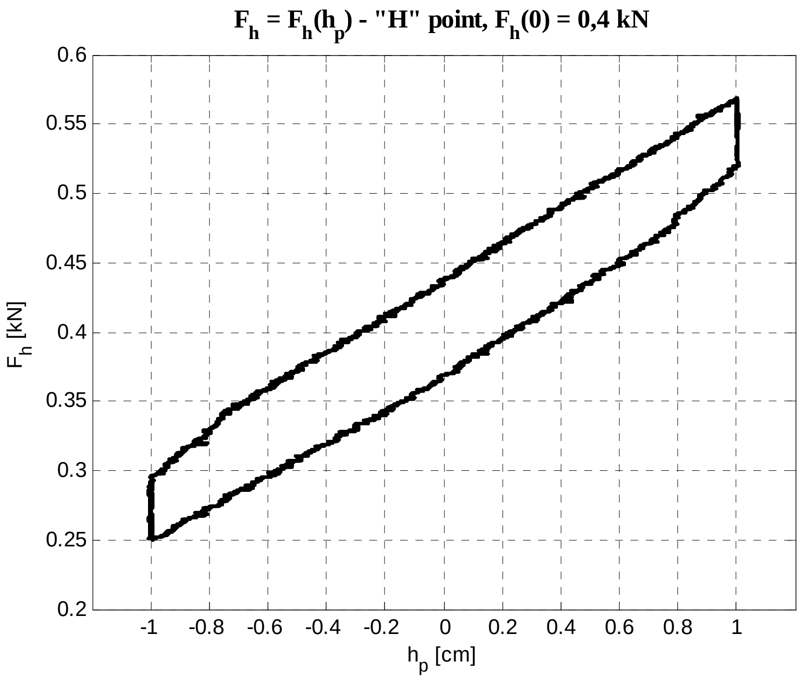
$$F_{rd} = k_F \left(\frac{z_{2r}}{z_{2r\max}}\right)^n, \tag{9}$$

where n is odd number and $n >> 0$.

The force of passive resistance F_{rp} is identified from static characteristic of a loaded conducting mechanism

$$F_{rp} = -\varepsilon \,\mathrm{sign}\,\dot{z}_{2r}, \tag{10}$$

where the constant ε is equal a half width of hysteresis (see fig. 3).



Source: (9)

Fig. 3 – Static characteristic of a loaded conducting mechanism

1.2 Air pressure inside the pneumatic spring

By the assumption of the validity of adiabatic action in a pneumatic spring is

$$p_2 \left(\frac{V_\mathrm{p}}{m_\mathrm{v}} \right)^{\kappa} = \mathrm{const} \ , \tag{11}$$

where m_v is mass of air in the pneumatic spring. The derivative of (11) gives

$$\frac{\mathrm{d}p_2}{\mathrm{d}t} = p_2 \ \kappa \left(\frac{1}{m_\mathrm{v}} \frac{\mathrm{d}m_\mathrm{v}}{\mathrm{d}t} - \frac{1}{V_\mathrm{p}} \frac{\mathrm{d}V_\mathrm{p}}{\mathrm{d}t} \right), \tag{12}$$

$Q_\mathrm{m} = \mathrm{d}m_\mathrm{v}/\mathrm{d}t$ denote mass flow into the pneumatic spring.

1.3 Air flow through the servo valve

The relation for pneumatic spring filling (see fig. 4b, control voltage of valve $u_1 \geq 0$) from a tank with constant pressure $p_1 = 0.87$ MPa

$$Q_\mathrm{m}(t) = u_1(t - \tau_\mathrm{d}) k_\mathrm{v1} \sqrt{p_1 [p_1 - p_2(t)]} \tag{13}$$

and for air discharge from spring (control voltage of valve $u_1 < 0$) into the atmosphere

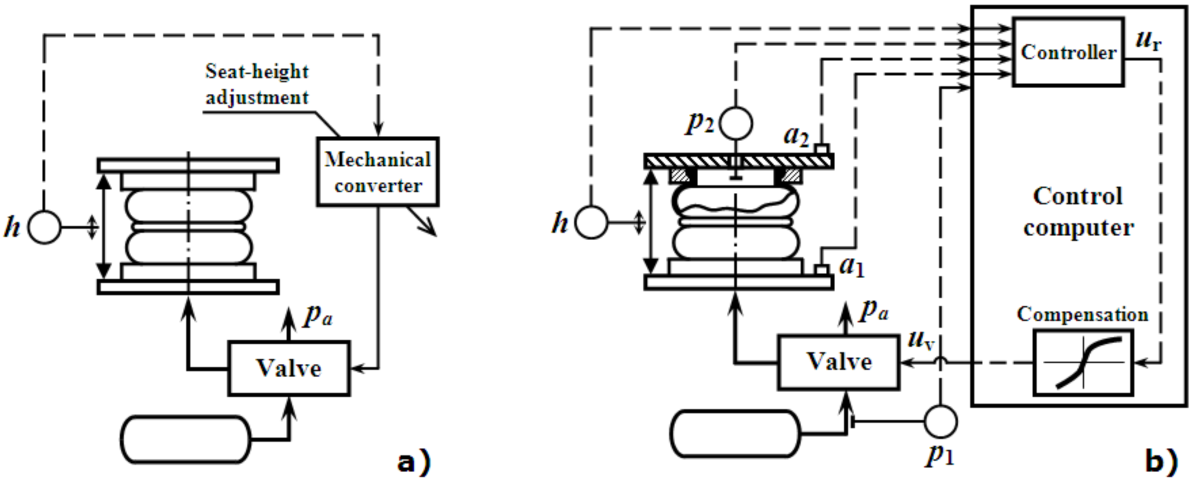
$$Q_\mathrm{m}(t) = u_1(t - \tau_\mathrm{d}) k_\mathrm{v2} \sqrt{p_2(t) [p_2(t) - p_\mathrm{a}]} \tag{14}$$

where τ_d is transport delay and k_v1 and k_v2 are experimentally determined flow coefficients (9).

The nonlinear equations (13) and (14) were linearized with compensation function inside the control computer. Equations (8) and (12) were converted into the state space form and substituted by the system of four first order equations (5).

2 DRIVER'S SEATS HEIGHT ADJUSTMENT

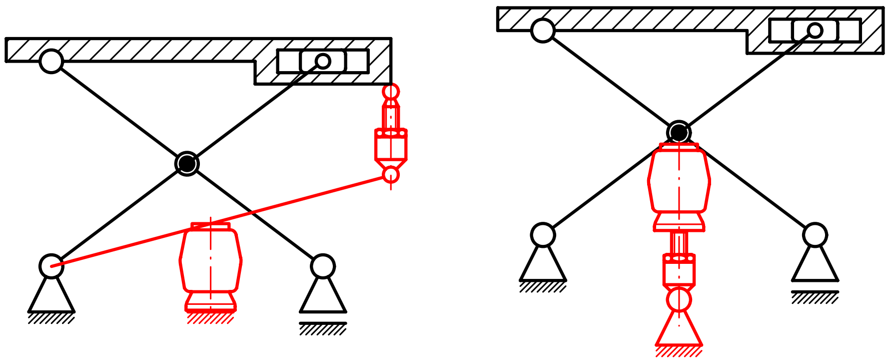
The both basics principles of the passive and the active seats height adjustment are shown in fig. 4. In the case of a passively suspended driver's seat vibration isolation stand, such adjustment is realized by the position mechanical controller (see fig. 4a). In the case of actively suspended vibration isolation stand height adjustment system is realized by the sophisticated control algorithm (see fig. 4b).



Source: (9)

Fig. 4 – The principles of the passive and the active seats height adjustment

There are many ways to design a mechanical height adjustment of the passively suspended driver's seat vibration isolation stands. For example, let us mention the systems described in (1) and (2). In the fig. 4 we can see two possibilities of a separation of height adjustment and seat suspension.



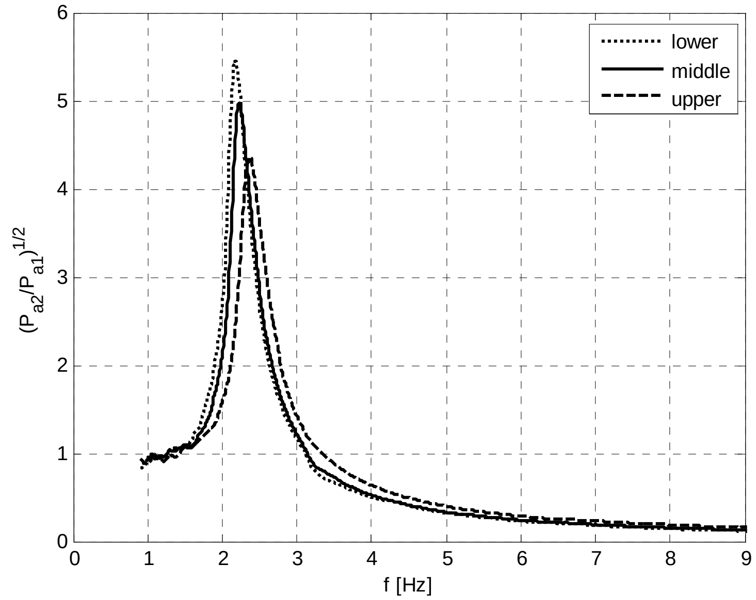
Source: Author

Fig. 5 – Some possibilities of a separation of height adjustment and suspension

3 LABORATORY RESULTS

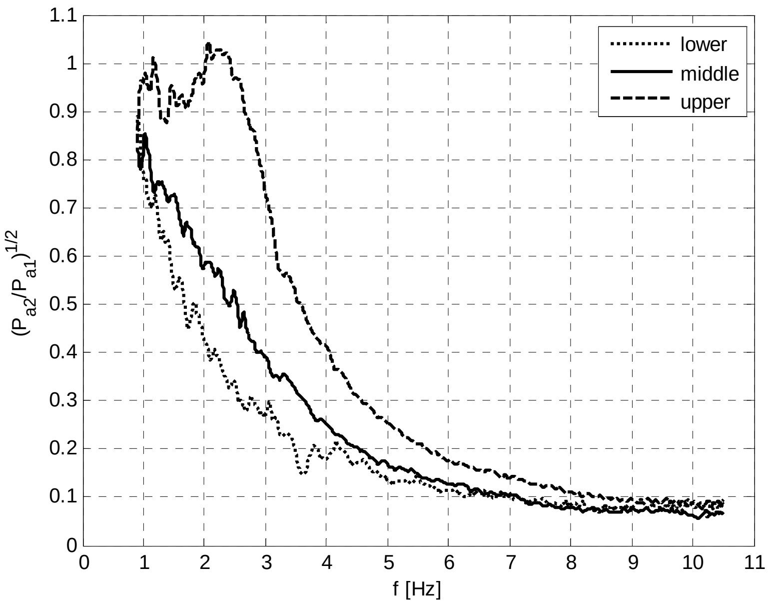
The developed control algorithm for active vibration isolation of driver's seat was tuned for middle equilibrium height adjustment. The transmissibility characteristics of the controlled system also for lower and upper equilibrium height adjustments are presented on fig. 7 and 8.

Laboratory testing of driver's seat vibration isolation stand was done with seat base vertical movements, with artificial signals (harmonic, "CHIRP") or, mainly, with signals gained by road measurements (many gained on TATRA proving ground). Seat was loaded by simple weights or by a 2-DOF dummy. Test results were however for simple weights and dummies very similar and their minor differences are not discussed here. Following results were gained with the dummy. Vibration isolation stand was excited by a signal CHIRP (see fig. 6 and fig. 7) and quasi-stochastic signal TATRA (see fig. 8).



Source: Author

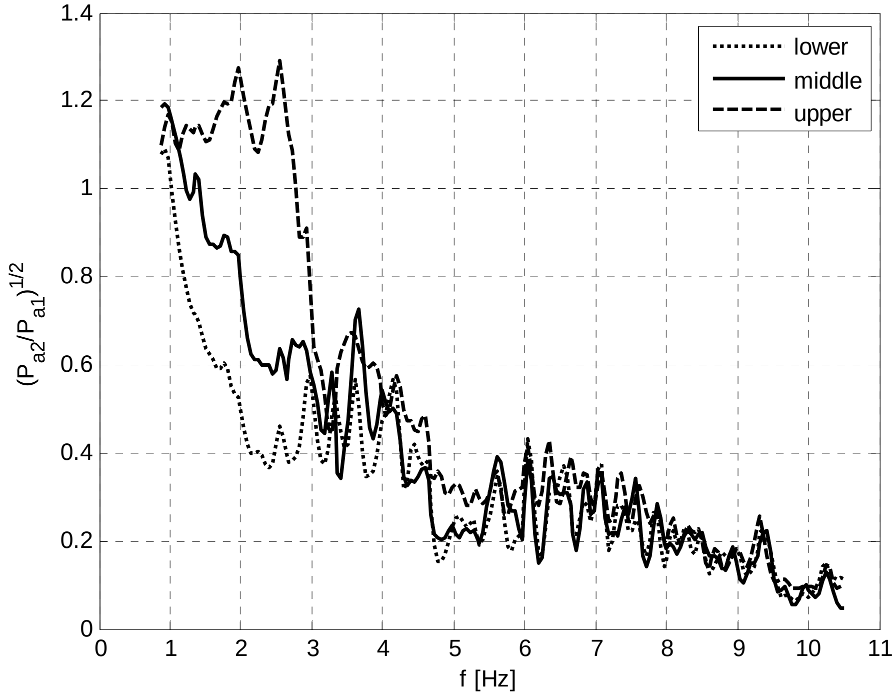
Fig. 6 – Acceleration amplitude transmisibility of the system without control (CHIRP signal was used for excitation)



Source: Author

Fig. 7 – Acceleration amplitude transmisibility of the system with control (CHIRP signal was used for excitation)

P_{a1} is the power spectral density (PSD) of the seat base acceleration a_1 and P_{a2} is the PSD of the seat cushion acceleration a_2.



Source: Author

Fig. 8 – Acceleration amplitude transmisibility of the system with control (TATRA signal was used for excitation)

CONSLUSIONS

In results of measurements are presented the acceleration amplitude transmisibility characteristics of the driver's seat without and with control. All these characteristics are measured in cases of three equilibrium height adjustments: lower, middle and upper. The developed control algorithm for active vibration isolation of driver seat was tuned for middle height adjustment. From the viewpoint of vibration isolation it is possible to say that in case of lower height adjustment is the frequency characteristic better than in case of middle height adjustment. The vibration isolation with mentioned feedback algorithm in case of upper height adjustment could be used as well, it is enough satisfactory.

ACKNOWLEDGMENT

This paper has been supported by the Ministry of Education of the Czech Republic under projects – research plans MSM 0021627505 "Theory of transport systems"

REFERENCES

(1) ŠKLÍBA, J. O vlivu umístění pneumatické pružiny v odpruženém podstavci sedačky. In *Proceedings of National Conference Engineering Mechanics '95*. Praha: ITAM CAS, 1995.

(2) ŠKLÍBA, J.; LOUDOVÁ, J. On a Next Possibility of the Accuracy of a Driver Seat Model. In *Proceedings of National Conference Engineering Mechanics 2000*. Praha: ITAM CAS, 2000.

(3) APETAUR, M.; BUCHTA, J.; JANEČEK, B.; KUPKA, L.; ŠKLÍBA, J. Actively Controlled Air-suspended Driver's Seat – From Laboratory to Tatra Proving Ground. *Perner's Contacts*, 2010, Volume 5, No. IV, p. 7–19. ISSN: 1801-674X.

(4) KUPKA, L. Vývoj aktivně řízené sedačky řidiče. In *Sborník příspěvků konference Dopravní systémy 2009*. Pardubice: Institut Jana Pernera, o.p.s., 2009, p. 227–239, ISBN 978-80-86530-63-5.

(5) BARBORA, J.; BUCHTA, J.; JANEČEK, B.; LUFINKA, A.; MAREK, M.; ŠKLÍBA, J.; APETAUR, M.; KUPKA, L. Research of Actively Controlled Air-suspended Seat for Vehicles. *JVE Journal of Vibroengineering*, 9, 2009, Volume 11, Issue 3, Publication 476, p. 421 – 433, ISSN 1392-8716.

(6) KUPKA, L.; JANEČEK, B.; ŠKLÍBA, J. Laboratory Verification of the Active Vibration Isolation of the Driver Seat. *Recent Advances in Mechatronics.* Berlin – Heidelberg – New York: Springer, 2007, p. 453–457, ISBN 978-3-540-73955-5.

(7) KUPKA, L.: Nelineární model vibroizolačního podstavce sedadla řidiče s nůžkovým vodicím mechanismem a jeho linearizace. *Perner's Contacts*, 2011, Volume 6, No. IV, p. 188–204. ISSN: 1801-674X.

(8) KREJČÍŘ, O. *Pneumatická vibroizolace*. [Dissertation thesis.] Liberec: VŠST, 1986.

(9) KUPKA, L.: *Active Vibration Isolation System of Driver Seat*. [Dissertation thesis.] Liberec, 2008. Technical University of Liberec. Faculty of Mechatronics, Informatics and Interdisciplinary Studies.