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ACTIVE VIBRATION CONTROL SYSTEMS FOR DRIVER'S SEATS – FROM BASIC RESEARCH TO FIRST COMMERCIAL APPLICATIONS

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Summary: *The research into various types of active vibration control systems in driver's seats was pursued at IMMM SAS since early sixties. The contribution describes the research into electro-hydraulic and into various types of electro-pneumatic active vibration control systems for driver's seats, specifically those with fully electronic control, making the use of a conventional hydraulic damper obsolete. Newer world-wide research efforts are briefly reported. Hints on the start of commercial exploitation of the described control principles are given too.*

1. Introduction

The research on application of active vibration control systems (AVCS) in driver's seats goes back to the last quarter of the previous century. Also at the IMMM SAS, under the auspices of late Prof. Ing. Igor Ballo, DrSc., the research into various types of AVCS was pursued since early sixties. The contribution describes this research and hints on current start of commercial exploitation.

In the simplest way a vibration control system (VCS) is usually designed as a parallel spring and damper combination, operating in a guiding mechanism, tuned by suitable selection of parameters in such a way to mitigate vibrations in required/assumed frequency band. On the other hand this VCS has to be stiff at near zero frequencies to withstand static loading forces (e.g. due to load weight, etc.). Such VCS acts as a unidirectional vibration energy absorbing system, i.e. vibrational energy on the output (at the isolated mass) is less than on the system input (drawn from the excitation). The system can be described in the time domain by governing equation of motion of a single degree-of-freedom (DOF) oscillatory system (neglecting ever present friction), or, subject to harmonic excitation and neglecting the transient response, by the respective Frequency Response Function (FRF).

A single DOF (SDOF) passive VCS either suppresses well steady state vibrations with frequency above its natural frequency, but then is prone to long lasting excessive overshoots around damped natural frequency or, in contrary, is not attenuating vibrations well; however, amplification of vibration with frequency around its damped natural frequency is low. For random excitation with occasional shock excitation, as common in land transport means, it is very difficult to design a SDOF passive VCS that satisfies both conflicting and mutually exclusive requirements. Moreover, real components are by no means linear and without

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friction or hysteresis (especially the common hydraulic damper) and this complicates the situation further. A remedy to some of these problems may be an AVCS, which can be designed in such a way as to mitigate the vibrations in the frequency band of interest without amplifying vibrations around its natural frequency.

For illustration, the courses of FRF modules $|\mathbf{G}(\Omega)|$ of two sets of VCS – a passive one (dashed lines) and an active one of the sky-hook type (bold), having the same amplification at the normalized natural frequency $\Omega = \omega/\omega_0 = 1$, are depicted in Fig. 1. Note the better attenuation properties of the active one for $\Omega \geq \sqrt{2}$ than that one of a passive VCS, as well the course of an ideal sky-hook type AVCS with no amplification at the resonance (the lowest bold curve). It will be shown, that it is possible to reach similar FRF course with an electro-pneumatic AVCS, whose dynamic properties were really measured.

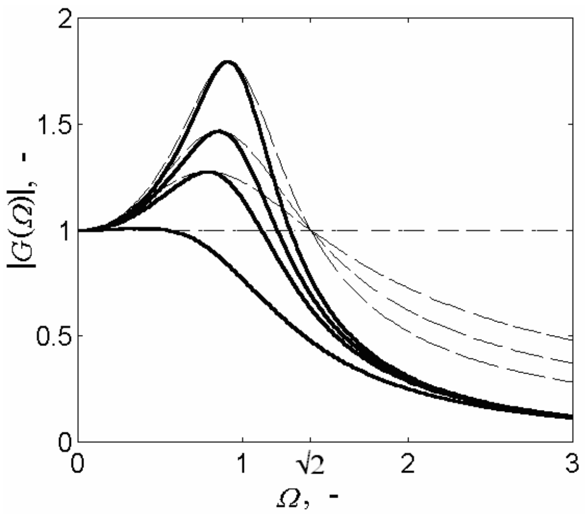


Fig. 1: FRF modulus of a passive VCS (dashed) and of an AVCS (bold) (Stein, 2003)

2. Electro-hydraulic active control system for driver's seats

2.1. Overview

First theoretical and experimental research of AVCS was reported in the USA (Ruzicka, 1969), Germany (Helms, 1974; Kauss, 1981), in the former Soviet Union, e.g. Frolov et al. (1977) and Balagula & Gajcgori (1977) and latter in Poland (Nizol et al., 1990); all using hydraulic cylinders and servo valves. Also IMMM SAS devoted much of its research capacity to the research of electro-hydraulic AVCS, as reported by Ballo et al. (1984), Ballo (1993) and Stein & Ballo (1991).

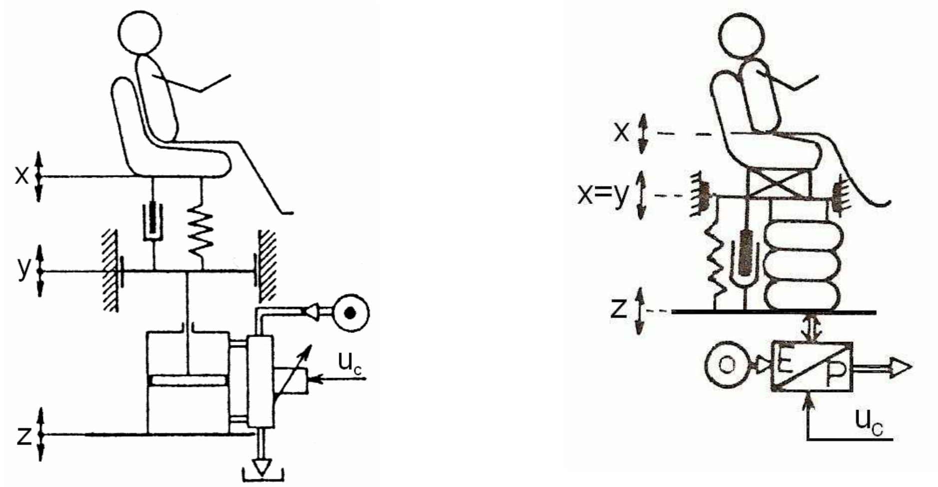
Essentially two operating principles can be used (Stein & Ballo, 1991) – Fig. 2:

i. The series one, when a standard driver's seat is mounted on a vibration isolation platform, driven by a hydraulic cylinder, which acts predominantly in the vertical (z–axis) direction.

ii. A parallel one, when the actuator (hydraulic cylinder or air spring) is mounted in parallel to a conventional spring – damper combination.

The linear hydraulic actuator can be controlled either by a servo valve or by a proportional valve. Two control principles can be essentially used:

a. The feed back principle, often denoted as the sky-hook principle, introduced by Karnopp (1983), based on the well known feed-back control principles.
b. The feed-forward principle (error variable measurement), introduced by Ballo (1976) and by Frollov et al. (1977).



(a) Electro-hydraulic series combination (b) Electro-pneumatic parallel combination

Fig. 2: Active vibration control system operating principles (Stein & Ballo, 1991)

2.2. Experimental setup description

The system developed at IMMM SAS was essentially a series one: a standard driver's seat was mounted on a vibration-insulating platform, driven by a hydraulic cylinder - Fig. 3, after (Stein & Ballo, 1991). For experimental purposes a two-channel electro-hydraulic testing machine of type EDYZ 4-2M, made by the Inova Company, Prague, was used; both channels operating as positional servo-systems with integrated displacement transducers and hydraulic amplifiers. One channel was used to generate the sought vibratory excitation (harmonic or random) in the vertical direction; the second one was part of the AVCS. The AVCS was mounted on a 2 m long rigid arm pivoted on one side. The other side, driven by the vertically mounted servo-hydraulic cylinder of the first channel, performed a nearly vertical movement with displacement of up to ± 50 mm in the frequency range 0 to 8 Hz. Both cylinders were fed from a hydraulic supply at pressure of up to 100 bars and feed rate of 80 l/min. The hydraulic actuator of the AVCS was mounted nearly horizontally. An ingenious linear mechanism transformed the horizontal movement to a vertical one of a platform, guided by rollers in a guideway to prevent any lateral movement. On this platform a standard driver's seat was mounted, either a GDR made ZT 300 mechanical tractor seat or Czech made Karosa 281.0 type pneumatic seat.

The excitation sub-system was driven either by a harmonic signal from a low frequency function generator or by a narrow band random signal. This signal was generated from stationary ergodic signal with normal distribution, generated by the GENAP III random signal generator, which was then filtered by a series combination of a high-pass and low-pass filters, as defined in then valid ISO 7096 (1982) standard for a selected machine class (see Fig. 3). The output signal, corresponding to the vertical acceleration encountered in the respective earth moving machine class was further transformed by an integrator network to a displacement signal to drive the positional servo-system.

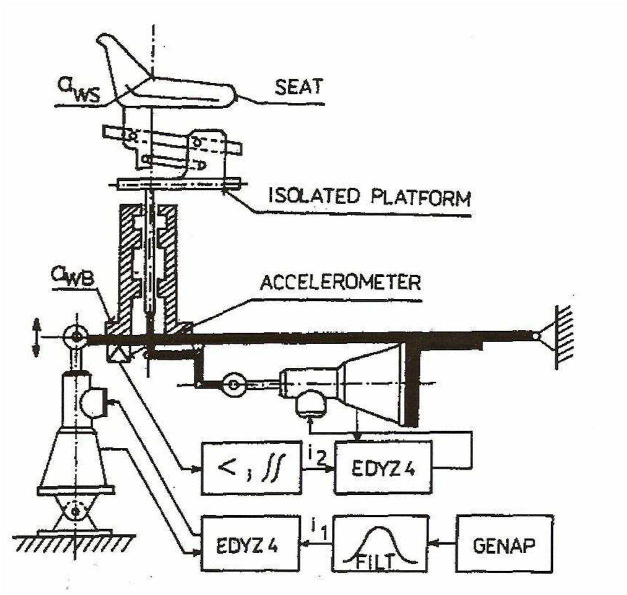


Fig. 3: The vibration simulator and the electro-hydraulic AVCS (Stein & Ballo, 1991)

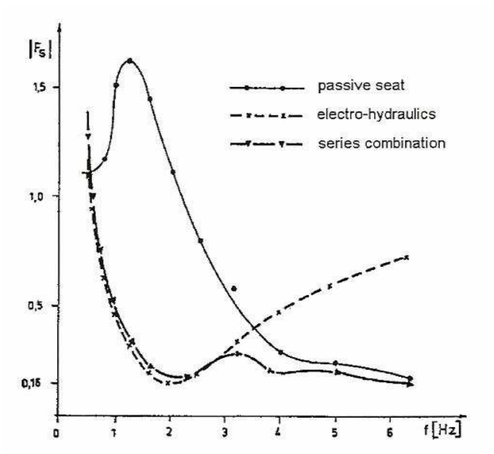
The test stand did not comply with safety requirements for tests with human subjects. Hence, a dead mass loading by bags filled with lead shots of various masses was used instead, as customary in seat dynamics research. The loading was by 75 kg, as suggested by ISO 7096 (1982), and used in the measurements reported in following.

An electronic controller governed the active system control loop. The positional servo-loop was furnished by the EDYZ system; hence only the feed forward control signal, corresponding to the seat base (floor excitation) in the vertical direction was required. The respective sensor (piezoelectric accelerometer) was mounted near the guideway of the isolated platform. The amplified signal, corresponding to the base vertical acceleration was then transformed by a double integrator to appropriate vertical displacement signal to drive the second positional servo system.

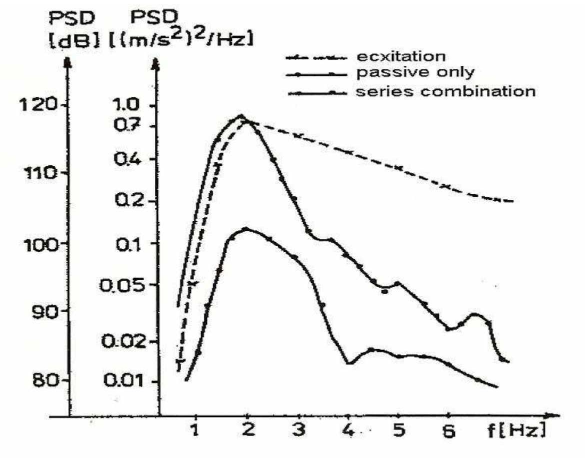
The electronic controller was first set up on MEDA 41TC type analogue controller, which facilitated a simple and rapid variation of the controller structure and parameters setting. When the controller structure and parameters were optimised the controller was build from standard operational amplifiers in a form of a bread-board (Stein, 1986).

2.3. Selected experimental results

The experiments were performed in two stages. In the first stage the mechanical tractor seat ZT 300 was used and the AVCS was tested using excitation for 3rd class of earth moving machinery (wheeled loader or tractor). The frequency response function (FRF) modulus $|F_s|$ was evaluated when the AVCS was subjected to harmonic excitation of constant acceleration amplitude value in the frequency range 0 to 8 Hz (Fig. 4a). Note the course of the passive seat suspension system only, whose resonant frequency is around 1.3 Hz and modulus value at this frequency is approx. 1.62. Note the performance of the series combination of the AVCS and of the passive seat suspension system, which has a minimum around 2.0 Hz, where the excitation has most of the vibratory energy. The AVCS U-form shaped FRF modulus conforms to simulation results (Stein et al., 1979), made on MEDA 41TC analogue computer. The deterioration at low frequency is due to the controller's integrators performance, whereas the deterioration at higher frequencies is due to poor dynamics of the electro-hydraulic positional system. However, the combined "slow-acting" system performs very well: when subjected to stationary random excitation corresponding to the 3rd machine class after ISO 7096 (1982) standard the performance, expressed as the so called SEAT factor (a commonly used metrics in the seating dynamics research) improved from the value of 53 % for the passive seat suspension to the value of 15 %♦. The improvement is approx. 3.5-fold.



(a) Acceleration frequency responses



(b) Vertical acceleration PSD curves

Fig. 4: Acceleration FRF modulus and the PSD courses (Stein & Ballo, 1991)

In the second stage the driver's seat of Karosa 281.0 type was used and the bread-board controller was used. The seat was subjected to class 2 excitation (tractor scraper with suspended front axle), which is more severe excitation as that one used in the previous case. The resonance frequency was 1.1 Hz and the vibration mitigation properties expressed as the SEAT factor was 65 % for the passive seat and 22 %, respectively, for the combined system, i.e. improvement is 3-fold. For illustration the courses of acceleration Power Spectral Densities (PSD) are depicted in Fig. 4b. Note the improvement around the excitation maximum of 2.0 Hz, where the acceleration affecting the seated driver is 2.8-times less severe than when the AVCS is switched off. The passive seat increases acceleration transmission a little, due to its resonance properties.

2.4. Further development

The above encouraging results were presented in mid eighties to the representatives of the Czechoslovak industry, but at that time no much attention was paid to this rather complicated and power consuming system, whose practical use seemed not viable on cost reasons. Moreover, in those times in the machine industry there was deep mistrust in using electronics at all. Applied research could not be initiated and further basic research was abandoned.

♦ The SEAT factor is defined as the ratio of weighted root-means-square acceleration value, measured under the seated driver, to the same characteristics, measured at the seat base, expressed, for example, in per-cent. The weighting factor accounts for the frequency dependent sensitivity of human body to vertical direction vibration. The less the SEAT factor value the better the seat vibration mitigation properties. Suspension seats with SEAT factor of excess of 100 % (i.e. amplifying vibration, compared to no seat at all) are still in use.

In 2001 the John Deere Company of USA introduced for their large wheeled agricultural tractors/towers series 8020 and 9020 an electro-hydraulic active seat (John Deere Active Seat™[•]), made in cooperation with Sears seating[*]. It is essentially based on the Karnopp's feed-back principle combined with a positional servo-loop. A positional sensor measures the seat stroke and an accelerometer measures the vertical acceleration the seat top is exposed to. The system utilises a standard Sears's seat scissors type mechanism with an air spring; however the hydraulic damper is replaced by a hydraulic actuator, supplied by hydraulic fluid from a control valve assembly, controlled by a digital controller (Dufner & Schick, 2002). Hence, it is a parallel AVCS system of the feed-back type. The improvement due to the use of this AVCS, according to information published in Dufner & Schick (2002) is following: the SEAT factor of the passive seat suspension system is approx. 81 %, whereas for the parallel AVCS system it is improved to 27 %. The improvement is approx. 3-fold, i.e. roughly the same as in laboratory experiments made at IMMM SAS. According to the authors: "field data shows that the John Deere Active Seat™ can allow for increased field speeds of 15 % to 20 %. Therefore farmer's and machine contractor's essential needs for increasing overall work output with lover operating costs can be met".

Klooster of the USA (Klooster et al., 2003) undertook a major improvement of the above mentioned device expanding its action from SDOF action in the vertical direction (z-axis) to a 3 DOF device: movement in the vertical direction (z-axis), in the fore-and-aft direction (x-axis) and the pitch angle are controlled. This seat, named "Hyper-active seat" utilizes a parallel manipulator design, which allows for the independent control of motion in the x–z plane (Fig. 5a). The seat is powered using three hydraulic actuators, the same as employed in the John Deere Active Seat™, which are controlled using proportional flow valves, driven via a multipurpose data acquisition card, type National Instrument 6052E from a digital controller developed using Matlab® programming environment tools. Two single axis acceleration sensors were used, mounted on the seat base and at the upper platen, below the seat cushion (in Fig. 5a marked in red).

Klooster evaluated the Hyper-Active seat using step and frequency response in multiple degrees of freedom. He concluded: "The performance of the vibration cancellation controller was evaluated against an uncontrolled Hyper-Active seat. The controller showed excellent results in the 1 to 7 Hz range. The displacement of the amplitude was also decreased about 40 % over the uncontrolled seat. The RMS average acceleration was reduced, thereby increasing the potential of the operator to work longer. These experiments have shown that the Hyper-Active seat was able to significantly reduce vibrations in the harmful frequency range. It can be concluded the Hyper-Active seat has the potential to suppress vibrations that are harmful to the operator". The seat is capable to rotate in the x-z plane by some 20 degrees in less than 50 ms enabling to avoid grave injuries in case of a frontal crash. No tests with random excitation were made so performance using the SEAT factor metrics was not possible.

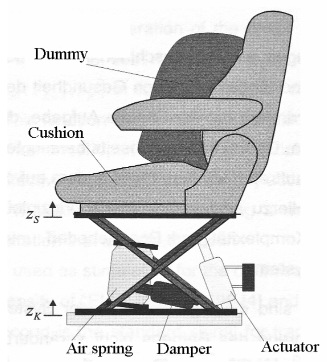
Another type of active electro-hydraulic seat was recently reported (Kühnlein, 2007). He tested a parallel electro-hydraulic AVCS, which is designed as an "add-on" to the mechanism of a conventional suspended seat with air spring and hydraulic damper (Fig. 5b). The hydraulic actuator functioning as an additional force generator is controlled using both

[•] See: http://salesmanual.deere.com/sales/salesmanual/en_NA/tractors/2007/feature/operator_station_and_controls/8030/8030_wheel_op_station_dlx_cab_wactiveseat_0995.html

[*] See: http://www.searsseating.com/tech_innovate.cfm

the feed-back and feed-forward principles, analogical to that one described in Stein (1997). An extended non-linear controller model was developed, catering for driver's movement and limiting the seat upper part vertical acceleration to less than gravity acceleration. A laboratory dummy was tested with standard signals for example for the earth moving machinery classes EM 3 (wheel loader), described in the new version of ISO 7096 (2000), as well as for class AG2 of agricultural tractors after ISO 5007 (1990). The results were compared to those obtained with a conventional seat. The SEAT factor was improved from 60 % to 38 % for the AG2 class and from 80 % to 40 % (i.e. two-fold) for the class EM 3 machine. The seat relative displacement (stroke) remained unchanged in comparison to the conventional suspended seat for the AG 2 class, whereas it slightly improved for the class EM 3 excitation.

(a) Planar electro-hydraulic AVCS (Klooster, 2003)



(b) Vertical electro-hydraulic AVCS (Kühnlein, 2007)

Fig. 5: Newer electro-hydraulic AVCS demonstrator seat systems

3. Electro-pneumatic active vibration control systems for driver's seats

Electro-pneumatic AVCS were much treated at IMMM SAS too, as driver's seats with air springs are commonly used. Ballo (1988) described the proposed principle of operation, introducing the feed-forward control (vibration compensation) approach. Cho and Hedrick (1985) reported an analogical application in railway engineering; however using the sky-hook principle. A lot of research of electro-pneumatic AVCS was undertaken in Poland, too, e.g. by Golas & Kowal (1986), Engel & Kowal (1990a, 1990b), Kisielewski (1991) and Michalowski (1994). The main problem was the physical complexity of the flow properties of the compressed air (Cho & Hedrick, 1985), which had to be theoretically mastered first - see Gajarsky (1984). Based on thorough Gajarsky's theoretical work a laboratory demonstrator seat was developed at the IMMM SAS.

3.1 The demonstrator seat system

The schematic lay out of the full-scale driver's seat demonstrator system with active electro-pneumatic suspension is depicted in Figure 6. The pneumatic spring **2** acts via the lever **4** on the upper part **5** of a shear type guiding mechanism **3**, which is supporting the isolated mass (isolated body or seated operator) of mass M, situated in the soft seat cushion **6**; whose vertical acceleration $\ddot{x}$ has to be controlled. The structure is mounted on the base **1** that is excited mostly in the vertical direction by vibration with acceleration $\ddot{z}$. A relief steel spring **7** is situated between the base **1** and the upper part **5** to hold the mass M at the required static height h_0, approximately in the middle of the possible travel of the shear type mechanism **3**. Note, that no hydraulic damper is used. The active part is formed by the electro-pneumatic valve (E/P transducer) **8**, source of compressed air and outlet to the atmosphere.

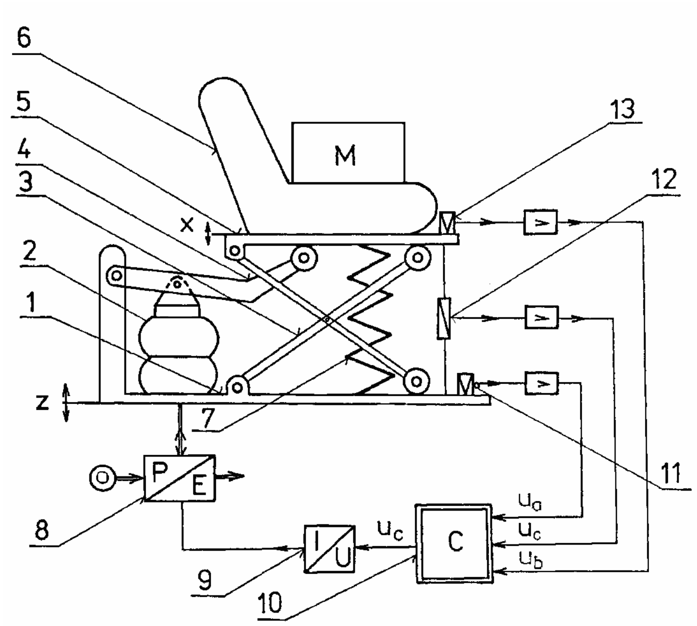


Fig. 6: Lay-out of the electro-pneumatic AVCS demonstrator seat system (Stein, 1996)

The demonstrator seat system is equipped with base vertical acceleration sensor **11** and upper part vertical acceleration sensor **13**. The respective amplified sensors output signals are fed to the electronic controller **10**, which transforms them to the output voltage u_C, controlling the electro-pneumatic transducer **8** via a voltage-to-current converter **9**. In the Phase I tests a conventional hydraulic damper was mounted between the seat base **1** and the upper part **5**.

Electronic feed-back damping, working on the Karnopp's sky-hook principle is implemented. The respective control loop is facilitated by the accelerometer **13** and associated amplifier. The respective signal u_b is fed to the controller **10**. The principal vibration control is facilitated by the feed-forward control loop, consisting of accelerometer **11**, associated amplifier and the respective part of the controller **10**. A positional control loop made of relative displacement sensor **12,** associated amplifier and controller **10** is implemented too.

3.2 The electro-pneumatic transducer

The electro-pneumatic transducer (electro-pneumatic valve, position **8** in Fig. 6) governs the pressure changes in the internal volume of the air spring. The valves could be of two types:

i. Flow control transducer,
ii. Pressure control transducer.

Both transducers are essentially solenoid operated valves in which the position of a slide within the valve body is proportional to the electrical current in the solenoid coil, generating a magnetic force. The magnetic force is counteracted either by a spring situated in the valve or by the end part of the slide acting as a piston. In this way the slide opens or closes orifices in the valve body and so governs the flow of fluid in the valve. Such valves (E/P transducers) are well suited for heavy-duty operation and are less costly than servo-valves, employing a nozzle and flap combination. However, their frequency bandwidth is limited to lower frequencies compared to the servo-valves. Both hydraulic and pneumatic proportional valves are in use; here only the pneumatic ones will be treated.

The pressure control proportional transducer operates on the so-called force balance principle, that is, the force exerted by the outlet pressure on the slide piston part is counteracted by the magnetic force of the solenoid. Hence for constant input current it maintains a constant pressure in the outlet. The internal self-balancing force feedback compensates for external disturbances (external temperature changes and/or leakage) and so maintains the set pressure in the adjacent volume. This is not so for the case of the proportional flow control valve. Here an external positional feed-back loop has to take action in such situations; else the pre-set datum position would not be kept for long. Hence, this approach is more complicated and expensive than that one with the pressure control valve.

3.3 Phase I experimental results

Phase I research was based on theoretical and simulation research of Gajarsky (1984). The results were presented, i.e. in Ballo et al. (1992), Stein et al. (1992) and in condensed form in Stein (1996a). Two flow control valves, made by SMC Corp. of Japan, were used: one for compressed air inflow into the air spring and a second one for air outflow from the air spring into atmosphere, each controlled by a separate voltage-to-current converter. This set-up facilitated a fail-safe operation in case of any malfunction of the AVCS, relying in this situation fully on the conventional air spring – hydraulic damper combination. The controller was set up on MEDA 41 TC analogue computer and facilitated rapid change in the structure and simple parameters optimisation.

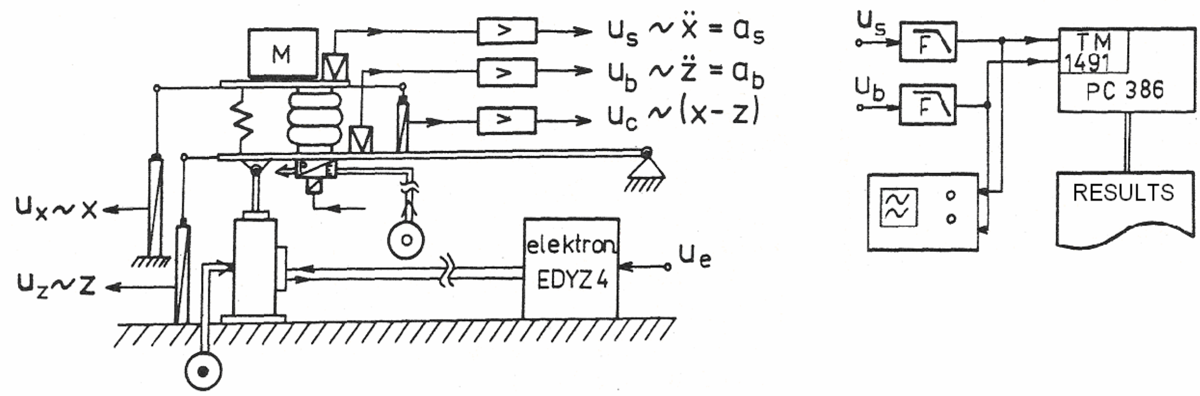
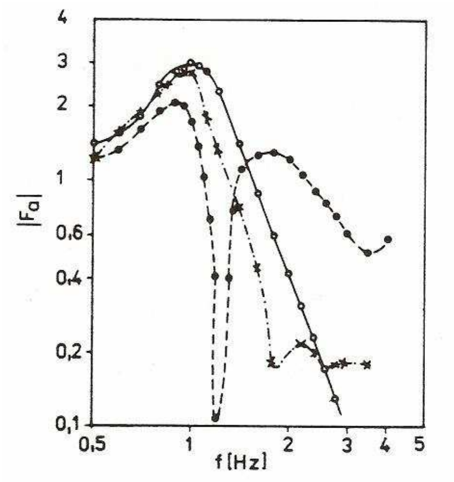


Fig. 7: The experimental set-up for tests of the electro-pneumatic AVCS demonstrator

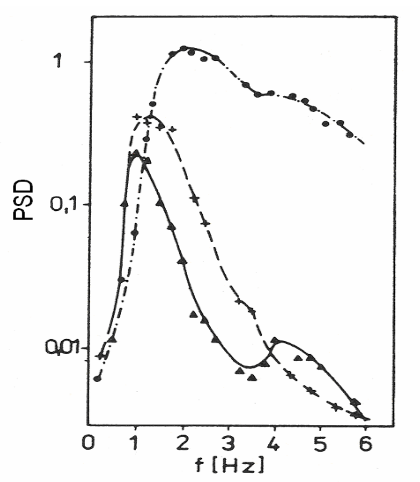
The experimental set-up, similar to Fig. 3, is depicted in Fig. 7. Except of respective accelerations $\ddot{z}, \ddot{x}$ also displacements $z, x, (x - z)$ were measured by LDVT transducers, all variables been converted to respective voltages. The evaluation procedure is based on the EN 30236-1 (1994) standard and is facilitated by proprietary PC-based acquisition and evaluation system (Stein et al., 1994); based on then available PC of the Intel 80386 type and a proprietary made data acquisition board, type TM 1491, with build-in anti-aliasing filters.

Selected Phase I results, when a conventional hydraulic damper was augmented by the AVCS action working on the feed-forward principle, are depicted in Fig. 8.

In Fig. 8a the FRF modules are shown for dummy loading of 75 kg and base vertical excitation by constant amplitude displacement of 15 mm. Nearly perfect vibration suppression at the frequency 1.25 Hz was obtained for the particular controller setting; but the resonance peak is not suppressed. By manipulation of the controls different FRF courses are obtained.



(a) **FRF modulus**: solid – passive system, dashed & dash-dotted – active system (various settings)



(b) **PSD curves**: dash-dotted – excitation for the 3rd machine class, dashed – passive system, solid – active system

Fig. 8: Passive and active systems performance characteristics (Stein, 1996)

In Fig. 8b the PSD curves are depicted, based on seat base vertical excitation by stationary random signal, corresponding to the 3rd machine class according to ISO 7096 (1982), i.e. for wheel loader. Note a rather small difference in the vibration suppression by the passive and the active system. The SEAT factor improvement was from 36 % down to 23 % (Stein et al., 1992). The already low SEAT factor value for the passive system was due to proper tuning of the passive system – its damped natural frequency was approx. 1.10 Hz, compared to the excitation band mid-frequency of some 2.0 Hz.

3.4 Phase I conclusions

The additional effect of the active part was assessed as too little to justify practical application of this particular electro-pneumatic AVCS. Further simulation studies were undertaken (Ballo, 1993); however, on the AVCS demonstrator seat system no verification of the promising simulation results could be obtained. A detailed analysis of the seat suspension mechanical design revealed a badly designed installation of the hydraulic dampers which were in fact defunct (Stein, 1996a). As no reconstruction of the suspension mechanism was possible another approach had to be followed, as suggested by the author: to remove the defunct hydraulic damper and fully rely on electronic control means for introduction of sufficient damping into the system, shown to be feasible in (Stein, 1998).

4. Electro-pneumatic AVCS for driver's seats with fully electronic damping control

4.1. Theoretical analysis

Following the above-mentioned fact an electro-pneumatic AVCS with fully electronic damping control was developed by the author, based on the common control principles, i.e. employing a combination of the feed-back and feed-forward control. In analogy to the Karnopp's sky-hook concept the author advocated the name sky-cloud for the feed-forward control approach (Stein, 2000a; Stein, 2002). Further text follows that one by Stein (2003).

Two principally different modifications of the electro-pneumatic AVCS were tested. The main difference is in the type of proportional electro-pneumatic valve (transducer) employed. **The first modification uses a proportional flow control E/P transducer 8**, with associated voltage/current convertor **9**, whose theory had been extensively treated e.g. in Gajarsky (1984), Stein et al. (1992) and Stein (1995). To maintain an equilibrium state in the static middle position a levelling control is required to cater for any pneumatic subsystem imperfections. This is facilitated by the feed-back loop, made of displacement sensor **12** with associated amplifier (output voltage u_c) and respective controller circuit within controller **10**.

In the second modification a proportional pressure control E/P transducer is employed instead (Stein, 1998; Stein, 2000). The transducer is based on internal force-balance technique and so inherently caters for any pneumatic subsystem imperfections. Hence the middle position is self-stabilising and no levelling control by a separate control loop is required, simplifying system design and reducing its costs.

The controller of pressure control valve is a simplification of that one for the flow control valve. The flow control approach will be described first and is partially pertinent to the situation when a hydraulic damper would be used, as was the case of Phase I research.

Starting with the mechanical scheme of Fig. 6 the control circuits are schematically indicated on Fig. 9. The control approach possesses two distinct features:

i. No passive damping is supposed (in fact a residual damping was observed; however, this would not suffice in practice). This observation was included in the simulation by a small linear viscous damper in parallel to the air and steel spring combination (Stein, 1998).

ii. The said feed-forward and feed back control approach consists of:

a/ *Feed back damping loop* made of load accelerometer **13**, respective pre-amplifier with transfer function (TF) $\mathbf{A}_B$, (outputting voltage u_b) feed-back controller with TF $\mathbf{R}_B$; catering for the damping.

b/ *Feed forward vibration compensation loop* made of base excitation accelerometer **11**, respective pre-amplifier with TF $\mathbf{A}_C$ (outputting voltage u_a) feed-forward controller $\mathbf{R}_C$; catering for base excitation compensation.

c/ *Position stabilising loop* made of relative displacement sensor **12** (a LVDT transducer) with respective pre-amplifier with TF $\mathbf{A}_X$ (outputting the voltage u_c) and the position controller $\mathbf{R}_X$. This controller is needed only if the flow control valve is employed. For pressure control a DC voltage at the input w suffices for setting the required datum.

d/ *A summing element and the output amplifier*, driving via voltage-to-current converter **9** the electro-pneumatic transducer **8**. Both are assumed to be part of the transducer/actuator sub-system, whose input is the control voltage u_C.

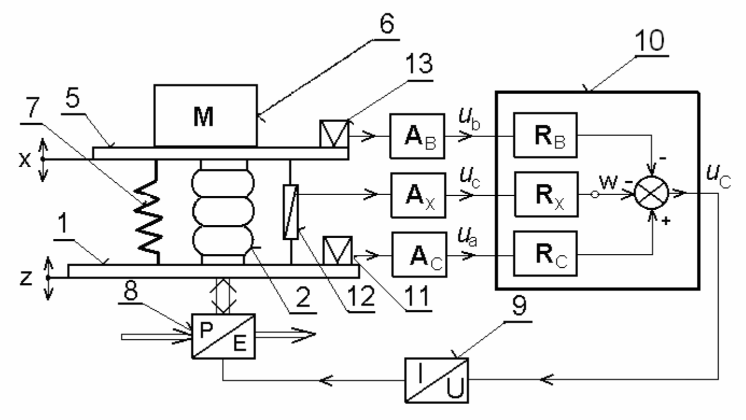


Fig. 9: Schematic sketch of the control circuits

The description of the control approaches was described in detail in (Stein, 1995) and in (Stein, 2000a). Here it will be briefly explained on the basis of the condensed equation of motion of a SDOF oscillatory system:

$$\ddot{x} + \left(\frac{b_{\mathrm{r}}}{M}\right)\cdot\left(\dot{x}-\dot{z}\right) + \frac{f_{\mathrm{Gb}}}{M} + \frac{f_{\mathrm{Ga}}}{M} + \left(\frac{k_{\mathrm{ef}}}{M}\right)\cdot\left(x-z\right) = 0\,, \tag{1}$$

where: $k_{\mathrm{ef}}/M = \omega_0^{\,2}$, and ω_0 is the natural angular frequency of the mechanical system,

$b_{\mathrm{r}}/M = 2\xi_{\mathrm{m}}\omega_0$ is the proportional damping coefficient of the system mechanical part,

$f_{\mathrm{Gb}}/M = 2\xi_{\mathrm{e}}\omega_0$ is the additional feed-back (the sky-hook term),

$f_{\mathrm{Ga}}/M = z\cdot k_{\mathrm{ef}}/M$ is additional force for base vibration compensation (the sky-cloud term).

The damping can be expressed in the form of damping ratios ξ_{m} and ξ_{e} describing the influence of residual mechanical and sky-hook type electrical damping, substituting the conventional mechanical viscous damper. The estimated value of $\xi_{\mathrm{m}} \cong 0.06$ (Stein, 1998) corresponds to transmissibility value of 8 at resonance, which would not suffice for practical applications. Additional damping is a must to comply with pertinent ISO standards for driver's seats. Rewriting Eq. (1):

$$\ddot{x} + 2\xi_{\mathrm{m}}\omega_0\left(\dot{x}-\dot{z}\right) + 2\xi_{\mathrm{e}}\omega_0\dot{x} + \omega_0^2 x = \left(\frac{k_{\mathrm{ef}}}{M}\right)z - \frac{f_{\mathrm{Ga}}}{M}\,. \tag{2}$$

If the term f_{Ga}/M, catering for the feed-forward control, is neglected for a moment the respective FRFs $\mathbf{G}(\mathrm{j}\Omega)$ of such an oscillatory system can be expressed as:

$$\mathbf{G}(\mathrm{j}\Omega) = \frac{\mathbf{X}(\mathrm{j}\Omega)}{\mathbf{Z}(\mathrm{j}\Omega)} = \frac{1 + 2\,\mathrm{j}\xi_{\mathrm{m}}\Omega}{\left(1-\Omega^2\right) + 2\,\mathrm{j}\Omega\left(\xi_{\mathrm{m}}+\xi_{\mathrm{e}}\right)}\,, \tag{3}$$

where: $\Omega = \omega/\omega_0$ is the dimension-less normalised frequency, related to the natural angular frequency ω_0 and ξ_{m}, ξ_{e} are the respective damping ratios. The Fourier transforms are denoted by bold capital letters. Note, that by this equation either the behaviour of a standard SDOF mechanical vibration control system is described - if ξ_{e} is zero, or the behaviour of a standard 2^{nd} order control system - if ξ_{m} is zero (Stein, 1998). For a given mechanical design both ω_0, ξ_{m} are fixed and so damping can be facilitated by the sky-hook damping term – i.e. by the feed-back gain represented by the damping ratio ξ_{e}. By introduction of the sky-hook type damping with $\xi_{\mathrm{e}} \in (0.2, 0.8)$ marked improvement is reached, as illustrated in Fig. 1.

The resonance amplification vanishes for $\xi_e \geq 0.6$, as already noted. These simulations are fully compatible with those of Margolis (1983).

The approach to the design of the controllers will be described for the steady state, using the Laplace transform with s as the complex operator:

1. The relation between the system mechanical variables and the corresponding signals is:

$$\mathbf{U}_b = \mathbf{A}_B\mathbf{X}s^2, \quad (4a)$$

$$\mathbf{U}_a = \mathbf{A}_C\mathbf{Z}s^2, \quad (4b)$$

$$\mathbf{U}_s = \mathbf{A}_X(\mathbf{X} - \mathbf{Z}) \text{ (only applicable for the flow control approach)} \quad (4c)$$

2. Applying Laplace transform onto Eq. (2) under steady state conditions and re-arranging Eq. (5) is obtained, where the total dynamic force Laplace transform $\mathbf{F}_G(s)$ caters for both vibration compensation and electronic damping:

$$\mathbf{X}s^2 + 2\xi_m\omega_0(\mathbf{X} - \mathbf{Z})s + \omega_0^2\mathbf{X} = \left(\frac{k_{ef}}{M}\right)\mathbf{Z} - \frac{\mathbf{F}_G}{M}. \quad (5)$$

3. The dynamic force has the following linear form:

$$\mathbf{F}_G(s) = \mathbf{N}_i(s)\cdot[\mathbf{R}_B(\mathbf{A}_B\mathbf{X}s^2) + \mathbf{R}_C(\mathbf{A}_C\mathbf{Z}s^2)], \quad (6)$$

where $\mathbf{N}_i(s)$ stands for the particular electro-pneumatic subsystem dynamic properties (Stein & Tököly 2000b; Stein, 2002). $\mathbf{R}_B$, $\mathbf{R}_C$ stands for respective controllers TFs.

4. After substituting into Eq. (5) and rearranging, following equation is obtained:

$$M\mathbf{X}s^2 + b_r(\mathbf{X} - \mathbf{Z})s + \mathbf{N}\mathbf{R}_B\mathbf{A}_B\mathbf{X}s^2 + k_{ef}\mathbf{X} = k_{ef}\mathbf{Z} - \mathbf{N}\mathbf{R}_C\mathbf{A}_C\mathbf{Z}s^2. \quad (7)$$

In the left hand side of Eq. (7) additional sky-hook type electronic damping has been introduced, described in Eq. 2 by the damping ratio ξ_e. On the other hand, to reach a non-trivial rest of the oscillator the right hand side of Eq. (7) should be zero. This physically means that no base vertical vibration is transmitted to upper part by the spring of stiffness k_{ef}.

5. From these two conditions the corresponding linear control laws for the TFs of the partial ideal controllers $\mathbf{R}_B$, $\mathbf{R}_C$ follow:

$$\mathbf{R}_B(s) = b_e\,\mathbf{N}_i^{-1}\mathbf{A}_B^{-1}s^{-1}, \quad (8a)$$

$$\mathbf{R}_C(s) = k_{ef}\,\mathbf{N}_i^{-1}\mathbf{A}_C^{-1}s^{-2}. \quad (8b)$$

6. It can be assumed that the electro-pneumatic subsystem can be described as a *linear SISO "black-box", i.e. as a voltage controlled generator* of force $\mathbf{F}_G$, described by its TF $\mathbf{N}_i$(s):

A/ Force generator using flow control was analysed first by Gajarsky (1984). Under assumptions used the dynamic force f_d as function of control voltage u_C can be expressed:

$$f_d = S_0 \cdot p_d = S_0 \cdot p_s \cdot \frac{\mathrm{r}\theta}{V_0} \cdot \int (q_0 a_0) \cdot u_C(t)\mathrm{d}t, \quad (9)$$

p_s, p_d are the static and the dynamic pressure components,

S_0, V_0 are the geometrical characteristics of the air spring, representing the air spring internal cross-section and volume, assumed to be constant,

q_0, a_0 are valve specific constants, not available to direct measurement,

θ, r are the absolute temperature of the air in the air spring and the gas constant of air.

Eq. (9) stipulates, that p_d depends on θ and p_s; however, predominantly, on the properties of the flow control valve. The governing relation is the proportionality of f_d to the time integral of the control signal u_C. Eq. (9) can be approximated by a TF of a first order astatic system $\mathbf{N}_f$:

$$\mathbf{N}_{\mathrm{f}}(s) = k_{\mathrm{f}}(p_{\mathrm{s}}, \theta) \cdot \frac{c_{\mathrm{f}}}{\mathrm{s}}. \tag{10}$$

Parameter c_f and coefficient k_f were identified from measured data (Stein & Tököly, 2000b).

B/ For the force generator using pressure control the dynamic pressure p_d variations in the adjacent air-spring internal volume V_0 are independent of Θ. For low frequencies, the sub-system TF $\mathbf{N}_p(s)$ can be approximated by a static pressure dependent gain k_p:

$$\mathbf{N}_{\mathrm{p}}(s) = k_{\mathrm{p}}(p_{\mathrm{s}}). \tag{11}$$

Above certain frequency the flow eventually ceases to follow the imposed command signal. It is stipulated that the TF $\mathbf{N}_p(s)$ can be described by a critically damped second order system:

$$\mathbf{N}_{\mathrm{p}}(s) = k_{\mathrm{p}}(p_{\mathrm{s}}) \cdot \frac{c_{\mathrm{p}}s + 1}{a_{\mathrm{p}}^2 s^2 + 2a_{\mathrm{p}}b_{\mathrm{p}}s + 1}. \tag{12}$$

The frequency independent parameters a_p, b_p, c_p and the static pressure dependent transducing coefficient k_p were determined by identification of measured data (Stein & Tököly, 2000b).

7. Hence, for the linear approach, described by Eqs. (5) and (7), assuming constant, frequency independent, gains $\mathbf{A}_B$ and $\mathbf{A}_C$, and the above derived relations for the respective E/P transducers, the idealised controllers $\mathbf{R}_B$, $\mathbf{R}_C$ descriptive laws are (K_{ij} are constants):

i. For flow control: $\mathbf{R}_B = K_{f1}$; $\mathbf{R}_C = K_{f2}/s$, (13a, b)

ii. For pressure control: $\mathbf{R}_B = K_{p1}/s$; $\mathbf{R}_C = K_{p2}/s^2$. (14a, b)

8. However, the ideal integrators, stipulated by Eqs. (13) and (14), are not practically realisable, as noted, e.g. in (Margolis, 1983). Hence non-ideal integrators (first order LP filters) have to be used with appropriate common time constant T_I i.e.:

i. For flow control: $\mathbf{R}_B = K_{f1}$; $\mathbf{R}_c = K_{f2}/(sT_I + 1)$; (15a, b)

ii. For pressure control: $\mathbf{R}_B = K_{p1}/(sT_I + 1)$; $\mathbf{R}_C = K_{p2}/(sT_I + 1)^2 + K_{p3}/(sT_I + 1)$. (16a, b)

4.2. Experimental results for harmonic excitation

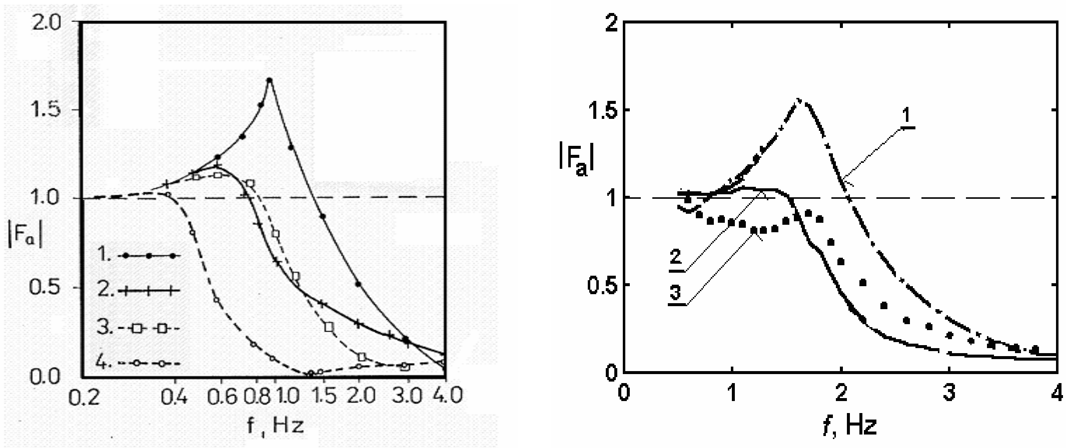
Two sets of FRF modulus were measured for flow control approach and for pressure control approach under harmonic excitation with constant displacement amplitude of 15 mm, using the vibration simulator described above:

A/ The absolute vertical displacement FRF defined as $\mathbf{F}_a = \mathbf{X}/\mathbf{Z}$.

B/ The so-called relative vertical displacement FRF, defined as $\mathbf{F}_r = (\mathbf{X} - \mathbf{Z})/\mathbf{Z}$. It is the measure of the relative displacement of the upper part **5** in respect to the lower part **1** for constant displacement excitation of the lower part z. This characteristic describes the relative movement of the seat suspension system in respect to the base, i.e. the seat stroke.

Fig. 10a and 11a depict both characteristics for the flow control approach. Accordingly, Figs. 10b and 11b depict the same characteristics for the pressure control approach; however measured at a different occasion. These figures clearly illustrate the extent of vibration control attainable. Two different settings of the controller $\mathbf{R}_B$, corresponding to electronic damping $\xi_e = 0.50$ and $\xi_e = 0.70$, respectively, were implemented when the flow control system was measured. Sky-cloud control is implemented in some cases, as indicated.

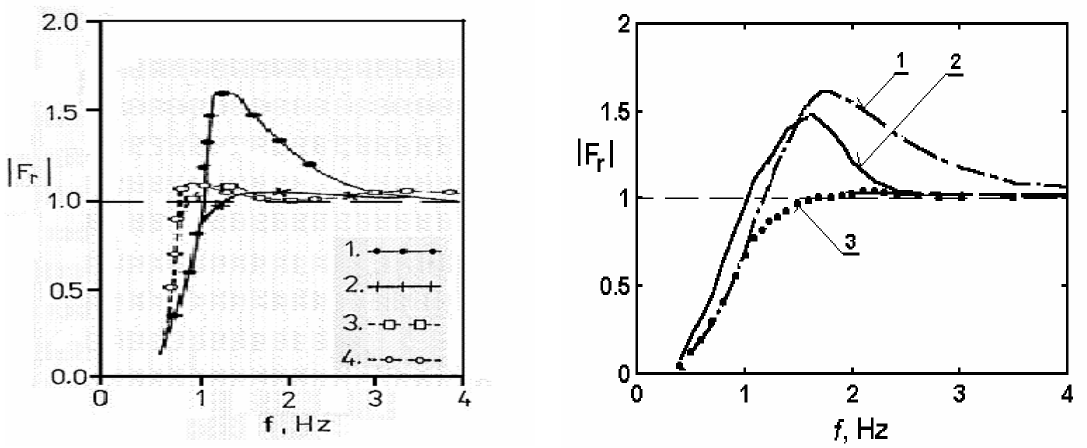
The best absolute displacement FRFs neatly follows that one stipulated theoretically in Fig. 1. Note also, that if the sky-cloud control is activated the respective relative displacement FRFs lack magnification at the resonance frequency. The extent of feed-back damping is quite good too. The 8-fold amplification at resonance with the residual mechanical damping was reduced to approx. 1.7, as required by the pertinent ISO standard. Note the very low cut-off frequency of approx 0.5 Hz for the flow control system.



(a) **Flow control**: 1 & 3 - $\xi_e = 0.50$, 2 & 4 - $\xi_e = 0.70$, 3 & 4 and with sky-cloud control

(b) **Pressure control**: 1 - sky-hook only, 2 & 3 - sky hook and sky cloud (different gains)

Fig. 10: Absolute displacement FRF modulus for the demonstrator electro-pneumatic AVCS



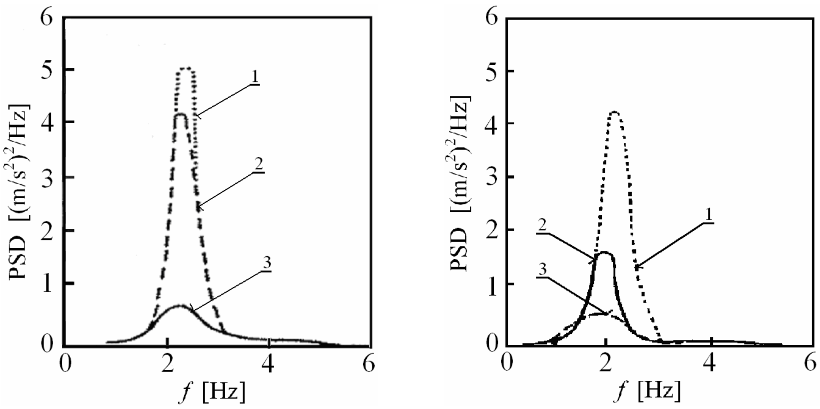
(a) **Flow control**: 1 & 3 - $\xi_e = 0.50$, 2 & 4 - $\xi_e - 0.70$, 3 & 4 and with sky-cloud control

(b) **Pressure control**: 1 - sky-hook only, 2 & 3 - sky hook and sky cloud (different gains)

Fig. 11: Relative displacement FRF modulus for the demonstrator electro-pneumatic AVCS

4.3. Experimental results for specific random excitation

Both systems were evaluated for use for heavy agricultural tractors (classes AG 2 and AG 3, with tractor mass 3.6 t to 6.5 t and above 6.5 t) according to the standard ISO 5007 (1990). The evaluation procedure was based on the EN 30236-1 (1992) standard, as above. The vibration simulator was excited by stationary narrow-band random acceleration signals, generated in the same way as before. The PSD curves for both flow control and pressure control are depicted in Fig. 12. In the second case the sky-hook type damping was improved, resulting in better performance.



(a) **Flow control** (AG 2 class) (b) **Pressure control** (AG 3 class)

Fig. 12: Vertical acceleration PSD courses: 1 – base excitation PSD, 2 – on-seat PSD for the sky-hook damping, 3 – on-seat PSD for the sky-hook and sky-cloud damping

More quantitative insight into AVCS demonstrator seat system performance can be gained by evaluating the SEAT factors. The respective SEAT factor data, as obtained in various measurements, are condensed in Table 1. It is seen, that by combining the feed back and the feed forward control a marked improvement can be reached. However, it is also seen, that if large enough feed-back gain is implemented satisfactory performance can be attained too (the SEAT factor is less than 100 %, i.e. the vibration is attenuated).

Table 1: SEAT factors in per-cent for different E/P valves and various controllers settings

AVCS system	Tractor class	sky-hook only		sky hook & cloud		Improvement
		$\xi_e = 0.5$	$\xi_e = 0.7$	$\xi_e = 0.5$	$\xi_e = 0.7$	
FLOW CONTROL (Stein, 1996)	AG 2	124	77	48	33	2.5 – fold
	AG 3	149	84	73	39	2.0 – fold
PRESSURE CONTROL (Stein, 2000a)	AG 2	170	—	40	—	—
	AG 3	—	70	—	46	1.5 – fold

4.4. Conclusion on the electro pneumatic AVCS driver's seat demonstrator system

Presented results indicate vibration mitigation properties of the described electro-pneumatic AVCS demonstrator seat system with electronic damping control, employing either a flow or a pressure control electro-pneumatic transducer and air spring as the actuator. Large extent of vibration suppression was demonstrated. The dynamic properties of the system can be tuned so that the FRF closely follows the theoretical one, i. e. having no amplification at resonance.

5. Latest development of the electro-pneumatic AVCS for driver's seats

The research into electro-pneumatic AVCS for driver's seat was terminated around year 2000. The results were disseminated, see Guicking (2008), and also within the VIBSEAT Project*.

Recently, due to introduction of the Directive 44/2002/EC (2002) as from June 2005 some newer development can be foreseen. According to the Directive 44/2002/EC (2002) (implemented in the Slovak Republic by the Governmental ordnance No. 416, 2005 Col.) a worker subjected to whole body vibration (which in many cases is the vibration in the vertical direction) for a standardised 8-hours shift is subject the following limits Mohr (2007):

a/ For the value of the characteristic measure $A(8) < 0.5\ \mathrm{m.s^{-2}}$ – *the action value* no measures are required from the employer;

b/ For the value of the characteristic measure $A(8)$ between $0.5\ \mathrm{m.s^{-2}}$ and $1.15\ \mathrm{m.s^{-2}}$, the employer has *the obligation* to take technical, organisational and other measures at his discretion to decrease the vibration exposure by technically feasible ways; in the same instant *to inform, educate and train workers* concerned in using vibration mitigation means and *perform health surveillance*.

c/ No work ought to be performed if *daily exposure value* of $1.15\ \mathrm{m.s^{-2}}$ in 8 hours is exceeded.

The assessment and/or measurement of the characteristic measure $A(8)$ is governed by the EN 14253 (2003) standard (in Slovakia by the STN EN 14253 (2007) standard).

To enable the employer – owner of e.g. the earth moving equipment, etc. to follow suit of this Directive, various means became recently available: i.e. improved human vibrometers; human vibration dosimeters suitable for mounting on the driver's seats; guidance tables of likely weighted root-means-square acceleration values in all three orthogonal axes (Mohr, 2007) and, last but not least, markedly improved driver's seats of novel design. Important European seat manufacturers are about to introduce advanced pneumatic driver's seats with an active vibration control system, employing some of the ideas described above. Unfortunately, the available information is still very scarce, with a few quantitative data:

A novel electro-pneumatic active seat was described by Meyer and Ortman (2007). It is noted there, that vibration mitigation properties in the vertical direction are improved in comparison to standard pneumatic seats. Moreover, the seat performance is independent of damper setting and of the operator's mass. The SEAT factor value is improved for excitation according to ISO 7096 (2000) machine class EM 1 (which is similar to that one of the AG 2 class wheeled tractors) from just below 100 % for a conventional seat to some 60 – 70 % for a seat with the electro-pneumatic AVCS. Essentially, the Karnopp's sky-hook principle is used. There is no difference in external appearance of the novel seat; the actuators, sensors and the digital controller are build in as "add-on" into conventional seat suspension system.

Another advanced seat for large agricultural tractors or earth moving machines is described in company press releases as having electronically controlled dynamic properties, which improves the SEAT factor by some 40 % in comparison to a conventional seat with an air spring♦. "The novel seat employs a relative displacement and vertical acceleration sensor mounted on the seat suspension system upper platen. The signals are processed by a digital

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* See: VIBSEAT project web page: *http://www.soton.ac.uk/~vibseat/index.htm*.

♦ "Presse-Infromation Grammer A. G." on the occasion of Agritechnica 2007,
"Presse-Infromation Grammer A. G." on the occasion of Bauma 2007.

controller, which in turn governs a flow control electro-pneumatic valve. By changing volume flow the air spring stiffness characteristics is being automatically changed. In this way air spring stiffness characteristics is selected which, depending on frequency, mitigates vibration”

A seat suspension system, based on same principles as described above was tested in laboratory by Kupka et al. (2007), using real field-measured cabin floor vertical acceleration. Marked improvement in comparison to a standard seat with an air spring was demonstrated.

Conclusion

It seems, that due to increased demands on worker’s health and on efficient utilisation of expensive and complex machinery, also in view that automobile grade electronic maturity, the time has come to introduce on industrial scale sophisticated driver’s seats with electro-pneumatic AVCS, based on principles researched in the past at the IMMM SAS. Also there is a newer development in the electro-hydraulic AVCS for driver’ seats. It is acknowledged (Guicking, 2008) that IMMM SAS made important contribution to this technology.

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