

AN ELECTRO-PNEUMATIC ACTIVE VIBRATION CONTROL SYSTEM FOR THE DRIVER'S SEAT OF AGRICULTURAL TRACTORS

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The aim of this contribution is to present some results of the study of driver's seat with an electro-pneumatic active vibration control system applied for driver's seats of agricultural wheeled tractors. The actuator is parallel to the conventional pneumatic seat suspension and a proportional electro-pneumatic valve is used as the control element. The description of the full scale dummy system is augmented by a short theoretical description of the 1 DOF model of this system and by some results of measurements on a laboratory vibration simulator. The seat vibration control properties are evaluated according to the ISO Standard 5007:1990 pertinent to evaluation of seats for agricultural wheeled tractors.

1. Introduction

The contemporary conventional passive Vibration Control Systems (VCS) for vehicles and other transport means have reached a rather sophisticated level of vibration abatement. They cannot however, meet all the requirements of pertinent health regulations. Therefore new methods of vibration control have been developed. Some of them are already manufactured for luxury cars or subject to performance tests for special vehicles. These systems are based on the use of fast and expensive electro-hydraulic actuators and of a digital state variable control. The practical use of pneumatic elements for Active Vibration Control Systems (AVCS) is, to author's knowledge, rather limited. Much effort in the field of research of AVCS has been spent especially in Poland, e.g. [5, 6, 10-13], and elsewhere. As often reported, e.g. in [1-3, 7, 19] the effort at this Institute has been concentrated on the research of AVCS based on the compensation principle.

In the last paper on this topic [3], some advantages of such a system were shown. Some first stage experimental results were published, e.g. in [2, 19], but they were not satisfactory. It was noted later that one of the drawbacks was the wrong mounting position of the hydraulic damper in the scissors type guiding mechanism. Instead of

vibration damping it rather hampered the proper action of the active part. Hence it was decided to remove the hydraulic damper and to use the "sky-hook" damper working on the electronic principle, as described first by KARNOPP in [9]. The new system reported further below, uses artificial damping by employing the feed forward control combined with the well proven feed forward control of the compensation principle. For the theoretical description of the system the interested reader is referred to [2, 7, 16, 19]. An abridged explanation of the control system will be given below according to [17]. Furthermore some experimental results pertinent to use for driver's seats for agricultural wheeled tractors will be given together with the assesment of the required pneumatic power for proper operation of the system.

2. Description of the system

The schematic layout of the full scale dummy seat with active electro-pneumatic suspension used in the experimental research is depicted in Fig. 1. The pneumatic spring 2 with inside absolute pressure p_2 acts on the upper part 5 of a scissors type guiding mechanism 3, which is supporting the cushion 6 with seated operator of mass M that vibrations in the vertical direction x_1 have to be controlled. This structure is mounted on the base 1 which is excited mostly in the vertical direction with the displacement x_2 . A relief steel spring 4 is situated between the base 1 and the upper part 5 to hold the supposed minimal mass M at the required static height h_0 (static middle position), approximately in the middle of the possible travel of the scissors type mechanism 3. The active part is formed by the electro-pneumatic transducer 8,

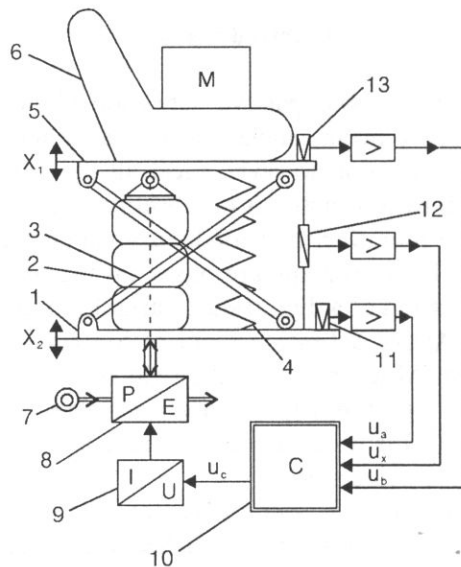


Fig. 1. Schematic lay-out of the dummy system (description in the text).

the source of compressed air 7 of absolute pressure p_1 and the outlet to the atmosphere of absolute pressure p_3 . The dummy system is operated by the relative displacement sensor 12, the base vertical acceleration sensor 11 and the upper part vertical acceleration sensor 13. The respective amplified sensor output signals are fed to the electronic controller 10 which transforms them to the output voltage u_c controlling the electro-pneumatic transducer 8 by a matched voltage-to-current converter 9.

In the experimental realization emphasis was given to the proper design of the mechanical part in order to minimize the influence of geometrical properties of the air spring 2 and the proper choice and installation of the electro-pneumatic proportional transducer 8. A commercially available transducer with a matched voltage-to-current converter 9 was used. To simplify further the research work, a constant mass M made of lead shots in bags was fixed to the cushion 6 in order to simulate the seated driver. The relative displacement sensor 12 was a commercially available inductance sensor with a corresponding amplifier. The vertical acceleration sensor 11 and 13 were commercially available piezoelectric accelerometers with appropriate impedance converters connected to the respective inputs of the electronic controller 10. The electronic controller is a simple linear electronic system. It is essentially a three loop linear controller as shown in Fig. 1. It is immaterial wheather this control structure is realized by a digital computer, by a DSP with a suitable analogue input/output subsystem, by an analogue controller or by an analogue computer. In our case, after thorough tests with the MEDA type analogue computer, an analogue controller has been built of standard electronic components and used.

3. Short theoretical description

Physically the system can be treated as an oscillatory system with one degree of freedom working in the vertical direction. The simplified mathematical description of the mechanical part of the system is based on the following equation of motion [2, 17]

$$M \cdot \ddot{x}_1 + k^* \cdot (x_1 - x_2) + f = 0, \quad (3.1)$$

where k^* is the effective spring constant of the parallel combination of the steel spring 4 and the pneumatic spring 2 of Fig. 1; the mass M stands for the seated driver. The dynamic properties of the seat cushion are neglected for simplicity and only the suspension system is considered. For a more detailed analysis of a driver's seat, the interested reader is referred to the paper of RAKHEJA *et al.* [15].

The force f , generated by the air spring 2 as actuator, consists of three components:

- damping force, proportional to the isolated body absolute velocity \dot{x}_1 (proportionality constant b);
- compensation force, proportional to the base vertical acceleration \ddot{x}_2 ;
- static force for maintaining the proper middle position. This static force is not further considered.

It can be shown (see e.g. [7, 16, 19]) that the relation between the output force $f(t)$, acting on the upper part 5 to control the voltage $u_c(t)$ of the combination of the voltage-to-current converter 9, actuator 2 and the proportional transducer 8, is linear if some strict constraints on the fluid flow in the transducer are maintained. It has an integrating character and therefore the respective Frequency Response Function (FRF) of the complex argument $p = j \cdot \omega$ has the form $G_A = H_A/p$. At the assumption of linearity and the system stability the Fourier transform of the exerted force was derived in [17] in the following form (the respective Fourier transforms are denoted by capital letters):

$$F = G_A \cdot N \cdot [R_B \cdot (A_B \cdot X_1 \cdot p^2) + R_C \cdot (A_C \cdot X_2 \cdot p^2)] \quad (3.2)$$

where A_B and A_C are the gains of the accelerometer preamplifiers and R_B and R_C are the partial controllers FRF. The controllers are realized in the controller block 10.

After subjecting Eq. (3.1) to the Fourier transform and introducing Eq. (3.2) the following equation holds (where ω_0 is the system undamped natural frequency):

$$\begin{aligned} X_1 \cdot p^2 + G_A \cdot N \cdot R_B \cdot A_B \cdot X_1 \cdot p^2 / M + \omega_0^2 \cdot X_1 = \\ = \omega_0^2 \cdot X_2 - G_A \cdot N \cdot R_C \cdot A_C \cdot X_2 \cdot p^2 / M. \end{aligned} \quad (3.3)$$

The left side of Eq. (3.3) would correspond to a damped linear oscillator with the corresponding mechanical damping b^* if according to the principle of the "sky hook", as described by KARNOPP [9], the middle term would correspond to $b^* \cdot X_1 \cdot p / M$. On the other hand, to reach a non-trivial rest of the oscillator the right hand side of Eq. (3.3) should be zero. From these two conditions the corresponding control laws for the respective controllers R_B and R_C follow:

$$R_B = b^* \cdot (G_A \cdot N)^{-1} \cdot A_B^{-1} \cdot p^{-1} = b^* \cdot (H_A \cdot N)^{-1} \cdot A_B^{-1}, \quad (3.4)$$

$$R_C = k^* \cdot (G_A \cdot N)^{-1} \cdot A_C^{-1} \cdot p^{-2} = k^* \cdot (H_A \cdot N)^{-1} \cdot A_C^{-1} \cdot p^{-1}. \quad (3.5)$$

The term N could be looked upon as a correction term to H_A . If their combination equals to 1, then it follows that the damping controller R_B is essentially a P-controller, with gain corresponding to the damping b^* , and the compensation controller R_C corresponds to a P-I controller. The only problem, as noted by MARGOLIS [14], is the suitable design of this controller. Note that the compensation loop is fully effective only if equation (3.5) is met both in phase and amplitude in the required frequency band.

4. Test procedure

The ISO Standard 5007:1990 [21] is an international standard which describes the requirements and procedure for laboratory tests of driver's seats for agricultural tractors with unsprung rear axle in a reproducible manner. It distinguishes essentially two tests:

a) **test damping:** the seat is loaded with two test masses of 40 kg and 80 kg, respectively, and a sinusoidal vibration displacement of a 30 mm pp amplitude is applied to the seat base at frequencies within the range 0.5 Hz to 2.0 Hz with a step of 0.05 Hz at maximum. The ratio V of the weighted rms vibration acceleration in the vertical direction at the seat surface a_{wfs} to that one at the seat base a_{wfb} , measured by the same method, shall be determined in the given frequency band and graphically displayed. The maximum value and the respective frequency is of interest, since it describes the damping properties of the driver's seat under standardized harmonic excitation.

b) **random vibration test:** with two test persons of 50 kg \pm 1 kg and 98 kg \pm 5 kg of that not more than 5 kg or resp. 8 kg could be comprised of lead balasts in a belt. The seat is subjected to a narrow band stationary random excitation in the vertical direction, with a normal amplitude probability distribution. The spectral density of the acceleration power (PSD) is defined in the Standard for the agricultural wheeled tractor classes. The vertical acceleration PSD is generated so as to describe the vibration excitation on agricultural wheeled tractors of different sizes/masses during their normal operation in reproducible manner. The most important data for those three tractor classes are given in the following Table 1, the respective vertical acceleration PSD courses are shown in Fig. 2.

Table 1. Tractor classes according to the ISO 5007:1990 Standard

| Tract. class | Unbalanced mass m, kg | Centre frequency f_p , Hz | Base vertical wght. acceleration a_{wB} , $m \cdot s^{-2}$ |
|--------------|--------------------------|--------------------------------|---|
| 1st | up to 3 600 | 3.25 | 2.05 |
| 2nd | 3 600 to 6 500 | 2.35 | 1.50 |
| 3rd | over 6 500 | 2.20 | 1.30 |

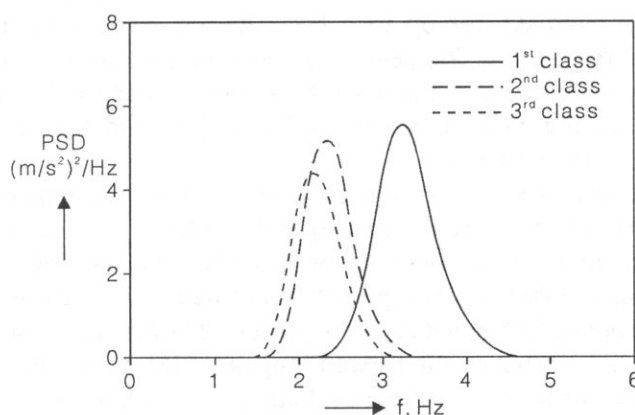


Fig. 2. Course of the vertical acceleration PSD curves for various agricultural tractor classes according to ISO 5007:1990 Standard.

The evaluation procedure is similar to that for earth moving machines as described by the ISO 7096:1982 Standard [22] (see e.g. [18, 20]). The resultant value of the so called corrected value of frequency weighted RMS acceleration transmitted to the seated operator a_c is given. This value describes the overall vibration control properties when subjected to a random vibration. The only important difference to the ISO 7096:1982 Standard is that there are no limit values given, i.e. no direct relation to the standards pertinent to the vibration influence on man such as the ISO 2631/1:1985 Standard [23] or respective national hygienic regulations. The so called SEAT Value, introduced by GRIFFIN [8], could be readily calculated from the measured data. They describe the vibration attenuation by the seat as a relative measure. For good seats this value is below 100%, but experimental evidence exists that seats with no vibration attenuation at all (i.e. with SEAT value above 100%) are sometimes used.

A further development of the procedures of standardized laboratory testing of driver's seats in the European Standard EN 30326-1:1994 [24] which is in fact the ISO 10326-1:1992 Standard. This Standard could be viewed as a "2nd generation Master Standard" describing the general requirements on seat testing in laboratory, whereas the specific conditions of application of this Standard for testing of seats for specific use (e.g. for use in earth moving machines or agricultural tractors, for passengers and crew of road vehicles, railway vehicles etc.) are dealt with in detail in the so called "Application Standards".

5. Experimental results

The model of the active vibration control system for the driver's seat for agricultural tractors was mounted on a beam of 2 m length, pivoted on one end and excited in the vertical direction by a servo-hydraulic cylinder on the other one. This laboratory test stand was able to exert an vibration amplitude up to 50 mm over a frequency range from 0.1 Hz up to 5 Hz. The test stand was driven either by a harmonic signal from a low frequency generator or by a narrow band stationary ergodic signal, generated from white noise by analogue shaping filters so as to fulfil the requirements of the ISO Standard 5007:1990 [21]. On the vertical vibration acceleration PSD at the seat base.

The test stand was fitted with two piezoelectric acceleration sensors, independent of the control sensors to measure the base a_b and cushion a_s vertical acceleration, the second one mounted in a special disk as described by the ISO Standard. The respective electrical variables were amplified, monitored on an oscilloscope and fed to a data acquisition board embedded in a 386 type PC. The evaluation was furnished by appropriate software written in the fortran language [18]. It was based on previous extensive experience with computerized evaluation of driver's seat vibration control properties.

The courses of the value V measured, according to the clause a, for both the passive and active systems, for a load of dead weight of 80 kg are shown in Fig. 3. The

solid lines represent a seat with artificial damping only for two different values ("low damping" #1 and "high damping" #2; the dashed ones represent a seat with both artificial damping and compensation engaged for both the "low damping" #3 and "high damping" #4 as above. Note the difference between these two curves above 1.3 Hz where the most vertical vibration energy of agricultural tractors is concentrated. The pertinent values for the seat with the "sky-hook" artificial damper are as follows:

- "low damping": $V_{\max}=2.60$, $f_{\max}=1.10$ Hz;
- "high damping": $V_{\max}=1.36$, $f_{\max}=0.90$ Hz.
- no such values could be given for the combination of both vibration control systems.

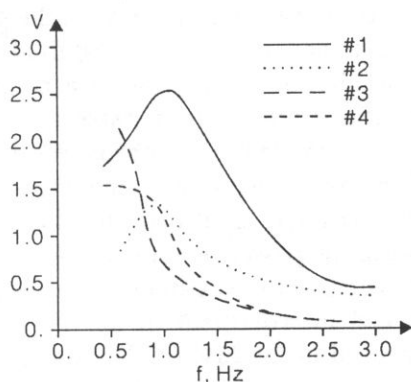


Fig. 3. Modulus of the active vibration control system response to harmonic excitation according to the clause a (description see text).

Table 2. Values of a_c measured according to the clause b of the ISO 5007:1990 Standard for simulated operator mass of $M=75$ kg and for the two cases of seat damping

| tract. class | System: | sky hook | | comp.+hook | | Improvement | |
|-----------------|--------------------------------------|----------|------|------------|------|-------------|-------------|
| | damping: | high | low | high | low | high | low |
| 1st | a_c , $\text{m}\cdot\text{s}^{-2}$ | 1.23 | 1.46 | 0.55 | 0.64 | $2.3\times$ | $2.3\times$ |
| | a_c , dB | 122 | 123 | 115 | 116 | 7 dB | 7 dB |
| | SEAT, % | 60 | 71 | 27 | 31 | $2\times$ | $2\times$ |
| | P , kW | 1.45 | 0.44 | 1.45 | 0.84 | | |
| 2nd | a_c , $\text{m}\cdot\text{s}^{-2}$ | 1.52 | 1.86 | 0.49 | 0.71 | $3.0\times$ | $2.6\times$ |
| | a_c , dB | 124 | 125 | 114 | 117 | 10 dB | 8 dB |
| | SEAT, % | 77 | 124 | 33 | 48 | $2\times$ | $2.5\times$ |
| | P , kW | 1.47 | 0.93 | 1.47 | 1.14 | | |
| 3rd | a_c , $\text{m}\cdot\text{s}^{-2}$ | 1.10 | 1.94 | 0.50 | 0.95 | $2.1\times$ | $2.0\times$ |
| | a_c , dB | 121 | 126 | 114 | 120 | 7 dB | 6 dB |
| | SEAT, % | 84 | 149 | 39 | 73 | $2\times$ | $2\times$ |
| | P , kW | 1.66 | 1.04 | 1.61 | 0.20 | | |

The results for random excitation, according to the clause b, for the tractor classes mentioned are given in Table 2. Note the difference in the absolute values of a_c for the two damping set. In fact, the system with "low damping" would not be suitable for practical use, since the a_c values are rather high and the SEAT value exceeds 100%, i.e. the seat would rather amplify the vibrations than to abate them. This is due to the fact that the seat resonant frequency f_{\max} is close to the excitation centre frequency f_p of Table 1 and the V factor is larger than for the seat with "high damping". This is specifically valid for the 2nd and 3rd tractor classes.

An important question for all active systems is their complexity and energy consumption in comparison to common passive systems. As described by BALLO in [3, 4], the energy consumption of an active pneumatic suspension is mainly due to the need of maintaining a necessary pressure drop between the source of compressed air and the free atmosphere so as to ensure a supercritical fluid flow in the proportional flow control valve. This consideration led to using a compressed air source with absolute pressure $p_1 = 0.9$ MPa. Because no suitable flowmeter was available, an indirect method was used. The average power consumption of the compressor P kW in a certain time period was measured without taking into account the efficiency and pneumatic losses. The values measured are also given in the Table 2. Note that there is virtually no difference between the power requirement for the active system with the "sky-hook" and that for a system with "sky-hook" and compensation for the "high damping" setting, but marked improvement in the vibration control is attained. Somewhat more power is needed in the case of "low damping" setting for a similar increase in the improvement of vibration control.

6. Conclusion

The experimental results presents indicate the technical feasibility of the usage of an electro-pneumatic active vibration control system for the suspension system of a driver's seat for agricultural tractors. Despite its complexity, it performs quite well on the test stand. The improvement in the vibration control properties by introducing the vibration compensation facility of the sky-hook damper is of the order of approx. 7 dB, i.e. twofold. In other words, by introducing the combined active vibration control, the vibrations are diminished in average to approx. 50% of the transmitted by the system with the sky-hook only. On the other hand, the better vibration control properties of the seat with both the vibration compensation and sky-hook are paid for by higher energy demand and increased complexity of the system. Note that the system employing the sky-hook with "low damping" is rather bad and would not be suited for practical use.

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