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ACTIVE VIBRATION CONTROL SYSTEMS FOR DRIVER'S SEAT OF EARTH-MOVING VEHICLES

I. BALLO

Institute of Materials and Machine Mechanics
Slovak Academy of Sciences
CS–836 06 Bratislawa Računska str. 75. Słovakia

In the article a survey of the activity of the Institute of Materials and Machine Mechanics of the Slovak Academy of Sciences in the field of research of active vibration control systems (AVIS) is given. The contribution starts with some remarks on the vibration control properties of common passive systems. In the next part the possibilities of combination of passive and active vibration control systems (VCS) are discussed. The next topic are the results of theoretical and experimental research of electro-hydraulic and electro-pneumatic acitve VCS for driver seats in earth-moving machines. The last part of the paper deals with the problem of power consumption of the AVIS.

1. Introduction

The problem of driver's insulation from excessive mechanical vibration is an ever actual research topic. The improvement of driver's insulation from excessive vibration means not only decrease of health risk to the driver and improvement of his work comfort, but often also marked improvement in the full utilisation of the working capacity of the machine. This problem is even pronounced in earth moving machines, which operate in rough terrain. Here the main vibration control elements are the tyres and the driver's seat.

In contemporary driver's seats the most common vibration control system is a passive one, which does not need any external source of energy for his proper operation. In most systems an air spring is used as the main resilient element. The use of this structural element enables to reach rather low natural frequencies of the vibration control system. In addition, by the control of the average static pressure in the air spring, a suitable static middle position of the isolated load can be maintained, regardless of the changes in its weight. Because only the average static pressure has been changed and not the instantaneous one, such systems are often referred to as "semi-active" ones. Despite the fact that these systems require supply of compressed air, they are treated as passive ones.

2. Some limits of passive vibration control systems

For the mathematical description of the vibration control system of the driver seat (in the first approximation), a linear damped mechanical oscillator of one degree of freedom is used. The vibration control effect of passive systems is improving in line with the decrease of the system's natural frequency. In contemporary systems extremely low natural frequencies of the order of 1 Hz and below have been reached. Further decrease of the system's natural frequency brings about many technical difficulties and hence further improvement in this direction is not sought for.

Also the strive for decrease of the system's natural frequency does not bring always any marked improvement in the overall vibration control properties as the simplified basic theory would indicate. This theory is based on the assumption of stationarity of the excitation, i.e. it assumes stationary vibration of the earth-moving machine's cabin floor in the vertical direction. Any "bumps" in the excitation, i.e. transient phenomena are not covered by this theory.

The effect of a transient in the excitation is illustrated in a typical example, with results depicted in graphical form in the Fig. 1.

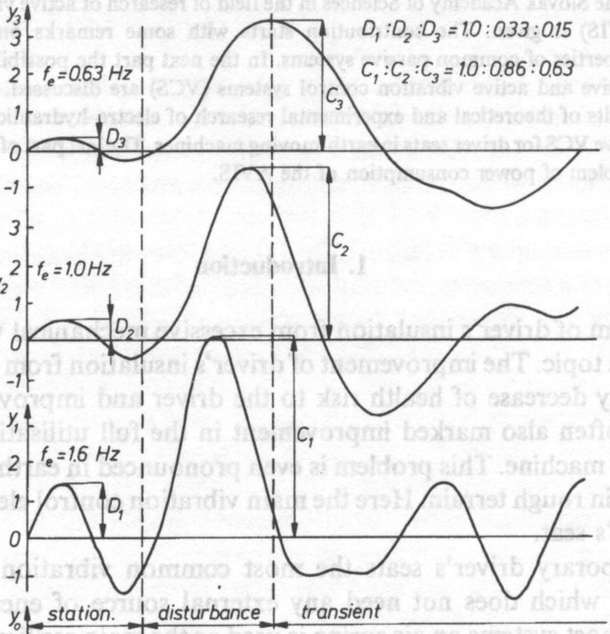

Fig. 1

In the lower part of the picture a harmonic excitation function is depicted, to which in the time interval t_1 to t_2 a disturbance is superimposed. In the upper part of the picture the reaction of three different passive vibration control systems is depicted, each of them having different natural frequency — f_e = 1.6; 1.0; 0.63 Hz in turn. The steady-state amplitudes ratio is 1; 0.33 : 0.15 in turn, but the maximal amplitudes of the transient are in the ratio of 1:0.86:0.63. From the simple example it is clearly seen that, by the decrease of the system's natural frequency from 1.6 Hz to 0.63 Hz, a 6.7-fold decrease in steady state response amplitude was reached. But the improvement in the amplitude of the transient response was only 1.6-fold.

Different reaction of the linear passive vibration control system to stationary and nonstationary excitation, as was illustrated in the picture of the previous example is also generally valid for narrow-band random excitation signals. This conclusion has an important implication for the assesment of vibration control properties of driver seats for earth-moving machines, agricultural tractors, etc. i.e. in general for any off-road vehicle operating in rough terrain. Under such conditions the non-stationary vibration of the vehicle and hence also of the cabin floor is the prevailing form of excitation. The protection of the seated driver from adverse effects of intense vibration is far less effective, as the basic theory and "perhaps" the international standards would predict.

3. Active vibration control systems

An active vibration control system is in principle a specific sort of servo-system. It uses an external source of energy. Various sources of external energy could be used. In this paper only fluids systems will be discussed, i.e. those employing the energy of compressed oil or air.

Active vibration control systems discussed in this paper consist essentially of three distinct parts:

a) Sensor or sensors of vibration. These measure the characteristic vibration time variable, which has to be controlled. In the case of driver's seat for earth-moving machines it is the cabin floor vertical vibration acceleration.

b) Electronic controller, in which appropriate control signal for the actuator is generated from the signal supplied by the sensor.

c) Actuator, which could have different fashion, depending on the energy source. In this paper two distinct actuators will be discussed.

1. An electro-hydraulic servo-cylinder, in which the input electrical signal governs the in-or outflow of hydraulic oil.

2. An air spring, in which the inflow and outflow of compressed air is controlled by governing signal.

From the point of view of control theory, the actuator can be situated in series or in parallel to the spring-damper combination of the seat suspension (Fig. 2a, b).

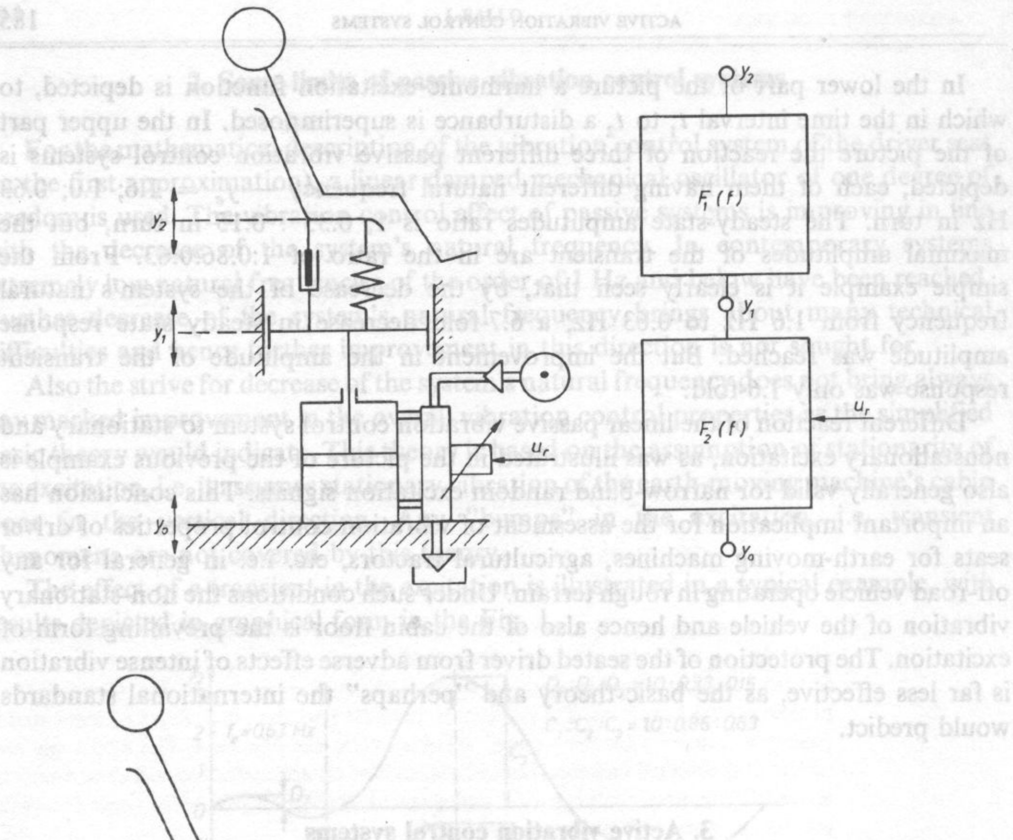

Fig. 2

The dynamic properties of the seat can be described by the transfer function $F1\ (f)$, that one of the series of active vibration control system AVIS by the transfer function $F2\ (f)$. According to the Fig. 2a the frequency response of the passive seat suspension $F1\ (f)$ can be improved by the series connection of the AVIS to the overall frequency response $F1'$:

$$\text{passive} : F1,$$
$$\text{passive} + \text{active} : F1' = F1 \cdot F2.$$

As it will be shown further, this series combination of active and passive suspension can exhibit far better vibration isolation properties than the passive suspension only.

In case of a parallel AVIS, the effect of the passive part $F1\ (f)$ is combined with the active part $F3\ (f)$ to form the resultant transfer function $F1''\ (f)$:

$$\text{passive} : F1,$$
$$\text{passive} + \text{active} : F1'' = F1 - F3.$$

This simplified approach would suggest that, under circumstances, the combined transfer function $F1''$ would tend to zero. However, it could be difficult to achieve this in practice.

As follows from the previous description, active vibration control systems are rather complicated devices in comparison to common passive ones. They need an external source of energy. The extent of energy consumption could be a sort of a burden. On the other hand, active vibration control systems have a markedly improved vibration control effect in comparison to contemporary passive ones.

4. Electro-hydraulic active vibration control system

The active vibration control system of electro-hydraulic type, which was investigated in our research laboratory, is of a series structure, Fig. 3. It consists of cylinder mounted on vehicle chassis *1*. The piston rod supports platform *3*, which can move in vertical direction only. A standard driver's seat *5* with the passive spring-damper vibration control system *4* is mounted on platform *3*. The oil flow in cylinder *2* is controlled by the servovalve or proportional flow valve *12*, which in turn is operated by the control voltage u_r (after necessary amplification in a power amplifier *11*). The control voltage u_r is generated in the electronic controller *10*. The AVIS is equipped with necessary sensors and preamplifiers, i.e. accelerometer *6* for the measurement of the chassis absolute acceleration. Also the relative displacement has to be measured by the displacement transducer *7* to maintain the static position. In our considerations it is supposed that all measuring chains are in the required frequency range linear and frequency-independent.

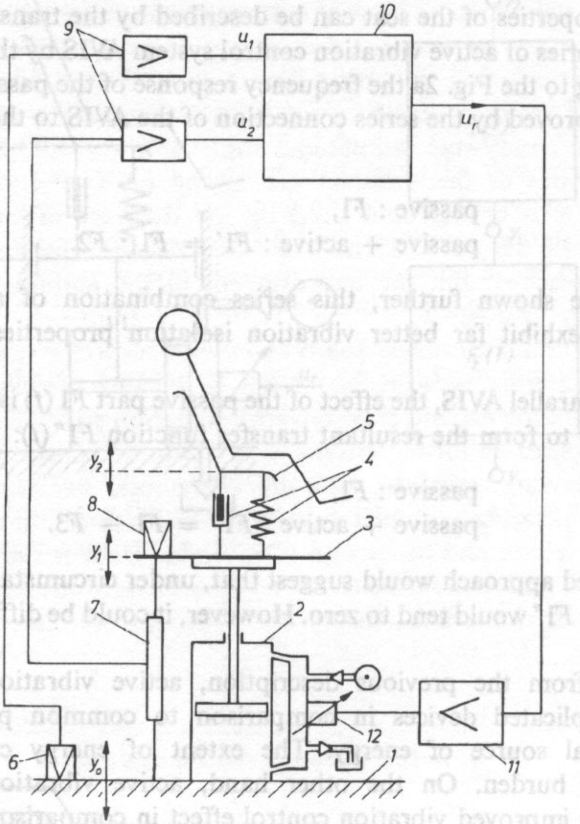

Fig. 3

In the last few years our research group undertook an extensive theoretical and experimental research of such active vibration control system.

Most interesting are the experimental results achieved with common driver seats. The seats tested together with the active system were:

1. German made driver seat of the type ZT 300, used in agricultural tractors. In this seat a torsion rod is used as the resilient element.

2. Czechoslovak made seat of type KAROSA 281.0 with an air spring and mechanico-pneumatic system for middle position control. This is a standard seat used in Czechoslovak made trucks, buses and earth moving vehicles.

For the evaluation of the vibration control properties of these systems following criteria were used:

a) Frequency response characteristics;

b) Power spectral density curves of the vertical acceleration on the seat base cabin floor and the seat cushion;

c) Assessment according to the ISO Standard 7096-82, which is a standard method for evaluation of vibration control properties of driver's seat for earth-moving machines.

d) Assessment according to the ISO Standard 2631/1-85, which describes the assessment of the vibration influence on the seated driver and states the allowed vibration level limits.

The frequency response

The amplitude-frequency response function of the seat ZT 300 alone and with the AVIS is depicted in Fig. 4. We can see, that the seat's resonant frequency is in vicinity of 1.3 Hz and the seat amplifies the vibrations in the frequency band 0.5 Hz – 2.2 Hz.

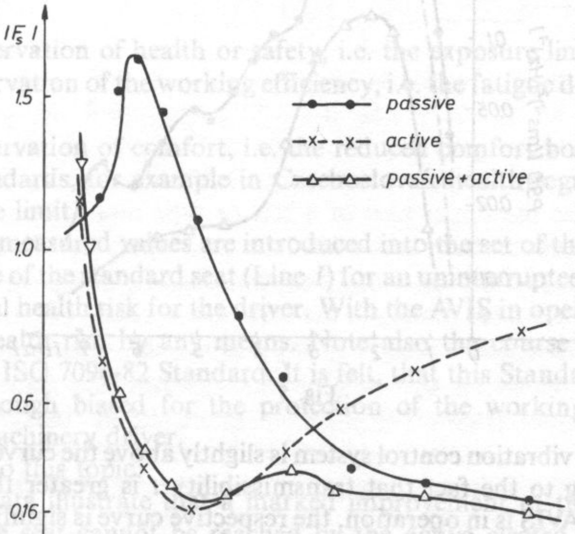

Fig. 4

On the contrary, the series combination of this seat and the AVIS suppresses the vibrations in this frequency band very well and the performance approaches the one of the passive system above 3.5 Hz. This is due to poor dynamic behaviour of the hydraulic cylinder employed. Anyhow, it practically proves that "slow" hydraulic servosystems are well suited for this task. On the other hand, there is a sharp increase below 0.5 Hz, due to chosen cut-off frequency of the electronic controller of 0.15 Hz. Note that the frequency response function of the AVIS alone has a *U*-form, in good agreement with the theoretical analysis and the simulation results.

The acceleration PSD

The acceleration PSD in the frequency range 0.5 – 20 Hz for the excitation according to the 3-rd machine class of the ISO standard 7096 was measured. Typical curves are depicted in Fig. 5. Note, that the course of the vertical acceleration PSD curve on the

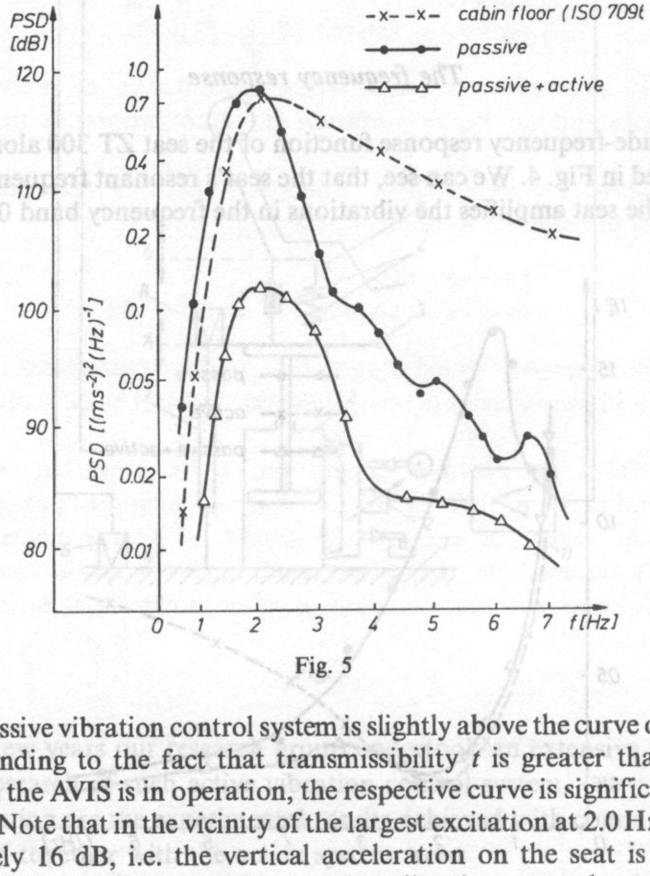

Fig. 5

seat with the passive vibration control system is slightly above the curve of the excitation PSD, corresponding to the fact that transmissibility *t* is greater than one. On the contrary, when the AVIS is in operation, the respective curve is significantly below the former curves. Note that in the vicinity of the largest excitation at 2.0 Hz, the difference is approximately 16 dB, i.e. the vertical acceleration on the seat is approximately 2.8-times lower than in the case of the passive vibration control system only.

Results according to the ISO 7096-82 Standard

Both seats passed the limits set down by the ISO 7096-82 Standard, even when the active part was turned off.

The permissible vibration exposure

The ISO 2631/1-85 Standard distinguishes three main criteria for the vibration influence on the human being (Fig. 6).

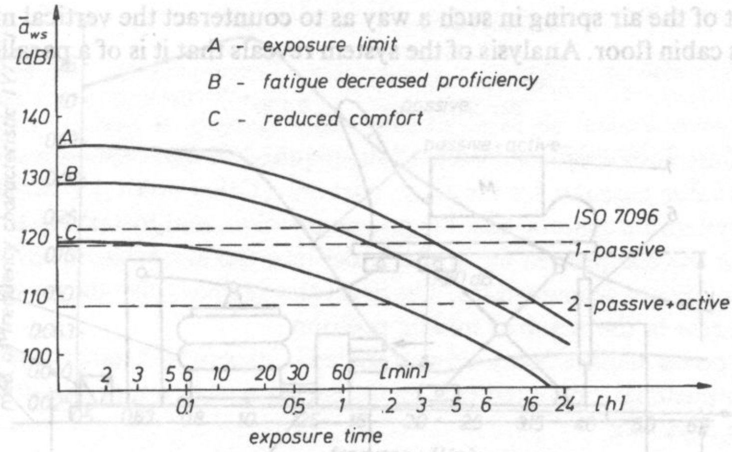

Fig. 6

A) The preservation of health or safety, i.e. the exposure limit;

B) The preservation of the working efficiency, i.e. the fatigue decreased proficiency boundary;

C) The preservation of comfort, i.e. the reduced comfort boundary.

In some standards, for example in Czechoslovak health regulations, the middle curve is absolute limit.

In Fig. 6 the measured values are introduced into the set of these curves. It can be seen, that the use of the standard seat (Line *1*) for an uninterrupted 8-hour-shift brings about a potencial health risk for the driver. With the AVIS in operation (Line *2*) there should be no health risk by any means. Note also the course of the straight line representing the ISO 7096-82 Standard. It is felt, that this Standard is a compromise criterion not enough biased for the protection of the working conditions of the earth-moving machinery driver.

Conclusion to this topic:

The above data illustrate that a marked improvement of the vibration control properties of the seat cannot be reached by the active electro-hydraulic vibration control system alone, but with a series combination of both the passive and active parts. This brings about some disadvantages, the most important being a rather complicated construction (in a real earth-moving machine there is not enough free space to build in the hydraulic cylinder in appropriate position) and also a rather large energy consumption (see later). Therefore this kind of active vibration control system for driver's seat has not found practical use till now.

5. Electro-pneumatic active vibration control system

Some disadvantages of the electro-hydraulic active vibration control system should be suppressed by the pneumatic active vibration control system. Essentially it is working on the principle of proper control of inflow and outflow of compressed air into and out of the air spring in such a way as to counteract the vertical movement of the vehicle's cabin floor. Analysis of the system reveals that it is of a parallel structure.

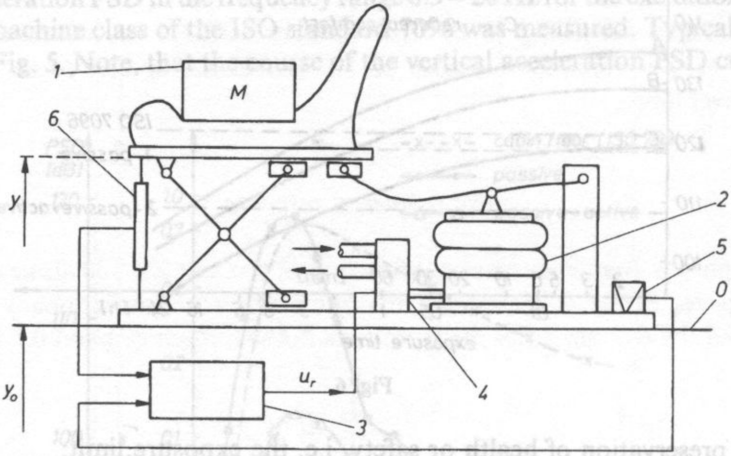

Fig. 7

The electro-pneumatic AVIS consists (Fig. 7) of an air spring *2*, acting through a lever mechanism on the upper part of a scissor type mechanism. It supports the upper part with the seat cushion and with the seated operator of mass $M(1)$, whose vibrations in the vertical direction are to be controlled. The whole system is mounted on the base *0*, which is excited mainly in the vertical direction. The air spring *2* acts as the static load bearing element, passive spring and the active part actuator, all in one structural element. The in-flow and out-flow of the compressed air into and from the air spring into atmosphere is controlled by electro-pneumatic transducer *4*, which in turn is governed by the voltage u_r, generated by the electronic controller *3*. The electro-pneumatic transducer *4* is a proportional type device with sufficiently wide frequency band which, by electrically controled changes in the orifice aperture, governs the flow of compressed air into and out of the air spring. The air spring acts as a force generator and moves through the lever mechanism the upper part with the seated operator in such a way as to conteract the vertical vibrations acting on the base *0*. The system is equipped with relative displacement sensor *6* and the accelerometer *5*.

This system was not studied in such an extent as the electro-hydraulic one. A lot of effort was spent in controller synthesis. It seems, that this crucial part of the system has to be of a rather complex structure to facilitate good vibration control properties of the active part in a suffciently broad frequency band. By theoretical analysis and measurement on a dummy system the transfer function was determined. In Fig. 8 both the whole system frequency response curve and that one of the passive part of the dummy system, obtained when the active sub-system was switched off, are depicted. The difference ilustrates improvement of the vibration control properties of the dummy seat exerted by the active part.

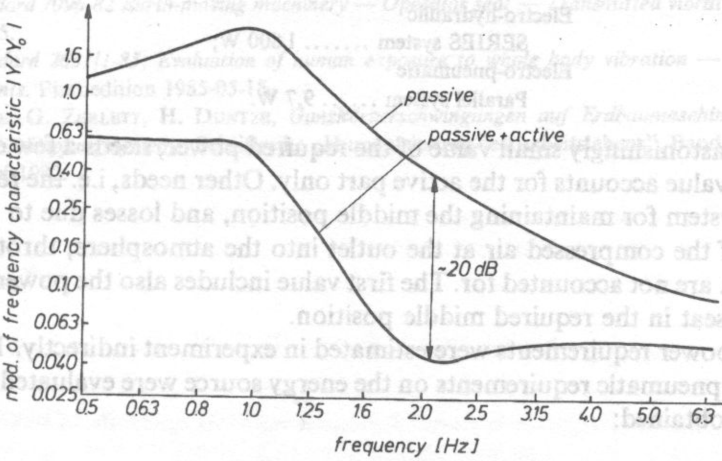

Fig. 8

It could be seen, that the theoretical improvement of the vibration control properties around the frequency of 2 Hz is nearly 20 dB. In realistic systems an improvement by 10–15 dB could be reached, depending on the excitation form and system tuning.

6. The energy consumption of the active vibration control systems

The energy consumption of the active vibration control systems plays very important role in their practical feasibility. For a linear active vibration control system subjected to a stationary random excitation, whose power spectral density is known, it is possible to estimate theoretically the average power necessary for the system's operation. The average power requirements are given by the formula:

$$\overline{P} = \int_0^{\infty} |H_{\Delta\dot{y}}|\,|H_f| \cos(\theta_{\Delta\dot{y}} - \theta_f)\, G_{y_0 y_0}\, d\omega\,,$$

where $|H_{\Delta\dot{y}}|$ — modulus of the frequency response function of the relative velocity of the seat cushion in respect to the seat base (cabin floor), $\theta_{\Delta\dot{y}}$ — phase angle of the above frequency response function, $|H_f|$ — modulus of the frequency response function of the force exerted by the active part only, θ_f — phase angle of the frequency response function of the force exerted by the active part, $G_{y_0 y_0}$ — power spectral density of the vertical displacement of the seat base (cabin floor), ω — frequency.

Inserting numerical values into this formula the theoretical value of the power requirements of the active part only could be estimated. When some realistic values pertinent to the systems used for experimental investigation are introduced, the following theoretical values of average power requirements could be obtained:

Electro-hydraulic
SERIES system .... 1800 W;
Electro-pneumatic
Parallel system ...... 9.7 W.

The second, astonishingly small value of the required power, needs a few explanatory words. This value accounts for the active part only. Other needs, i.e. the requirements of the sub-system for maintaining the middle position, and losses due to air friction, expansion of the compressed air at the outlet into the atmosphere, throttling in the actuator, etc. are not accounted for. The first value includes also the power needed for holding the seat in the required middle position.
The overall power requirements were estimated in experiment indirectly. The average hydraulic or pneumatic requirements on the energy source were evaluated. Following values were obtained:

Electro-hydraulic
Series system .... 2400 W;
Electro-pneumatic
PARALLEL system
— for positional control ...... 800 W;
— for the active part only .... 80 W.

It seems to be in relatively good agreement with the theoretical results.

7. Conclusion

In line with the successive introduction of automation means and microelectronic subsystems into the chassis of various types of vehicles, the perspectives of introduction of active vibration control means increases. These systems could markedly improve the vibration control properties of the driver's seats, and so improve the ride comfort of the driver, decrease driver's health risks and improve the utilisation of the machine.

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