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ACTIVE SYSTEMS IN THE VIBRATION CONTROL OF HEAVY MACHINE DRIVER'S SEATS

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The paper deals with vibration damping properties of passive vibration control means used in driver's seats of heavy earth moving machines.

The significant improvement of vibration control reached by an active vibration control system of an electrohydraulic type is described. The advantages and perspectives of electropneumatic active vibration control systems for driver's seats is stressed.

1. Introduction

In heavy working, earth-moving and building machines as well as in tractors, the seat is the main vibration control equipment protecting the driver-operator. Among the favourable consequences of good vibro-isolation for the driver, emphasis is mainly laid upon the protection of his health. On most of the contemporarily used seats, however, acceleration values lie deeply below the limits of the increased health hazard, defined for example by the international standard ISO 2631. Hence the protection of the driver's health should not be a problem today. It should be a matter of course.

Attention is therefore focussed upon satisfying a more severe criterion. The acceleration values on the seat are required to be below the limit of the driver's increased fatigue, as presented by ISO 2631, by standard SEV and by the national health care regulations.

There still exists a third, significant aspect of the driver's good vibro-isolation. It follows from the linkage between the effective value of acceleration on the seat $a_{sef}[\mathrm{ms}^{-2}]$ and the speed of the vehicle in its direct run $v[\mathrm{ms}^{-1}]$, equat. (1) [14]. The man-driver or the man-operator, respectively, makes use of this linkage because he tends to rise the speed of the vehicle under conditions of good vibration control. Conversely, when insufficiently protected, he will lower the speed until effective acceleration value on the seat drops to an acceptable value. This state, however, often means running at a speed that lies deep below the technical capability od the engine

$$a_{\mathrm{sef}} = k\sqrt{v} \tag{1}$$

where k denotes a coefficient, depending on the construction of the vehicle and seat and on the unevenness of the track.

The real driver seats with a passive or semiactive vibroisolating system are of complex structure. From the stand point of mechanics this complexity is of dual nature. Firstly, the seat has several eigenfrequencies and a corresponding set of cubic eigenfunctions. Secondly, there are some elements in the construction of the seat with nonlinear properties. For example: clearances in the kinematic joints, damping devices etc.

Nevertheless, many real seats appear as a system close to a linear one in a limited frequency band.

In estimating the dynamic properties of a seat a passive or semiactive vibration control system, the cabin base motion being stochastically perpendicular, the amplitude characteristic $|F_S|$ Fig. 1 is decisive. Its significance is related to the fact that at a given power spectrum density of the acceleration of floor vibration $\Phi_B[(\mathrm{ms}^{-2})^2/\mathrm{Hz}]$ it determines the power spectrum density of acceleration on

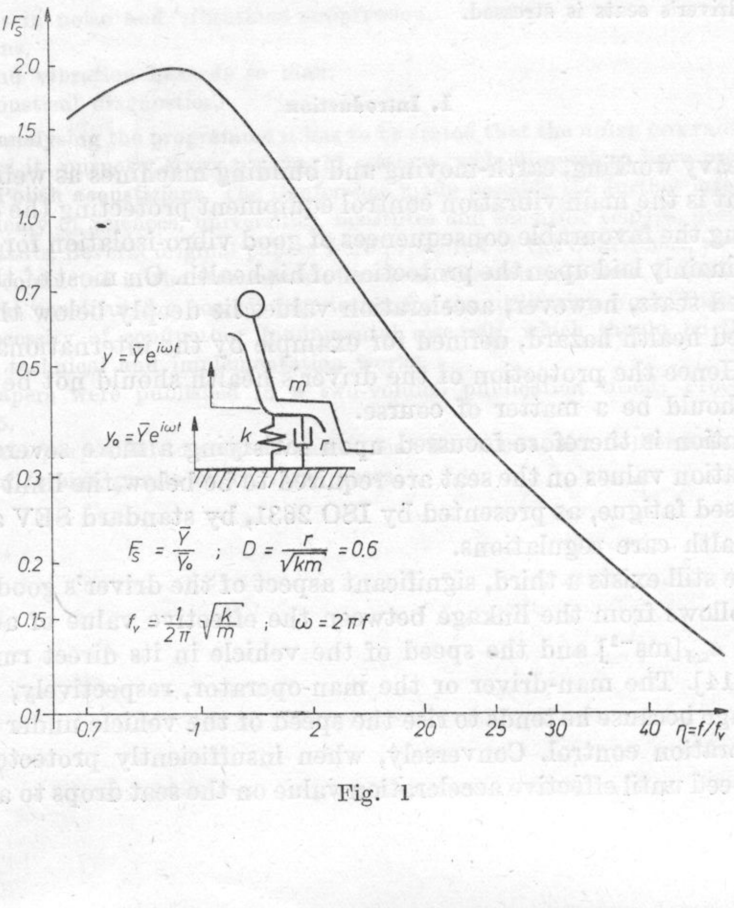

Fig. 1

the seat saddle $\Phi_S\,[(\mathrm{ms}^{-2})^2/\mathrm{Hz}]$, in terms of the relation

$$\Phi_S = |F_S|^2 \Phi_B. \tag{2}$$

Some results calculated according to this equation, are presented in Fig. 2.

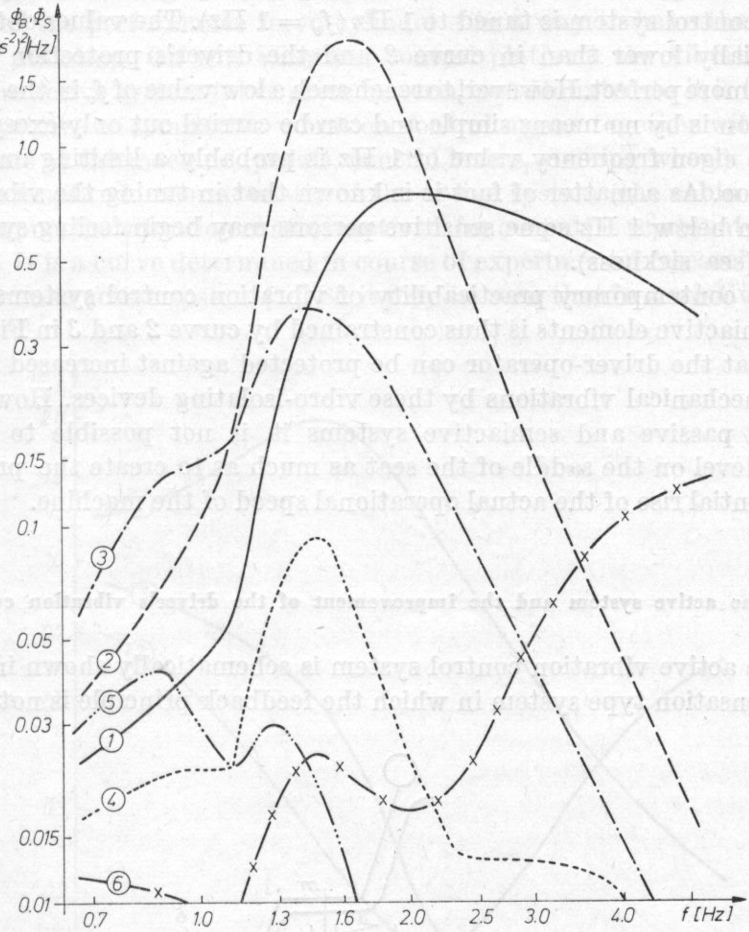

Fig. 2

Curve *1* on this figure shows the typical behaviour of the power spectrum density of perpendicular acceleration on the cabin floor. It is contained in the recommendation of ISO 7092 [12] for the 3rd class of earth movers, i.e. short-base machines with wheel undercarriage.

The power spectrum density of the acceleration of saddle vibration under passive vibration control is represented by curves *2* and *3*.

Curve *1* illustrates the situation when the eigen frequency of the passive vibration control system is $f_v = 1.5$ Hz. Many seats used at present have a vibration control system tuned to that value. They usually satisfy the criteria of the ISO 7096 standard, hence they are considered good.

Curve *2* shows, however, that in the ferquency band up to 2 Hz the power spectrum density of acceleration is higher on the saddle than on the floor, even if the area beneath the curve, that is the effective value of acceleration, may be lower than on curve *1*.

Curve *3* shows the behaviour of the power spectrum density when the vibration control system is tuned to 1 Hz ($f_v = 1$ Hz). The values obtained here are essentially lower than in curve *2* and the driver's protection is also intrinsically more perfect. However, to reach such a low value of f_v in the actual seat construction is by no means simple and can be carried out only exceptionally.

The eigen frequency value of 1 Hz is probably a limiting one on other grounds, too. As a matter of fact it is known that in tuning the vibration control system below 1 Hz some sensitive persons may begin feeling symptoms of kinetosis (sea sickness).

The contemporary practicability of vibration control systems with passive or semiactive elements is thus constrained by curve *2* and *3* in Fig. 2. It can be seen that the driver-operator can be protected against increased fatigue caused by mechanical vibrations by these vibro-isolating devices. However, with the aid of passive and semiactive systems it is not possible to lower the vibration level on the saddle of the seat as much as to create the prerequisites for an essential rise of the actual operational speed of the machine.

2. The active system and the improvement of the driver's vibration control

The active vibration control system is schematically shown in Fig. 3. It is a compensation type system in which the feedback principle is not utilized in

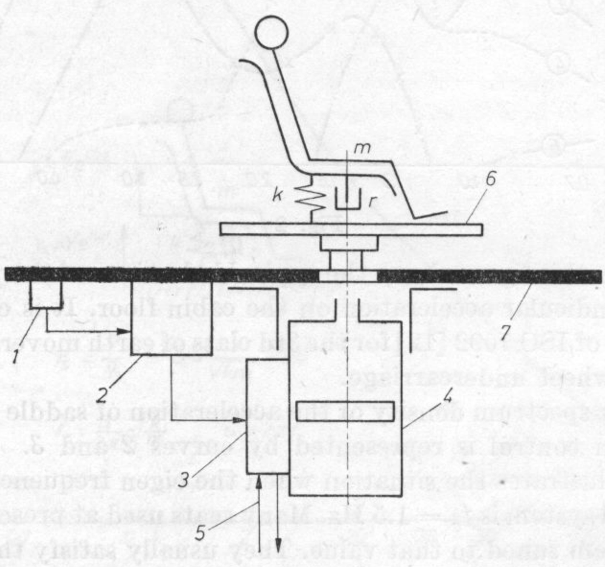

Fig. 3

the frequency band above 0.5 Hz. This improves the noise conditions in the system, removing the problems connected with its stabilization.

To the floor of a working machine *7* an accelerometer is mounted *1*. The signal from this sensor, adapted in the electronic regulator *2* is passed to the transducer *3* controlling the inflow and outflow of the working medium *5* into the organ of the performance servo-cylinder *4* in a way to have the platform *6* practically standing. On this vibration control platform the vibration control effect of the active system takes place materializes. Mounted to it is the driver's seat with passive or semiactive vibration control means together with the control elements of the machine (pedals, control levers, steering wheel).

The amplitude characteristic of the active system (the cabin floor being taken for input, the vibration control platform for output) is represented by curve *2* in Fig. 4. It is a curve determined in course of experimental research into a real electrohydraulic active system [1]. So it may be seen that in a relatively narrow

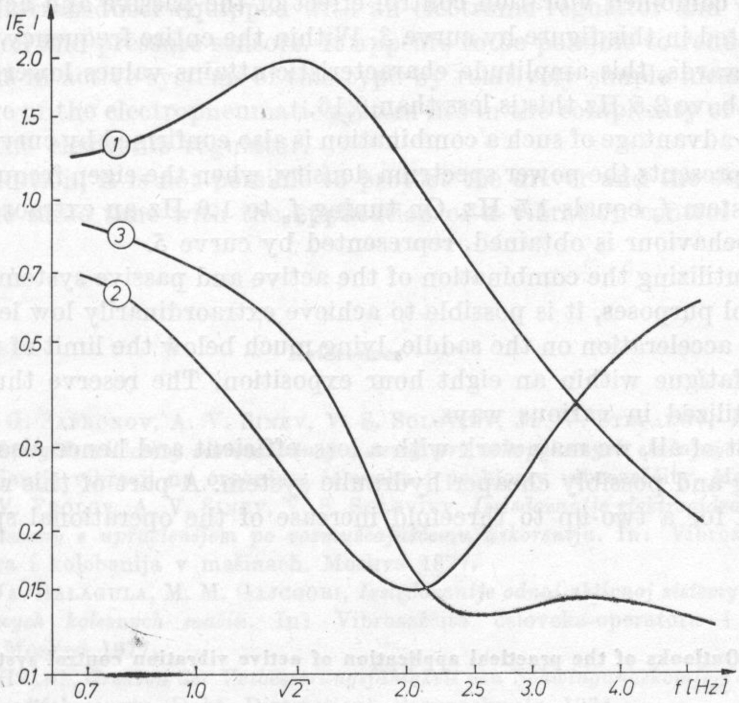

Fig. 4

frequency band around 2 Hz very favourable, low values are attained. However, the amplitude characteristic rises both towards low and high frequencies. Towards the low values this is caused by the real properties of the electronic regulator, towards the high ones by the properties of the hydraulics.

From the viewpoint of vibration control especially the rise of the characteristic towards high frequencies is very unfavourable, because in the frequency band from 4 up to 8 Hz, where the human body's sensitivity to vibration is greatest, the vibration control effect of the active system is considerably limited. This fact is illustrated by curve *6* in Fig. 2. The curve indicates a high power spectrum density value on the seat saddle within the band above 4 Hz.

This unfavourable property of the active system might be restricted by improving the dynamic properties of the hydraulics. At the same time, however, its price would rise and – this being extraordinarily negative effect – its power demands, too.

Another way of improving the properties of an active system is to link it up with a passive vibration control system. Figure 3 indicates a situation, where a seat with a passive vibration control is mounted to the vibration controlled platform. Considering a currently used seat whose eigen frequency, f_v, is about 1.5 Hz, curve *1* in Fig. 4 has been calculated in order to represent its amplitude characteristic.

The combined vibration control effect of the passive and active system is represented in this figure by curve *3*. Within the entire frequency band, from 0.2 Hz upwards, this amplitude characteristic attains values lower than 1. In the band above 2.5 Hz this is less than 0.16.

The advantage of such a combination is also confirmed by curves in Fig. 2. Curve *4* represents the power spectrum density, when the eigen frequency of the passive system f_v equals 1.5 Hz. On tuning f_v to 1.0 Hz an extraordinarily favourable behaviour is obtained, represented by curve *5*.

By utilizing the combination of the active and passive systems for vibration control purposes, it is possible to achieve extraordinarily low levels of perpendicular acceleration on the saddle, lying much below the limit of the driver's increased fatigue within an eight hour exposition. The reserve thus obtained may be utilized in various ways.

First of all, we may work with a less efficient and hence also less power demanding and possibly cheaper hydraulic system. A part of this reserve may be utilized for a two-up to threefold increase of the operational speed of the machine.

3. Outlooks of the practical application of active vibration control systems

The properties of active systems and the feasibility of their application have already been examined by several authors [1–10]. Most of them presumed that the executive element of the active system would be a hydraulic servocylinder completed by an electrohydraulic linear transducer controlling the transflux of the fluid into and from the servocylinder.

The advantage of this conception lies in its relative simplicity, especially of its electronic part. It may be expected that subsequent development will allow the reduction of the system power demand from the contemporary 3 up to 5 kW to hundreds of Watts by order of magnitude.

The disadvantage of the electrohydraulic variant lies in the fact that the active portion of the vibration control system is to be combined with a current seat with a passive vibration control, as it is schematically shown in Fig. 3. This increases both the price and the sophistication of the system. Further, the electrohydraulic transducer is a relatively delicate device requiring, in addition to other measures, also sufficient oil filtration.

According to the survey paper [10], based on analog modelling, some of the flaws of the electrohydraulic system could be removed if a pneumatic executive organ would be used. An especially significant is fact that both active and passive protection can be implemented by a single air spring in which pressure would vary in order to compensate the exciting effects from the cabin floor. This implies that the contemporary design of the driver's seat with an air spring would remain unchanged. The air spring would only be linked to an electropneumatic transducer equipped with an electronic regulator and with a set of acceleration and pressure sensors. It appears to be possible to reduce the power demands in active systems of this type by relatively simple measures. The disadvantage of the electropneumatic system lies in the complexity of the transducer and the electronic regulator.

In addition, it is not possible to protect the driver and the steering elements at the same time with the application of a vibration control system of this kind.

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