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CONTROL PROBLEM IN A MECHATRONIC VIBRATION ISOLATOR

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Abstract

A new mechatronic vibration isolation system has been developed on the basis of system theory. The methods of this theory allow the typical mechanic reference problem in measuring and control easier to manage. This new mechatronic equipment integrates mechanics, pneumatics and electronics in synergetic co-operation. The electronic control is responsible for the dynamic behaviour and the special computing transfer block for the real time interaction and compensation of floor vibration disturbances.

Keywords: mechatronics, vibration isolation, compensation of disturbances.

1. Introduction

The method of active compensation of the disturbances influences is not new in the control science and in the mechatronics. There may be, however, several new fields and examples in the application of this method and the paper tries to give a contribution especially for the finding of optimal transfer functions for the blocks of real-time interaction on the basis of the system science.

The problem is even more complicated if there is a mixed technical system to be controlled as it is often the task in the mechatronics. In the presented plant there work mechanical, electrical and pneumatic systems synergistic integrated with electronics in each other.

2. About the Demonstration Example

The active vibration isolator for the microtechnics represents a typical synergetic system. The correct operation of precision instruments and measuring equipment and the processes of technology in the micromechanics and microsystem technology can be disadvantageous influenced by environment vibrations. The *Fig. 1.a* illustrates that the different types of environmental disturbances and sources of vibration cannot be described and eliminated with simple spring–mass systems and models. One of the main problems is the insufficiency of the attainable damping in the critical frequency range of the building vibrations (0.1 - 10 Hz) since the well-known commercial equipment cannot reach the desired vibration levels on the protected instruments, see for example in *Fig. 1.b*.

3. Classification of Vibration Isolation Methods

First of all some *new expressions* are to be explained on the basis of the *system science*. The equipment for vibration isolation work in a *passive* way like damped spring–mass systems if there is only a flow of material and/or energy in them. Systems are *active*, if they are controlled and there is also a direct compensation of disturbances of the environment and there is a flow of information beside of the flow of material and/or energy. According to this classification the conventional vibration isolators with pneumatic control for the position of the table are *semiactive systems*.

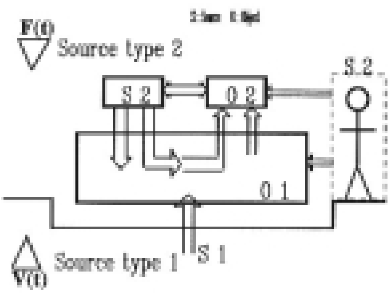

Fig. 1.a.

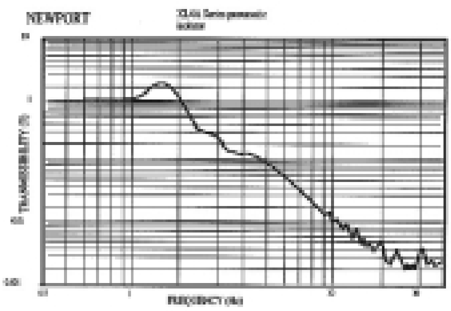

Fig. 1.b.

In passive vibration isolation systems or in systems with semiactive position control the mass of the equipment is supported generally by a damped elastic layer or by real springs. These springs are usually metallic, polymeric or pneumatic.

There are two groups of excitations on the protected equipment:

- Vibrations of the floor show the character of sources *type 'across-variables'*. The excitation by the floor shows the character of a velocity source that means: the floor has a permanent free motion related to the absolute Earth-reference (*Fig. 1.a*). But the mass of equipment has also the same reference. This is why the equation of motion for the velocity of mass shows the qualities of a *serial resonant system*. It behaves like a low-pass filter and this is the reason why such systems let the low-frequency vibration through the protected instrument.

- Excitations like the internal and external forces on the protected mass are sources *type ‘trough-variables’*. For these excitations the system works like a band-pass filter damping the vibrations except in the resonance band.

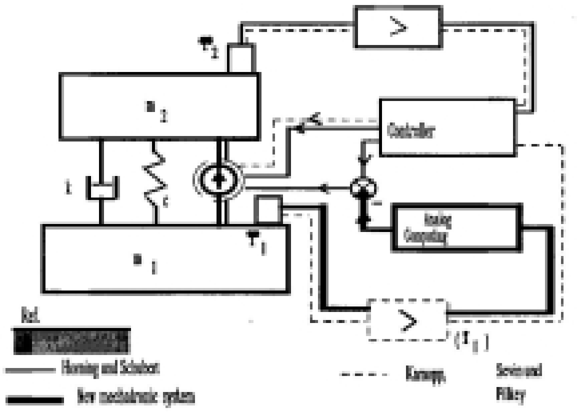

Fig. 2.a.

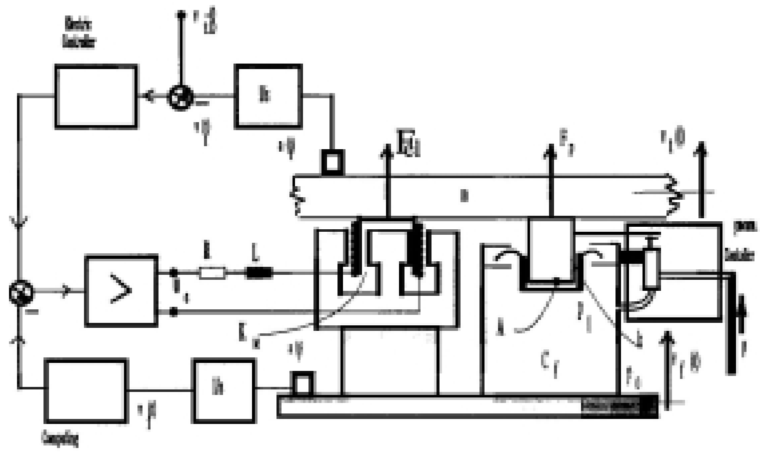

Fig. 2.b.

4. A New Conception for Active Interaction

HORNING and SCHUBERT [2], KARNOPP [3], SEVIN and PILKEY [4] suggested a control system with force actuator. The input signals of the controller are the accelerations of the isolated table and in special cases of the floor as well. The essential differences between typical methods for active vibration isolation and the new developed mechatronic system are shown in the *Figs.* 2.a, 2.b.

In the new system the output signal of the controller and of the computing block are added. This added signals control the force generator to eliminate the motion of the mass related to the absolute reference. This is why it is important to measure the absolute motion (velocity) of mass and floor. The solution of a similar problem is given in [5].

The conventional types of isolators have pneumatic springs with one or two chambers in each of their feet. The relative motion of table and foot regulates the air flow into the upper chamber. There is sometimes an adjustable valve between the upper and lower chambers to increase the dissipated energy. This control system can, however, only compensate the influence of the change of the static load. This is also an important property as well but the transfer function shows that the conventional vibration isolators also work as a low-pass filter if the source of excitation is the velocity of floor.

5. Dynamic Behaviour

To clear the dynamic problem it is sufficient to observe the one-dimension mathematical model of the system [1], [7], [8]. On the base of the dynamic model of the conventional isolators [1] the following transfer functions can be written:

$$\frac{V_t}{V_f} = T_v = \frac{b_3 s^3 + b_2 s^2 + b_1 s + 1}{a_4 s^4 + a_3 s^3 + a_2 s^2 + a_1 s + 1} \tag{1}$$

and

$$\frac{X_t}{F_t} = T_F = \frac{b_2 s^2 + b_1 s}{a_4 s^4 + a_3 s^3 + a_2 s^2 + a_1 s + 1}. \tag{2}$$

The parameters a_n, b_n in the equations are calculated with the technical system parameters of the commercial equipment. V means velocity, X means displacement and F means force [1].

Regarding the *Eq.* (1) it is obvious that the velocity of floor will be transmitted at low frequencies and in the resonance band to the table. The change of the static load can be eliminated as the force transfer function, the *Eq.* (2) shows.

A *new way for the solution* is suggested on the base of the construction shown in *Fig.* 2.b.

The picture showing the mass of the equipment (table) is supported by three pneumatic springs with control system for the compensation of the static load change. The mechanical-pneumatic system is completed by a dynamic force generator which leads the necessary energy for the compensation of the dynamic forces occurred by the acceleration of the table. These forces will be compensated by a non contact force actuator through the point marked with a circle in the *Fig.* 3.b.

The *Fig. 3.a* shows the graph and *Fig. 3.b* shows the signal flow in the new type of plant. The control signal for this electrodynamic actuator (linear motor) has to be calculated from the *motion signals of the table and of the floor at the same time.*

The state equations can be written in form of a signal flow. With a circle is marked the independent variable u_e (electric input voltage of the linear motor) in this signal flow. Also the important effects of disturbances given by the physical equations of motions are represented in this signal flow on three different points of the mechatronic part-system.

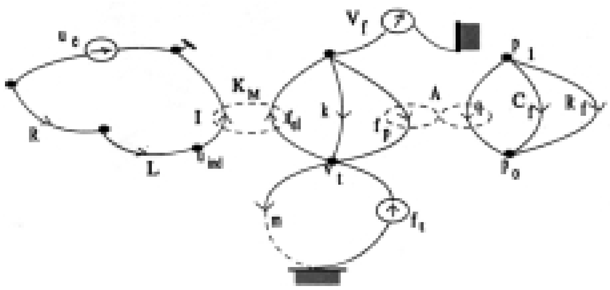

Fig. 3.a.

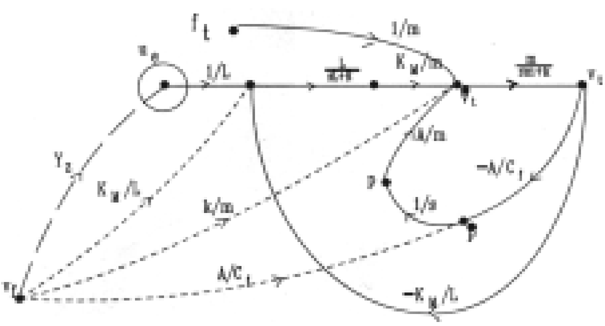

Fig. 3.b.

In the simplified one-dimension model with vertical motions there are three independent velocities to observe: The reference velocity zero of the Earth, the velocities of the floor and of the mass of the equipment. The mechanical model can be of course extended to a higher order system (usually one vertical translation and two or four rotations depending on the real construction). Further information about this can be found in [1].

6. Determination of the Computing Block Transfer Function

The dynamic behaviour of the new mechatronic system is, however, easier to observe on the one-dimensional motion model. *Fig. 3.b* shows that because of the physical reality the absolute velocity of floor influences three variables of the system.

The base for the calculation algorithm of the computing block in the new mechatronic control system is given by two transfer functions:

1. The velocity of floor acts on three physical ways on the output signal v_t (velocity of table). v_t influences the change of pressure in the chamber, the acceleration of table and the change of current in the linear motor. These effects can be assumed by the transfer operation Y_{phys} essential as well for the computation of compensation:

$$Y_{\text{phys}} = \frac{v_t}{v_f} = \frac{m}{sm+k} \cdot \frac{\dfrac{K_M^2}{smL+mR} + \dfrac{k}{m} + \dfrac{A^2}{smC_f}}{1 + Y_1 + \dfrac{A^2}{s^2mC_f + skC_f}}, \tag{3}$$

with

$$Y_1 = \frac{K_M^2}{s^2mL + s(mR+kL) + kR}. \tag{4}$$

2. On the other way the output can be influenced also by the input voltage u_e of the new suggested actuator:

$$Y_{\text{el}} = \frac{v_t}{u_e} = \frac{\dfrac{K_M}{s^2mL + s(mR+kL) + kR}}{1 + Y_1 + \dfrac{A^2}{s^2mC_f + skC_f}}. \tag{5}$$

The operation Y_Z between v_f and u_e *represents a suggested method to achieve the zero velocity on the table (equipment).*

The physical effects can be added in linear systems. It is a computing block with the transfer function Y_Z to be developed that produces according to the following operation the zero level velocity on the table.

$$v_t = (-1) \cdot Y_Z \cdot Y_{\text{el}} \cdot v_f + Y_{\text{phys}} \cdot v_f = 0, \tag{6}$$

where

$$Y_Z = Y_{\text{phys}} \cdot (Y_{\text{el}})^{-1}. \tag{7}$$

After substitution of the transfer functions Y_{phys} and Y_{el} the mathematical operation of the computing block can be calculated. Y_Z is given by the following equation:

$$Y_Z = K_M + \frac{k}{K_M} \cdot \frac{sL+R}{sT+1} + \frac{A^2}{C_f K_M} \cdot \frac{sL+R}{s} \qquad \text{with } T \to 0. \tag{8}$$

The optimal dynamic behaviour of the equipment is based on the usual control method shown in the *Fig. 4*. The parameters for the different gains were calculated by looking for optimum with the integral minimum method.

7. Computer Simulations of the Dynamic Behaviour and Control Measuring

After testing the one-dimension mathematical model by computer simulation also the complete system was analysed. To compare the simulated results with the real system a model was built. Because of the high costs of the measuring system only the most important perpendicular motions have been controlled on this model. The compensation effect could be verified like *Fig. 6* shows. Measuring in different buildings have shown that there are two types of floor excitations to observe: The main table motions ($v_Z \equiv v_S$; Ω_X; Ω_Y) result in consequence of the perpendicular velocity amplitudes of the floor. The other three motions (Ω_Z; v_X; v_Y) are usually hindered by the construction of the vibration isolators.

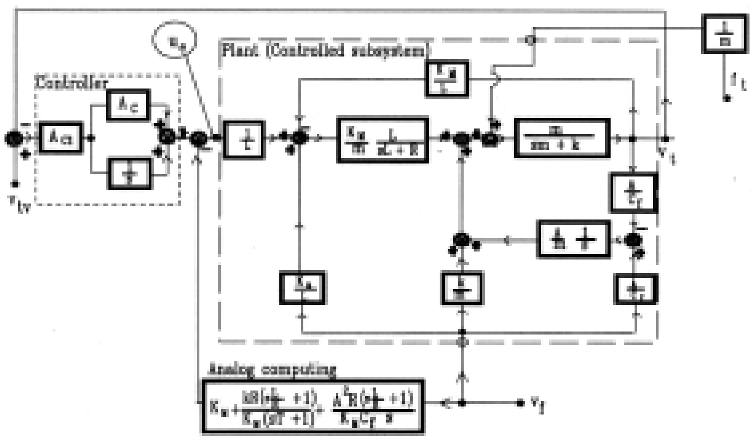

Fig. 4.

The *number of the controlled state variables of the table depends always on the actually given technical problem*. In this paper only the *main motions* (one vertical translation and two rotations around the horizontal axes) of the table are presented.

This main-motions-complete subsystem of main motions has *11 state variables*:

$$\underline{u}^T = [v_S, \Omega_X, \Omega_Y, i_1, \dots i_4, p_1, \dots p_4].$$

The four controllers have 4 further state variables. The control system is accordingly a 15 order system. Also the 8 state variables of the computing blocks are to be added to them. The simulation model consists of *23 linear first order differential equations* beside the numerous algebraic equations. Because of the values of the time constants a suitable simulation language was claimed. So it was found that the Meerman Automation's TUTSIM program [9] for engineering design and determination of optimal operation of continuous dynamic systems satisfies the demands.

7.1. Simulation Results

Some characteristic results of simulations are presented in the next figures. The time functions of the floor velocity below the foot Nr 1 (function 5) and of the velocities on the table above them (functions 1 – 4) are shown in the *Fig.5.a if the computing block is not working*. For the sinusoidal excitation at 2 Hz frequency the velocity amplitudes of floor appear undamped on the table. After *contacting the computing blocks the velocity levels decrease* spectacularly about 60 dB as it is shown in *Fig. 5.b*.

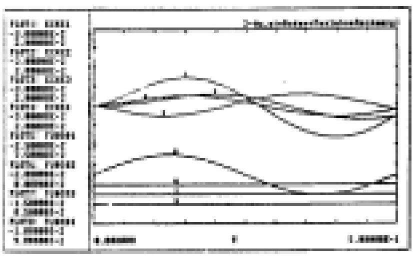

Fig. 5.a.

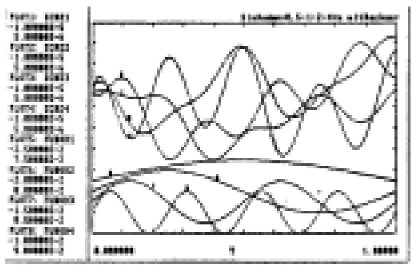

Fig. 5.b.

It is obvious, that these simulation results could be reproduced on an experiment model only with high efforts in technical equipment and measuring. The accuracy is limited first of all through the measuring system but also the accuracy of the computing blocks. This predicted effect has been verified by the measuring on an experiment model.

On the one-dimension vertical model the main problem was to detect the very low level velocities at low frequencies. High precision accelerometers (Hottinger) with integration blocks and seismic electrodynamic velocity sensors worked parallel to compare the efficiency of both measuring methods.

7.2. Results of Control Measuring

Fig. 6 shows *the measured result of the switch on process of the computing block*. The diagram verifies the property of the suggested new compensation method.

About 26 dB damping could be measured between the velocity amplitudes of the floor and of the table.

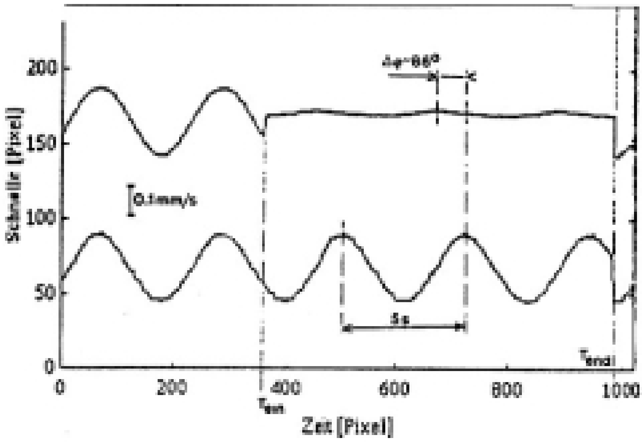

Fig. 6.

How the computing block works is presented in the last two figures. The compensation transfer function Y_Z consists of two important operations.

The accuracy of computation is determined at low frequencies below 1 Hz by the measuring of the absolute motion and above 1 Hz by the measuring of the absolute velocity of the floor. This behaviour can be followed in the *Fig. 7.a* and *7.b*.

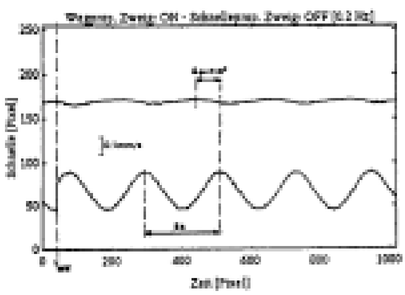

Fig. 7.a.

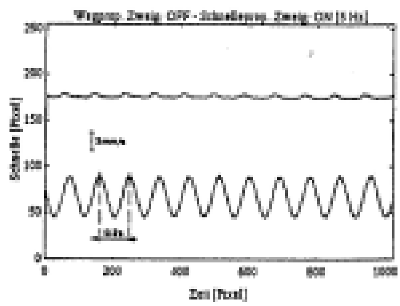

Fig. 7.b.

Summarising the results of the computer simulations and of the measuring it can be found that by application of the system science methods a high efficiency active vibration isolation equipment with direct compensation of disturbances could be realised for the microtechnics. Furthermore this working model is a proof for efficiency of the real-time interaction to compensate the effect of external disturbances.

It is shown that the use of the methods of system science makes the finding of the optimal transfer function for interaction easier.

References

[1] HUBA, A.: Mechatronischer Schwingungsisolator für die Mikro- und Ultrapräzisionstechnik. Dissertation. TU Ilmenau. 1994.

[2] HORNING, R. W. – SCHUBERT, D. W.: Shock and Vibration Handbook (Ed.: Harris, C.M.) McGraw-Hill Book Comp. 1988.

[3] KARNOPP, D. C.: Active and Passive Isolation of Random Vibration, *ASME, Design Engineering Conf.* Cincinnati. 1973.

[4] SEVIN, E. – PILKEY, W. D.: Optimum Shock and Vibration Isolation, The Shock and Vibr. Center, US Dept.of Defence. 1971.

[5] WELTIN, U. : Aktive Schwingungskompensation bei Verbrennungsmotoren, VDI-Ber. Reihe 12. No 179. Düsseldorf,. 1993.

[6] HUBA, A. – VARGA, A.: Aktív rezgéscsillapítás (Active Vibration Isolation) *Finommechanika-Mikrotechnika*, 28/3. 1989.

[7] KOLEV, E. – JUST, E.: Simulation des Schwingungsverhaltens gekoppelter Schwinger der Feinwerktechnik. *Finommechanika-Mikrotechnika*, 1992/5-6.

[8] CANNON, R. H. Jr.: Dynamics of Physical Systems, McGraw-Hill Book Comp. 1967.

[9] TUTSIM for IBM's PC, 1990. Meerman Automation, Neede, Holland.