RESEARCH INTO THE EFFECTIVENESS OF OPERATION OF THE SEAT SUSPENSION SYSTEM USED TO PROTECT INDUSTRIAL AND CONSTRUCTION MACHINERY OPERATORS FROM VIBRATIONS

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Summary

The paper deals with research into the effectiveness of operation of a seat suspension system as a means to protect industrial and construction machinery operators from vibrations. The basic categories of the sources of mechanical vibrations that disturb the functioning of heavy machinery have been discussed. The root mean square acceleration values of the vibrations to which the operators of the most popular heavy machinery are exposed during their work have also been given. The input signals have been determined that induce seat vibrations and whose spectral characteristics are representative for the excitations encountered in cabins of industrial and construction machinery of different kinds. A biomechanical model of the human body has been developed as a discrete mechanical system with many degrees of freedom and with viscoelastic links. The vibro-isolating properties of a semi-active and active seat suspension system with a scissor linkage mechanism used to guide the isolated object were investigated within the work carried out. The general forms of mutually opposing vibro-isolation criteria have been given, which were used to evaluate the effectiveness of operation of the seat suspension systems. Moreover, the vibro-isolating properties of a semi-active and an active seat suspension system were determined from simulation tests where the biomechanical model of the human body was applied and from experimental tests carried out with participation of seated humans and these data have been presented in the final part of this paper.

Key words: vibration, seat, seat suspension system

1. Introduction

The sources of mechanical vibrations that disturb the functioning of industrial and construction machinery may be classified into two basic categories. The first one covers the systems that generate vibrations in result of the process they run by themselves,

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as it is in the case of e.g. the systems exposed to vibrations generated by their internal combustion engines, which are often used to drive such machines. The other category covers the systems where vibrations are caused by external factors, e.g. vibration of the operator cabin of a machine moving on an uneven ground surface [3, 10]. Due to the exposure of heavy machines to a variety of vibration sources, the machine operators undergo periodic, almost-periodic, or random excitations [2].

In most cases, the vibrations are harmful processes, having a detrimental impact on the human operator of a machine. The danger arising from the exposure of a human body to vibrations increases in the case of high-intensity vibrations with a prolonged duration period. Fig. 1 gives the root mean square (RMS) acceleration values of the vibrations to which the operators of the most popular heavy machinery are exposed during their work [1]. The data presented are only to visualize the ranges of accelerations of the vibrations that induce operators' movements.

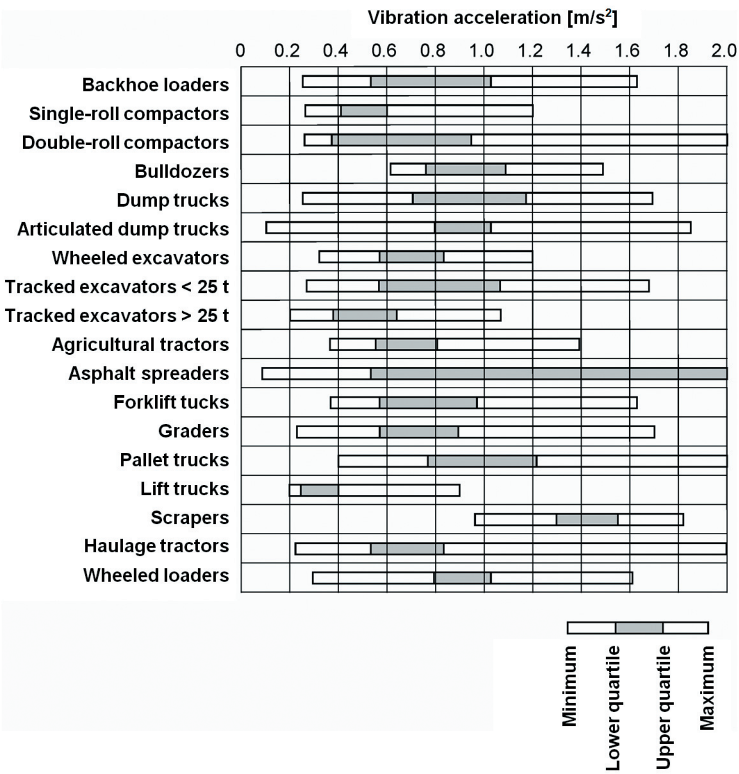


Fig. 1. Root mean square vibration acceleration values obtained from more than twenty measurements carried out for each of the machine types [1]

2. Model waveform representing the vibration excitement

The nature of the vibrations acting on machinery operators depends on the operations performed by a specific machine; as a rule, however, they are random vibrations with frequencies ranging from 1 Hz to 20 Hz [7]. To generate the inputs representative for different machine types, a continuous signal (rand) generator was used, which generated stochastic signal with uniform distribution:

$$\ddot{q}_{sz}(t) = a_{\max}\,\mathrm{rand}(t_k) \qquad (1)$$

where: $\ddot{q}_{sz}(t)$ is the time history of acceleration of the input vibration, $a_{\max}$ is the maximum value of the vibration acceleration signal generated, t is the time of a specific instant, and t_k is the time of duration of the observation period.

The generated signal $\ddot{q}_{sz}(t)$ should be filtered for the spectral characteristics of the input vibration acceleration to be appropriately shaped. The spectral classes of the excitation signals coming from the machine cabin floor and acting on operator's body in the vertical direction are regulated by an international standard [4]. They have been defined for different machine types and the time histories of the exciting vibrations should be reproduced in laboratory conditions and used as inputs to induce the motion of the vibration reduction systems under tests. Fig. 2 shows the power spectral densities (PSD) of the input signals representative for the following machine categories:
EM3 – wheeled loaders with curb mass exceeding 4 500 kg;
EM5 – wheeled bulldozers;
EM6 – tracked bulldozers and loaders.

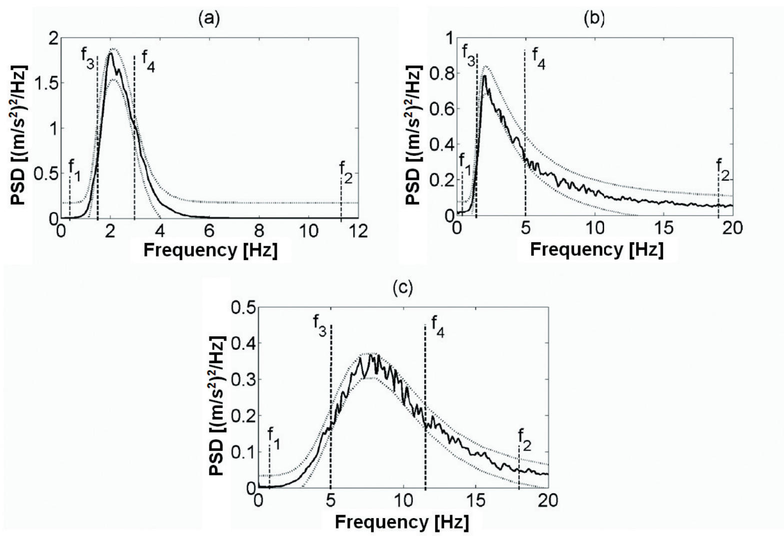
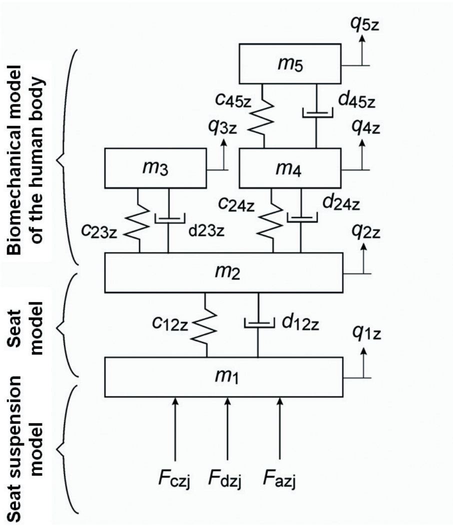


Fig. 2. Power spectral densities (PSD) of the input vibration accelerations (solid line) for signal classes EM3 (a), EM5 (b), and EM6 (c); tolerance zones for the spectral characteristics of signals according to [4] (dotted line)

3. Biomechanical model of a human body to ISO 5982

International standard ISO 5982 [5] defines a simplified biomechanical model where the human body is represented as a discrete system made up of four components connected together with viscoelastic links. The structure of such a model, which has been defined for a seated working human, has been illustrated in Fig. 3. Moreover, the values of masses of individual model components and of the stiffness and damping coefficients of the model links as recommended in the standard quoted have also been specified.



Parameters of the biomechanical model (to ISO 5982):
m_2 = 2 kg
m_3 = 30 kg (for a human body mass of 55 kg)
m_3 = 56 kg (for a human body mass of 98 kg)
m_4 = 6 kg
m_5 = 2 kg
d_{23} = 1 390 Ns/m
d_{24} = 387 Ns/m
d_{45} = 234 Ns/m
c_{23} = 36 200 N/m
c_{24} = 9 999 N/m
c_{45} = 34 400 N/m

Parameters of the seat model:
m_1 = 20 kg
d_{12z} = 868 Ns/m
c_{12z} = 109 000 N/m

Fig. 3. Biomechanical model of a human body normalized for the vibration acting in the vertical direction on a seated human, according to [5] (the meaning of the symbols used has been explained in the text)

The equations of motion of the system (fig. 3) may be written in the following matrix form:

$$\mathbf{M}\ddot{\mathbf{q}}_z + \mathbf{D}\dot{\mathbf{q}}_z + \mathbf{C}\mathbf{q}_z = \mathbf{F}_{sz} + \mathbf{F}_{az} \tag{2}$$

where: $\mathbf{M}$ is the diagonal matrix of masses, $\mathbf{q}_z$ is the vector of displacements of individual system components along the vertical direction of vibration impact, and $\mathbf{F}_{sz}$ and $\mathbf{F}_{az}$ are vectors of the exciting and controlling forces, respectively. The components of the matrices and vectors are as follows, as appropriate:

$$\mathbf{M} = \begin{bmatrix} m_1 & 0 & 0 & 0 & 0 \\ 0 & m_2 & 0 & 0 & 0 \\ 0 & 0 & m_3 & 0 & 0 \\ 0 & 0 & 0 & m_4 & 0 \\ 0 & 0 & 0 & 0 & m_5 \end{bmatrix}, \quad \mathbf{q}_z = \begin{bmatrix} q_{1z} \\ q_{2z} \\ q_{3z} \\ q_{4z} \\ q_{5z} \end{bmatrix}, \quad \mathbf{F}_{sz} = \begin{bmatrix} F_{sz} \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix}, \quad \mathbf{F}_{az} = \begin{bmatrix} F_{az} \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} \tag{3}$$

The symmetric matrices with the damping and stiffness coefficients ($\mathbf{D_z}$ and $\mathbf{C_z}$, respectively) of individual system links may be written in the following form:

$$\mathbf{D_z} = \begin{bmatrix} d_{12z} & -d_{12z} & 0 & 0 & 0 \\ -d_{12z} & d_{12z}+d_{23z}+d_{24z} & -d_{23z} & -d_{24z} & 0 \\ 0 & -d_{23z} & d_{23z} & 0 & 0 \\ 0 & -d_{24z} & 0 & d_{24z}+d_{45z} & -d_{45z} \\ 0 & 0 & 0 & -d_{45z} & d_{45z} \end{bmatrix} \tag{4}$$

$$\mathbf{C_z} = \begin{bmatrix} c_{12z} & -c_{12z} & 0 & 0 & 0 \\ -c_{12z} & c_{12z}+c_{23z}+c_{24z} & -c_{23z} & -c_{24z} & 0 \\ 0 & -c_{23z} & c_{23z} & 0 & 0 \\ 0 & -c_{24z} & 0 & c_{24z}+c_{45z} & -c_{45z} \\ 0 & 0 & 0 & -c_{45z} & c_{45z} \end{bmatrix} \tag{5}$$

In the case of using a system to reduce vibrations with non-linear forces F_{sz} inducing the system motion and non-linear forces F_{az} controlling the system operation, the forces may be described in the general form as follows:

$$F_{sz} = \sum_{j=1}^{k} F_{dzj}\left(\dot{q}_{1z} - \dot{q}_{sz}\right) + \sum_{j=1}^{k} F_{czj}\left(q_{1z} - q_{sz}\right) \tag{6}$$

where: F_{czj} and F_{dzj} define characteristics of the conservative and dissipative system components, respectively, as functions of displacements $q_{1z} - q_{sz}$ and velocities of motion $\dot{q}_{1z} - \dot{q}_{sz}$ of the system components. In turn, the non-linear characteristics of the controlling forces are functions of input signals u_z as well as deflections and velocities of motion of the system; this was written in the following form, according to the general relation:

$$F_{az} = \sum_{j=1}^{k} F_{azj}\left(u_z, q_{1z} - q_{sz}, \dot{q}_{1z} - \dot{q}_{sz}\right) \tag{7}$$

It should be noted that the biomechanical model of the human body as shown in Fig. 3 does not correspond to the anatomic human body structure; instead, it only describes the basic dynamic properties of the human body. Due to its specific features, the model cannot be used to analyse the vibrations of individual parts of the human body; however, it is suitable for modelling the dynamic action of the human body on the seat. Models of this type are used at the process of designing vibration reduction systems to reveal the resonances occurring in real interactions between the vibration reduction systems and the human body. If this is the case, seat vibrations are measured with the use of a vibration transducer placed at point q_{1z} (Fig. 3).

4. Synthesis of the system to control the seat suspension system

The form of the algorithm to define the controlling force F_{az} that should be introduced into the vibration reduction system for the kinematic excitations q_{sz} to be compensated has been based on a simplified model of the vibro-isolation system (Fig. 4).

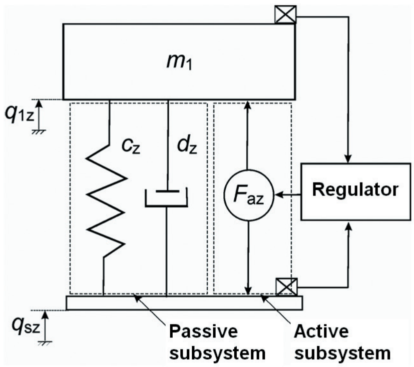


Fig. 4. Simplified model of a hybrid vibration reduction system

A relation that defines the desirable controlling force F_{az} and its first time derivative $\dot{F}_{az}$ has been derived in publication [9]. This was written in the form of the following equations:

$$F_{az} = K_{a1}\left(q_{1z} - q_{sz}\right) + K_{a2}\dot{q}_{1z}\,, \quad \dot{F}_{az} = K_{a1}\left(\dot{q}_{1z} - \dot{q}_{sz}\right) + K_{a2}\ddot{q}_{1z} \tag{8}$$

where: K_{a1} and K_{a2} are regulator settings.

In a semi-active seat suspension system (Fig. 5), the desirable controlling force (relation (8)) should be introduced into the system with the use of a controllable damping element. With this end in view, the conventional hydraulic shock absorber was replaced with a magnetorheological vibration damper. The damper used in the system was filled with a magnetorheological fluid (MR), which was a combination of a carrier fluid with ferromagnetic filings. The piston-in-cylinder movement, induced kinematically, caused the MR fluid to flow through a choke, in result of which a damping force was generated. The choke was surrounded by a solenoid to which electric current with controlled intensity was applied. The essence of the magnetorheological effect lies in changes in fluid viscosity in the working passage of the choke in result of changes in the electromagnetic field. The fluid viscosity changes induce changes in the resistance to the fluid flow through the choke passage, thanks to which the damping force in the system can be controlled [6].

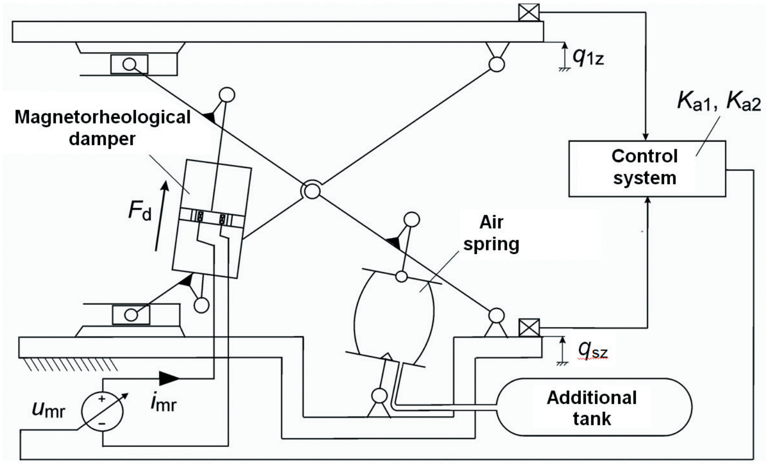


Fig. 5. Model of a semi-active seat suspension system

For a system to be developed that would control the semi-active seat suspension system with a magnetorheological damper, an inverse model of such a control system had to be built. Unambiguity of the model solution was achieved thanks to representing the damper reaction force by a simplified relation in the form as follows [6]:

$$F_d = d_{mr}\dot{l}_d + f_{mr}\,\mathrm{sgn}(\dot{l}_d) \tag{9}$$

where: l_d is the variable length of the damping element. The relations describing the variability of damping coefficient d_{mr} and damper friction force f_{mr} as functions of current intensity i_{mr} were determined within the work reported in publication [9] and they have been expressed in the form of the following equations:

$$d_{mr} = a_1 i_{mr} + a_0 \tag{10}$$

$$f_{mr} = b_2 i_{mr}^2 + b_1 i_{mr} + b_0 \tag{11}$$

where: a_1, a_0 and b_2, b_1, b_0 are parameters of the equations approximating the quantities under consideration.

For the magnetorheological damper described as above, an inverse model of the damper should be determined, i.e. the intensity i_{mr} of the current flowing through the damper, necessary for the desired force F_{az} (equation (8)) to be developed in the semi-active seat suspension system, should be calculated. The calculated current intensity value i_{mr} for the known damper motion velocity $\dot{l}_d$ was described by an equation:

$$i_{mr} = \frac{-b_1\,\mathrm{sgn}(\dot{l}_d) - a_1\dot{l}_d + \mathrm{sgn}(\dot{l}_d)\sqrt{\Delta}}{2b_2\,\mathrm{sgn}(\dot{l}_d)} \tag{12}$$

where:

$$\Delta = \left(b_1 \operatorname{sgn}\left(\dot{i}_d\right) + a_1 \dot{i}_d\right)^2 - 4 b_2 \operatorname{sgn}\left(\dot{i}_d\right)\left(b_0 \operatorname{sgn}\left(\dot{i}_d\right) + a_0 \dot{i}_d - \frac{F_{az}}{\delta_d}\right) \tag{13}$$

and δ_d is the damper leverage.

For the lag of the damping force control system to be reduced, a PD predictor was additionally used in the system. In consequence, the damper control signal was described by the following equation:

$$u_{mr} = \frac{1}{k_{mr}}\left(t_{mr}\dot{i}_{mr} + i_{mr}\right) \tag{14}$$

where: k_{mr} is the static gain of the magnetorheological damper and t_{mr} is the time constant of the damper.

In an active seat suspension system, a possibility of inflating and emptying the deformable chamber of an air spring with the use of proportional flow valves (Fig. 6) was introduced. The air spring was inflated from an external source of compressed air and emptied by being vented directly to the atmosphere. In such a system design, the internal pressure in the air spring could be adjusted relatively quickly, thanks to which the pressure changes could generate an additional force to control the operation of the seat suspension system.

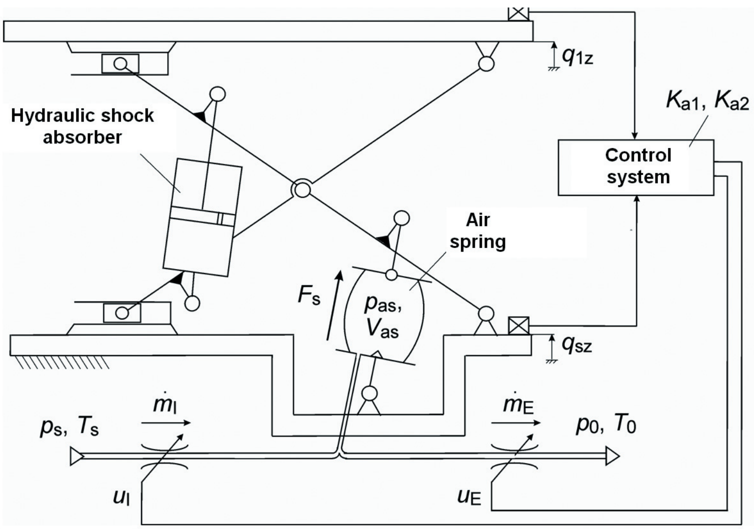


Fig. 6. Model of an active seat suspension system

In this case, the force described by relation (8) should be generated by appropriate changes in the air pressure p_{as} inside the air spring, obtained by means of the valves that control the inflating and emptying of the spring. Therefore, the effective values of the areas of flow through the proportional valves (A_I and A_E) must be calculated for the predefined value of the control force F_{az} and for the current conditions of operation of the active seat suspension system. For this task to be accomplished, an inverse model of the spring element was applied, which was used to calculate the values of the proportional valve control signals (u_I and u_E) that operated the inflating and emptying of the air spring.

To simplify the inverse model being built, an assumption was made that isothermal air transformation takes place inside the spring, which was written as follows, in accordance with relation (7):

$$\dot{p}_{as} = \frac{1}{V_{as}}\left(RT_0\left(\dot{m}_I - \dot{m}_E\right)\right) - p_{as}\dot{V}_{as} \tag{15}$$

where: V_{as} is the variable volume of the air spring, $\dot{m}_I$ and $\dot{m}_E$ are the mass airflow rates of the inflation and emptying of the air spring, R is the individual gas constant, and T_0 is the constant air temperature in the spring, with its value being equal to that of the ambient air temperature. Although the assumption of constant air temperature in equation (15) is a simplification of the model, it considerably facilitates the construction of the inverse model of the air spring fed with the compressed air.

If the relation between the air pressure and force is $p_{as} = F_s / A_{ef} + p_{as0}$ and if the variable volume of the air spring is defined as $V_{as} = A_{ef} l_s$, then the air spring force may be described in a simplified form as follows:

$$\dot{F}_s = \frac{1}{l_s}\left(RT_0\left(\dot{m}_I - \dot{m}_E\right) - \left(F_s + A_{ef} p_{as0}\right)\dot{l}_s\right) \tag{16}$$

where: l_s is the variable length of the spring element, A_{ef} is the effective area of the air spring, and p_{as0} is the air pressure in the state of static equilibrium of the system. The characteristics $\dot{m}_I$ and $\dot{m}_E$, which represent in the model the rates of airflow through the control valves, were modelled with the use of the description formulated by Saint-Venant and Wantzel [7].

The difference between the mass airflow rates $\dot{m}_I - \dot{m}_E$, which should be obtained when inflating or emptying the air spring in order to generate the desired force F_{az} (relation (8)) in the seat suspension system, was described by the following equation:

$$\dot{m}_I - \dot{m}_E = \frac{1}{RT_0}\left(\frac{l_s}{\delta_s}\dot{F}_{az} + \left(\frac{F_{az}}{\delta_s} + A_{ef} p_{as0}\right)\dot{l}_s\right) \tag{17}$$

where: δ_s is the spring force leverage.

For the airflow rates thus defined to be obtained with the use of the proportional valves, the effective flow area A_I through the inlet valve should vary in accordance with the following relations:

- In the case of subcritical flow, when $\dfrac{F_{az}/\delta_s + A_{ef}p_{as0}}{A_{ef}p_s} > \left(\dfrac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}$

$$A_I = \frac{\dot{m}_I}{p_s\sqrt{\dfrac{2\kappa}{RT_0(\kappa-1)}}\sqrt{\left(\dfrac{F_{az}/\delta_s + A_{ef}p_{as0}}{A_{ef}p_s}\right)^{\frac{2}{\kappa}} - \left(\dfrac{F_{az}/\delta_s + A_{ef}p_{as0}}{A_{ef}p_s}\right)^{\frac{\kappa+1}{\kappa}}}} \tag{18}$$

- In the case of critical flow, when $\dfrac{F_{az}/\delta_s + A_{ef}p_{as0}}{A_{ef}p_s} \le \left(\dfrac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}$

$$A_I = \frac{\dot{m}_I}{p_s\sqrt{\dfrac{2\kappa}{RT_0(\kappa-1)}}\sqrt{\left(\dfrac{2}{\kappa+1}\right)^{\frac{2}{\kappa-1}} - \left(\dfrac{2}{\kappa+1}\right)^{\frac{\kappa+1}{\kappa-1}}}} \tag{19}$$

where: κ is the adiabatic curve exponent, p_s is the external air supply pressure.

In turn, the effective flow area A_E through the discharge valve is calculated in accordance with the following relations:

- In the case of subcritical flow, when $\dfrac{A_{ef}p_0}{F_{az}/\delta_s + A_{ef}p_{as0}} > \left(\dfrac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}$

$$A_E = \frac{-\dot{m}_E}{\left(\dfrac{F_{az}}{\delta_s A_{ef}} + p_{as0}\right)\sqrt{\dfrac{2\kappa}{RT_0(\kappa-1)}}\sqrt{\left(\dfrac{A_{ef}p_0}{F_{az}/\delta_s + A_{ef}p_{as0}}\right)^{\frac{2}{\kappa}} - \left(\dfrac{A_{ef}p_0}{F_{az}/\delta_s + A_{ef}p_{as0}}\right)^{\frac{\kappa+1}{\kappa}}}} \tag{20}$$

- In the case of critical flow, when $\dfrac{A_{ef}p_0}{F_{az}/\delta_s + A_{ef}p_{as0}} \le \left(\dfrac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}$

$$A_E = \frac{-\dot{m}_E}{\left(\dfrac{F_{az}}{\delta_s A_{ef}} + p_{as0}\right)\sqrt{\dfrac{2\kappa}{RT_0(\kappa-1)}}\sqrt{\left(\dfrac{2}{\kappa+1}\right)^{\frac{2}{\kappa-1}} - \left(\dfrac{2}{\kappa+1}\right)^{\frac{\kappa+1}{\kappa-1}}}} \tag{21}$$

where: p_0 is the atmospheric pressure.

At an assumption made that the effective cross-section areas A_I and A_E of the orifices that restrict the airflow through the valves change proportionally to the voltages u_I i u_E of the valve control signals and that the lag of the airflow control system has been eliminated by the application of a proportional-derivative (PD) predictor, the valve control signals may be described by the following equations:

$$u_I = \frac{1}{k_I}\left(t_I \dot{A}_I + A_I\right) \tag{22}$$

$$u_E = \frac{1}{k_E}\left(t_E \dot{A}_E + A_E\right) \tag{23}$$

where: k_I and k_E are the static gains of the inlet and discharge valve, respectively, and $t_I$and t_E are the time constants of the corresponding valves.

5. System effectiveness evaluation criteria

The primary criterion used at this work to evaluate the vibro-isolating properties of the seat suspension system was the vibration transfer coefficient TFE_z. It was determined from the root mean square values of vibration acceleration:

$$\mathrm{TFE}_z = \frac{(\ddot{q}_{1z})_{RMS}}{(\ddot{q}_{sz})_{RMS}} \tag{24}$$

where: $(\ddot{q}_{1z})_{RMS}$ is the root mean square value of the acceleration of vibration of the isolated object (seat) in the case of vertical direction of the measured vibration impact and $(\ddot{q}_{sz})_{RMS}$ is the root mean square value of the acceleration of vibration of the motion excitation system (floor in the cabin of the machine under consideration).

The TFE_z coefficient value equal to 1 has the meaning that the vibration discomfort felt by the machine operator is equal to that to be felt as if no system were used to isolate the operator from mechanical vibrations. The value of this coefficient exceeding 1 shows that the vibration discomfort is even increased by the seat suspension system. Only if this coefficient value is below 1, the seat suspension system may be said to produce some desired effect of vibro-isolation of the machine operator from mechanical vibrations.

There is also another criterion, where the evaluation of vibro-isolating properties of a vibration reduction system is based on the maximum relative displacement of the system (s_{tz}), determined in accordance with the following relation [8]:

$$s_{tz} = \max_{t\in[0,t_k]}\left(q_{1z}(t) - q_{sz}(t)\right) - \min_{t\in[0,t_k]}\left(q_{1z}(t) - q_{sz}(t)\right) \tag{25}$$

where: $q_{1z}(t)$ is the displacement of the isolated object along the vertical direction of vibration impact, $q_{sz}(t)$ is the displacement of the motion excitation system, and t_k is the time of duration of the observation period.

For the evaluation criterion thus defined to be determined, the time history of the relative displacement of the vibration reduction system $q_{1z}(t)-q_{sz}(t)$ must be measured, from which the maximum deflection (rebound) of the system would be calculated for a specific input signal that would cause the system to move. It is required for many machines that the values of this criterion should be as low as possible, for adequate operator's contact with machine controls to be secured. Usually, the controls are installed on the unsprung part of the machine; hence, the amplitudes of their movements are in accordance with those of the input movements.

6. Testing of the effectiveness of operation of the seat suspension system with participation of seated humans

To verify the functioning of the engineering solutions of the seat suspension system as described above, experimental tests were carried out with the use of a test stand shown in Fig. 7. The seat suspension system was loaded in succession with seated humans whose body masses were 55 kg and 98 kg.

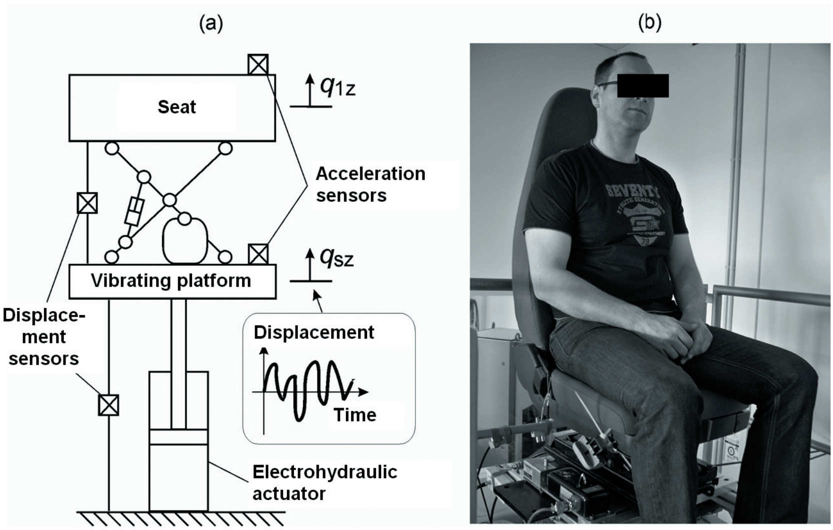


Fig. 7. Facility for experimental testing of a seat suspension system: (a) – schematic diagram; (b) – photograph

A collated summary of the transfer functions of the seat suspension system under examination, determined from the signals of the exciting platform vibration acceleration $\ddot{q}_{sz}$ (system input) and the seat vibration acceleration $\ddot{q}_{1z}$ (system output), has been shown in Fig. 8.

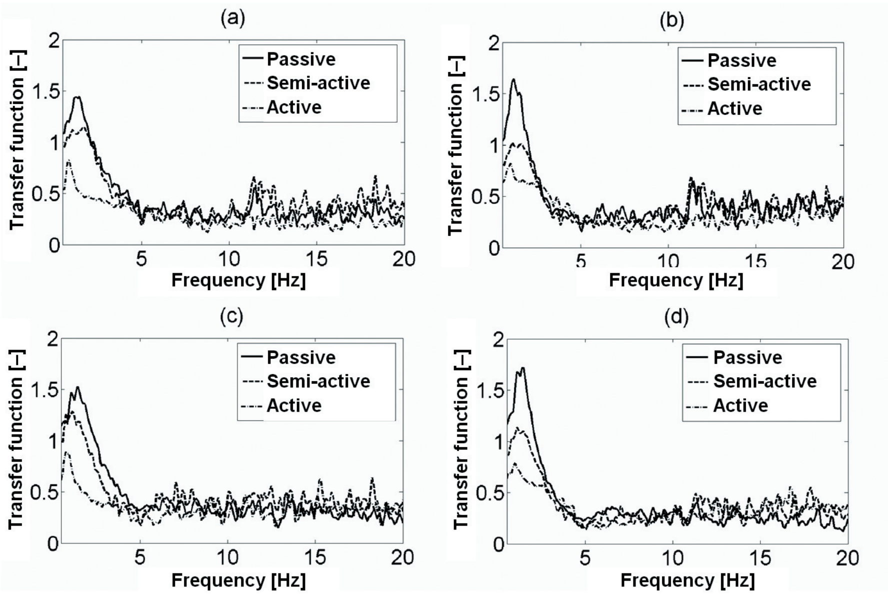


Fig. 8. Transfer functions of the seat suspension system, obtained for the EM5 input signal at different human body masses, i.e. 55 kg (graphs (a) and (c)) and 98 kg (graphs (b) and (d)), determined by computer simulation (graphs (a) and (b)) and experimental measurements (graphs (c) and (d))

Based on a qualitative assessment of the research results obtained, the operation of controllable, i.e. semi-active and active, seat suspension systems was found to be far more effective than that of a passive system (Fig. 8). The conventional passive system increased the vibration amplitudes by about 50% in a frequency range around the natural frequency of the system, although it efficiently protected the man from the harmful impact of vibrations with over-resonant frequencies. The semi-active system effectively lowered the vibration amplitudes at low frequencies of the excitation signals and simultaneously maintained satisfactory vibro-isolating properties at higher vibration frequencies. The active system proved to be the most effective one. It lowered the vibration amplitudes within the full range of excitation frequencies; however, it achieved the highest effectiveness at low vibration frequencies, i.e. at frequencies of up to about 4 Hz.

The simulation and experimental tests have shown that the vibro-isolation system models presented in this paper satisfactorily defined the dynamic properties of a real object, i.e. a real human protected from mechanical vibrations. Thanks to the models correctly formulated, the operation of the analysed vibration reduction systems, whose vibro-isolating properties were determined at the work described here (Table 1), was represented with a high degree of reliability. This is a proof that the biomechanical model adopted at the work was correct because it correctly represented the dynamic characteristics of the human body.

Table 1. Vibration transfer coefficients TFE_z and maximum displacement values s_{tz} for the seat suspension system solutions having been developed, obtained from computer simulations and experimental measurements

Seat suspension system	Input signal	Body mass 55 kg				Body mass 98 kg			
		Simulation		Measurement		Simulation		Measurement	
		TFE_z	s_{tz} [mm]	TFE_z	s_{tz} [mm]	TFE_z	s_{tz} [mm]	TFE_z	s_{tz} [mm]
Conventional passive	EM3	0.891	83	0.953	79	0.783	102	0.852	96
	EM5	0.547	48	0.585	47	0.472	54	0.496	60
	EM6	0.576	8	0.579	9	0.432	10	0.447	11
Semi-active	EM3	0.787	80	0.795	79	0.659	81	0.693	77
	EM5	0.498	46	0.492	49	0.406	44	0.401	48
	EM6	0.505	14	0.555	16	0.399	12	0.406	14
Active	EM3	0.444	77	0.481	81	0.465	80	0.515	85
	EM5	0.332	40	0.329	42	0.381	47	0.349	43
	EM6	0.384	12	0.356	14	0.322	15	0.366	16

7. Recapitulation

The experimental tests carried out with participation of seated humans showed that the vibro-isolation characteristics of the seat suspension system solutions under examination, determined from the simplified system model, were close to the vibro-isolation characteristics of the real system. This proves that the simplifications introduced when modelling the system to reduce mechanical vibrations were correct and that the model adequately represented the physical phenomena that took place in the real system. The differences between the experiment and simulation results obtained may be explained, above all, by diversified individual features of the specific persons that participated in the tests carried out in laboratory conditions.

8. References

[1] *Directive 2002/44/EC of the European Parliament and of the Council on the minimum health and safety requirements regarding the exposure of workers to the risks arising from physical agents (vibration).* Official Journal of the European Communities (2002), pp. 13–18.

[2] ENGEL Z.: *Ochrona środowiska przed drganiami i hałasem* (*Environmental protection from vibration and noise*). Wydawnictwo Naukowe PWN, Warszawa 1993.

[3] ENGEL Z., KOWAL J.: *Sterowanie procesami wibroakustycznymi* (*The control of vibroacoustic processes*). Wydawnictwa AGH, Kraków 1995.

[4] International Organization for Standardization: *Earth-moving machinery – Laboratory evaluation of operator seat vibration*. ISO 7096, Geneva 2000.

[5] International Organization for Standardization: *Mechanical vibration and shock – Range of idealized values to characterize seated-body biodynamic response under vertical vibration*. ISO 5982, Geneva 2001.

[6] KOŁEK K., ROSÓŁ M.: *Zastosowanie tłumika magnetoreologicznego w tłumieniu drgań* (*The application of a magnetorheological damper for vibration damping*). Pomiary Automatyka Robotyka 5/2007, pp. 5–8.

[7] MACIEJEWSKI I., KICZKOWIAK T., KRZYŻYŃSKI T.: *Research and development of seat suspensions for working machines*. Archives of Control Sciences, Vol. 19, No. 4 (2009), pp. 463–478.

[8] MACIEJEWSKI I., MEYER L., KRZYZYNSKI T.: *Modelling and multi-criteria optimisation of passive seat suspension vibro-isolating properties.* Journal of Sound and Vibration 324 (2009), pp. 520–538.

[9] MACIEJEWSKI I.: *Kształtowanie właściwości wibroizolacyjnych układów redukcji drgań stosowanych do ochrony operatorów maszyn roboczych* (*Shaping of vibro-isolating properties of vibration reduction systems used for the protection of heavy machinery operators*). Wydawnictwo Uczelniane Politechniki Koszalińskiej, Koszalin 2012.

[10] PREUMONT A.: *Vibration Control of Active Structures: An Introduction.* Kluwer Academic Publishers, London 2002.