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MODELLING OF THE VIBRATION DAMPING IN AN OPERATOR'S SEAT SYSTEM

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Summary. We present a model of vibrations in a farm tractor operator's seat system with the scope of reducing the vibration energy and of transferring the vibration frequencies into a range which is not harmful for the operator. The results of this analysis constitute a basis for creating a control system for damping seat vibrations in an on-line system using the tractor's on-board computer.

Key words: vibration amplitude, frequency, vibration damping, operator's seat.

INTRODUCTION

The design of an operator's seat should take vibration damping into consideration as much as possible in the process of choosing the materials for the construction of the seat [Majewski 1999, Tytyk 2001, Polish Norms, ISO]. It is possible to install vibration damping devices and pneumatic springs with controlled parameters in the seat mounting mechanism, always taking into account that the amount of damping should be adjusted to the elasticity and load parameters of the entire system [Butenin et al. 1985].

For this model hypothetical tractor operators where divided into three different groups by weight and thus three ranges of seat loads where considered. The calculation of the vibration process dynamics was carried out using "Gnuplot" software. The software uses pre-defined special functions together with implemented spline functions, Bezier curves and a non-linear data modeling process [Hertz 1995]. The modeling of the vibration dynamics for the system under consideration consists in determining the minimum amplitude conditions and thus the extreme value of the function at constant mass (m), elasticity *(k)* and exciting force (Ω) values. The following simplifications were made:

- the impact of frictional forces from the seat mounting mechanism was disregarded,
- the seat was loaded with an operator mass of 70, 80 or 95 kg,
- the unloading of the system caused by the operator's feet resting on, the platform and hands supported on the steering wheel was not taken into account.

FORMULATION OF PROBLEM

The modeling of vibration dynamics will aid in choosing the best construction parameters of a farm tractor seat as far as damping the energy of the vibrations is concerned. An analytical-structural model of vibration damping was created which takes into account the intervals of variability of the process parameters [Cieślikowski 2003]. The research goal is to show that it is possible to decrease the vibration amplitudes of the system as well as to transfer the frequencies into a range which is not harmful for the operator [Burski et al. 2005].. The results of this analysis constitute a basis for creating a control system for damping the seat vibrations in an on-line system using the tractor's on-board computer [Michalski 2000].

MATERIALS AND METHODS

The initial parameters for the calculations were determined from the results of actual measurements on real objects [cf. additional materials]. Fig. 1 presents a diagram of the model imitating the behavior of the real object in actual working conditions [Gryboś 1998].

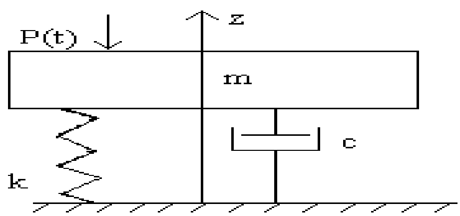

Fig 1. Vibration model of the system being considered

The vibration model consists of a mass *m* placed on the tractor chassis by means of a spring with elasticity *k* and a vibration damper with a damping coefficient *c*. The amplitude of the vibrations is considered only in the direction of the load force along the vertical axis *z* since most seat constructions are not designed to limit horizontal vibrations. The vibrations are incited with a periodically changing force *P = P(t)*.

Assuming that the frequency of the exciting force Ω can take on a wide range of values, we calculated the intervals in which the parameters of the vibration dynamics influence the vibration amplitude [Kucharski 2004]:

- *elasticity impact interval:* for small frequencies $\omega = \Omega \approx 0$, the amplitude changes mainly as a result of a change in the elasticity coefficient,

- *damping influence interval:* as the frequency of the exciting force increases and approaches the normal modes of the system $\omega_w = \frac{k}{m}$ then $\Omega \approx \sqrt{\frac{k}{m}}$, the amplitude of the vibrations at low damping conditions can take on very large values (in the range of resonance),

- *mass influence interval:* for exciting force frequencies $\Omega \rangle \sqrt{\frac{k}{m}}$ for which the effect of the elasticity and damping parameters is not important (supra-resonance interval).

Assuming a harmonic excitation with a frequency Ω and an exciting force amplitude $\overline{P}$ in accordance with the relation $P(t) = \overline{P}\sin\Omega t$ the differential equation for the motions is:

$$m\ddot{z} + c\dot{z} + kz = \overline{P}\sin\Omega\, t. \qquad (1)$$

Using the following substitutions:

$$h = \frac{c}{2m}\,,\ \omega^2 = \frac{k}{m}\,,\ \overline{p} = \frac{\overline{P}}{m}, \qquad (2)$$

and writing the general motion equation as:

$$\ddot{z} + 2h\dot{z} + \omega^2 z = \overline{p}\sin\Omega t. \tag{3}$$

Where: $z = \overline{A}\sin(\Omega t + \varphi),$ (4)
we obtain:

$$-A\Omega^2\sin(\Omega t + \varphi) + 2h\overline{A}\Omega\cos(\Omega t + \varphi) + \overline{A}\omega^2\sin(\Omega t + \varphi) = \overline{p}\sin\Omega t. \tag{5}$$

The solution for this equation is the formula for the vibration amplitudes:

$\overline{A} = \dfrac{\overline{p}}{\sqrt{(\omega^2 - \Omega^2)^2 + (2h\Omega)^2}}$ and the phase angle: $\varphi = -arctg\dfrac{2h\Omega}{\omega^2 - \Omega^2}.$ (6)

In the determined states the following equation for forced harmonic excitation is given using the Euler equations:

$$\overline{P}\sin\Omega t = \frac{\overline{P}e^{j\Omega t} + \overline{P}e^{-j\Omega t}}{2} = \frac{\overline{P}(\cos\Omega t + j\sin\Omega t - \cos\Omega t + j\sin\Omega t)}{2}. \tag{7}$$

After transformation we obtain:

$$(\omega^2 - \Omega^2 + 2h\Omega j)Ae^{j\Omega t} = \overline{p}e^{j\Omega t}. \tag{8}$$

From this equation the complex value of A can be calculated which contains in it both the vibration amplitude and the phase shift angle :

$$A = |A|e^{j\varphi} = \overline{A}e^{j\varphi} = \frac{\overline{p}}{(\omega^2 - \Omega^2 + 2h\Omega j)}. \tag{9}$$

To calculate the extreme value of the function, that is the vibration amplitude value, the elasticity coefficient k was taken as the variable and the other parameters made constant:

$$\overline{A} = \frac{\overline{p}}{\Omega^2\sqrt{(1 - \frac{(\frac{k}{m})^2}{\Omega^2})^2 + \frac{4(\frac{c}{2m})^2}{\Omega^2}}}. \tag{10}$$

The value of Ω was determined from averages and the values of m and $\overline{P}$ were assumed. The damping coefficient has a value of about *15,5 [N s m^{-1}]*. The excitation frequency is *5,72 [s^{-1}]*. was calculated assuming [Wardęga 2006].

These simulations allow us to:

- determine how the vibration amplitude depends on the operator's mass,
- determine how the amplitude changes for each of the various ranges of elasticity,
- examine the relation between the vibration amplitude and the frequency of the exciting force,
- determine the amplitude and the elasticity when damping coefficient, mass and exciting force frequency are given.

An analysis was carried out for the first case to examine the dependence of on the operator's mass and to see how the vibration amplitude changes in response to changes in the elasticity coefficient k. The resulting graphs are shown in Fig. 2, 3 and 4 for the three ranges of seat loads being considered. The horizontal axis corresponds to the parameter *k [kg mm^{-1}]* and the vertical axis to *[m]* (Gnuplot).

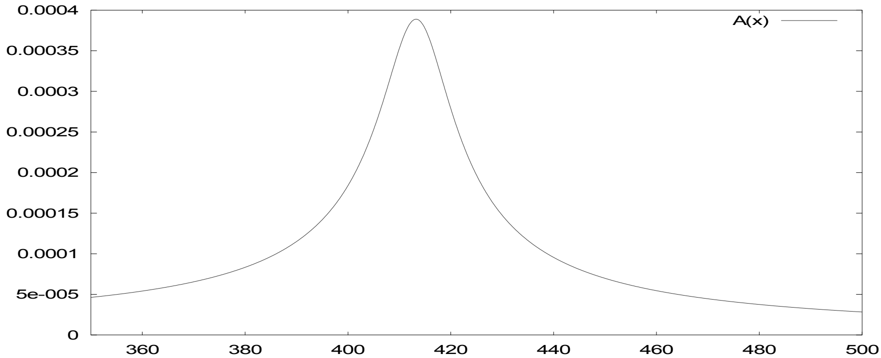

Fig. 2. Vibration amplitude in the vicinity of its maximum value as a function of the elasticity coefficient for a 70 kg operator

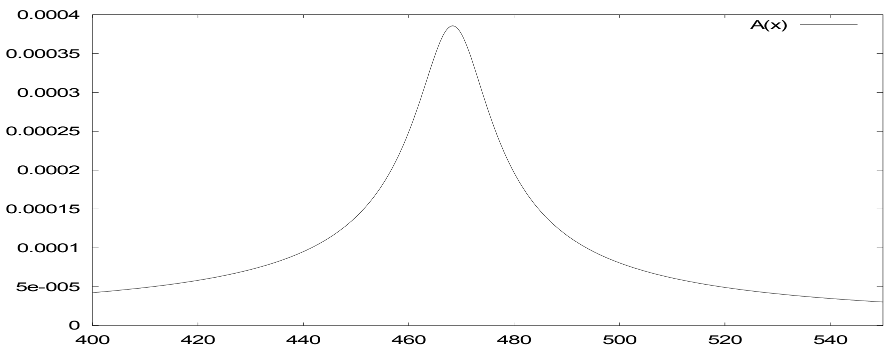

Fig. 3. Vibration amplitude in the vicinity of its maximum value as a function of the elasticity coefficient for an 80 kg operator

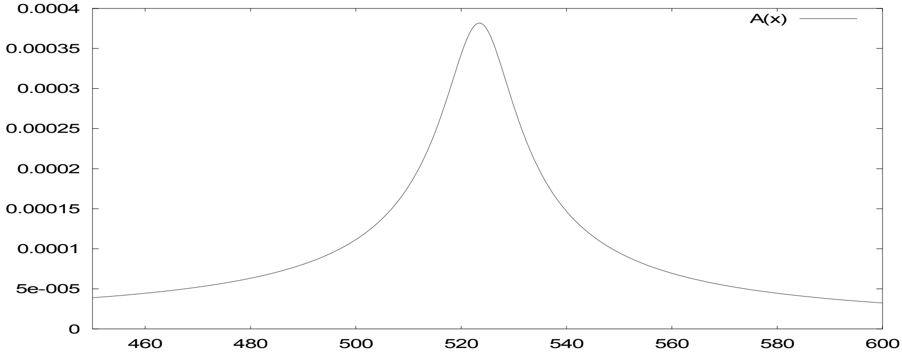

Fig. 4. Vibration amplitude in the vicinity of its maximum value as a function of the elasticity coefficient for a 95 kg operator

The second case deals with the changes in the frequency of the exciting force Ω for varying seat loads. The horizontal axis of the graph represents the value Ω and the vertical axis represents the parameter $\overline{A}$. A calculation of the extreme value of *A(Ω)* indicates that there are three types of *A(Ω)* curves which result from the relation between the parameters *ω* and *h*.

The analysis shows that the value of $\overline{A}$ grows as *h* decreases. For comparison, Figs. 5,6, and 7 show the amplitude for different values of the damping coefficient *h*: 0,1, 0,05, 0,70.

For the case h = 1, ω = 1 the maximum value occurs at the origin. For h = 0.70, ω = 1 there is a flat peak at Ω = 0 since the second derivative of *A* with respect to Ω is zero. The values of the vibration amplitudes for certain exciting frequencies Ω reach a maximum. Harmonic resonance appears which can cause damage to the seat mechanism. The smaller the relative damping coefficient, parameter *h*, the greater the increase of the vibration amplitudes in resonance is. Outside of resonance conditions, an influence of damping on the vibration amplitudes is not very large. For *h > 1* resonance is not observed [Kucharski 2004].

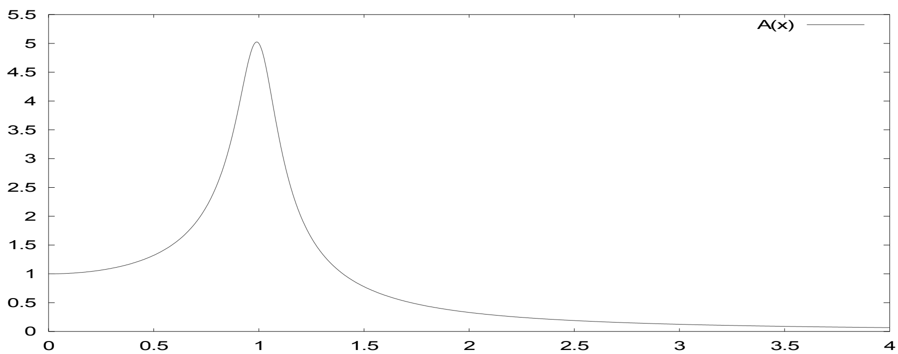

Fig. 5. Vibration amplitude as a function of the excitation frequency for h=0,1

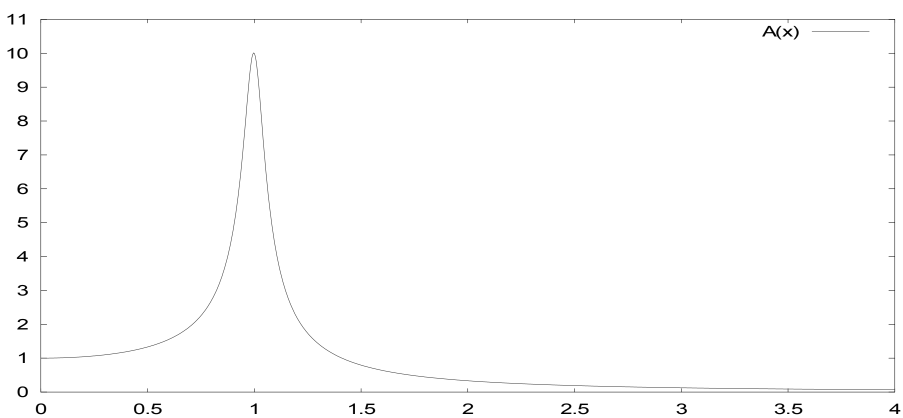

Fig. 6. Vibration amplitude as a function of the excitation frequency for h=0,05

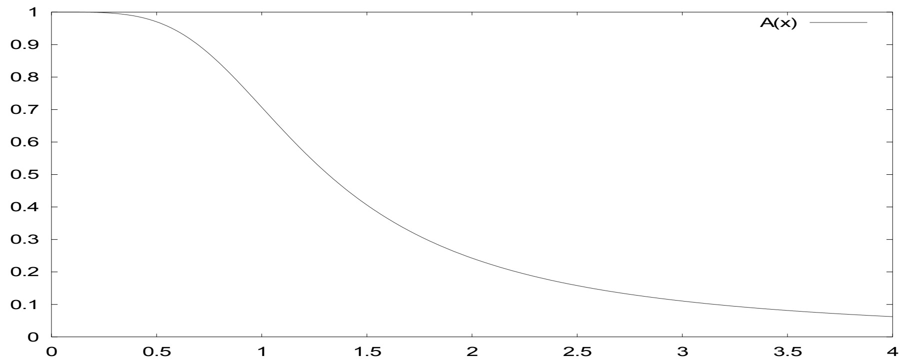

Fig. 7 Vibration amplitude as a function of the excitation frequency for h=0,70

CONCLUSIONS

The model here presented can be used for assuring a reduction of vibrations in a farm tractor operator's work station in such a way that the introduction of the appropriate parameters will lead to a reduction of the seat's vibration amplitudes [Krasowski, Burski 1998]. Both analyzed cases should be taken into account when designing the work station of a farm tractor operator because they have a decisive impact on the dynamics of vibration damping.

It should be emphasized that the choice of damping characteristics and of construction parameters using the assumed system model is just the first stage in the seat design process. The next step should be a simulation to determine whether the vibrations were effectively reduced. Further work should concentrate on creating a prototype which allows for the selection of the vibration dynamics parameters [Jasiński, Szreder 1996]. This relates to the task of perfecting the model and eliminating simplifications.

A numerical method to verify the model confirms that the used structure is correct and gives insight into the correct choice of vibration parameters for the construction of the seat mounting mechanism. If the load on the operator seat is increased, then it is necessary to change the elasticity parameter. This indicates that the seat should be adjusted to the operator weight groups according to the ranges which were proposed for this parameter.

The results of this analysis constitute a basis for creating a control system for damping the seat vibrations in an on-line system using the tractor's on-board computer for the following control stages [Cieślikowski 2005]:

I – recording of the static load on the seat immediately after igniting the tractor's engine. Fitting the seat lever mechanism with a pneumatic shock-absorber will make it possible to register the static load by comparing the difference in pressure before and after loading the seat. At the same time, the computer's memory also records the static sag of the seat mechanism by means of a transducer configured with the mechanism's coupling. We thus obtain the initial conditions of the mechanism as position elements for the calculation of the vibration amplitudes.

II – the correct elasticity of the shock-absorbing system is selected based on the changing vibration amplitudes recorded in the computer's memory and as a result of averaging the vibration parameters in the given time interval.

III - if the vibrations in the given time interval change, a new elasticity value is chosen for the damping system. Because changes in the system's elasticity are introduced by changing the pressure in the pneumatic shock-absorber system, a constant time should be considered for the introduction of rigidity parameters in addition to the time interval of the shock-absorber's pressure regulation.

IV - in each instance the seat vibration amplitude damping process calculates a control vibration frequency which creates a feedback loop in the control system in order to transfer the seat vibration frequencies into a range which is not harmful for the operator.

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CASE III : AXIAL-FLOW 2100,

DEUTZ-FAHR : M. 34.80, M. 35.40, M. 36.40, M. 35.75,

BIZON : REKORD Z058, BS Z110, SAMPO 2020,

STEYER : 8060, 8070, 8080, 8090, 8110, 8130,

JOHN DEERE 2200,

KNECHT : LZO212, LZO260, LZO290,

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MODELOWANIE PROCESU TŁUMIENIA DRGAŃ W UKŁADZIE OPERATOR – SIEDZISKO FOTELA

Streszczenie. W pracy przedstawiono modelowanie procesu drganiowego w układzie operator-siedzisko fotela ciągnika rolniczego w aspekcie obniżenia energii drgań z przeniesieniem częstotliwości do przedziału nieszkodliwego dla operatora. Wyniki analiz stanowią podstawę do opracowania nadążnego układu tłumienia drgań siedziska w systemie on-line z wykorzystaniem komputera pokładowego ciągnika.

Słowa kluczowe: amplituda drgań, częstotliwość, tłumienie drgań, fotel operatora.