LOAD-DEPENDENT CONTROLLER OF THE ACTIVE SEAT SUSPENSION WITH ADAPTIVE MASS RECOGNIZING

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Abstract: The paper deals with the load dependent control system of active seat suspension. This system based on the primary controller which evaluates the desired active force, the reverse model which calculates the input signal of force actuator and the adaptation mechanism which recognizes the actual mass loading. An optimisation procedure additionally presented in this paper allows to find the primary controller settings that minimizes the vibration of cabin's floor transmitted to operator's seat at the pre-defined value of the maximum relative displacement of suspension system.

Key words: Vibration Isolator, Active System, Adaptive Control

1. INTRODUCTION

In the case of typical working machines without flexible wheel suspension, a seat suspension is the only one system, which can protect the operator against vibration. Most often, the passive seat suspension amplifies the vibration amplitudes at resonance frequency. Natural frequency of the conventional seats with an air-spring and hydraulic shock-absorber is between 1-2 Hz (Maciejewski et al., 2009). Low effectiveness of passive seats for a low frequencies and high amplitudes of excitation signal endanger driver's health and finally causes cutting down the working time.

The main aim of the paper is to present an effective way for minimization the vibration risk, acting on the operators of earth-moving machines during their work. In particular the new control algorithm evaluation and the solving multi-criteria optimisation problem are discussed in this paper. The investigation concentrates on the currently manufactured seat, to obtain concurrent improvement at low frequency vibration (active systems) and also at higher frequency vibration (passive systems). The optimal seat should isolate the amplitudes of whole range of excitation frequency and has to show the high system robustness in response to varying mass loading.

2. MODEL OF THE ACTIVE SEAT SUSPENSION

In Fig. 1 a model of the active seat suspension system containing an air-spring and a hydraulic shock-absorber is shown. This system uses the active air-flow to the air-spring (inflating and exhausting) which is regulated by the control system and the proportional directional control valves. The proportional valves have an unlimited number of stable states that are proportional to the analogue input signal obtained from the controller (Beater, 2007). The proportional flow control valves are used in the active seat suspension system and therefore the mass flow rates $\dot{m}_I$ and $\dot{m}_E$ are proportional to the voltage input signal u_I and u_E, respectively. Inflating the air-spring is carried out by an external air compressor and the exhaust of the air-spring is released directly to the atmosphere. The air-flow changes the air pressure inside the air-spring and the variation of pressure creates an active force for the suspension system.

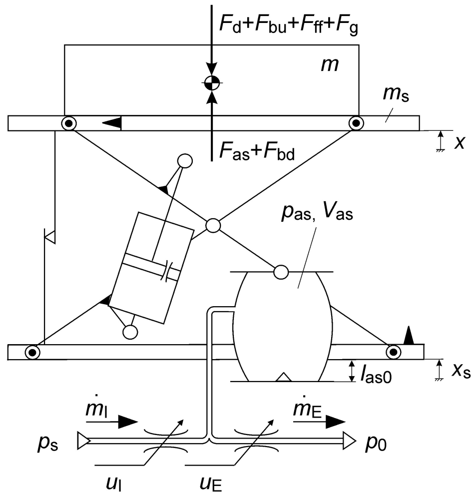

Fig. 1. Model of the active seat suspension

The equation of motion of this seat suspension has been shown in the author's previous papers (Maciejewski et al., 2009, 2010). The mathematical models of reduced forces acting on the suspended mass: the air-spring force F_{as}, the hydraulic shock-absorber force F_d, the forces of end-stop buffers: bottom F_{bd} and top F_{bu}, the overall friction force of suspension system F_{ff} and the gravity force F_g have been presented in those papers as well.

3. CONTROL SYSTEM DESIGN

Conventional seat suspension systems are sensitive to changes in their working conditions. The static and dynamic behaviour of the system depends on the mass m loading the suspension, because the suspension system spring force F_{as} is a function of the pressure inside the air-spring (Maciejewski et al., 2010). This results in differing vibro-isolating properties of the conventional seat suspension system for different weights of machine operator. The active seat should ensure the same vibration isolating properties of the suspension system for each operator. This effect can be obtained using a load-dependent controller for the active seat suspension system that is shown in Fig. 2.

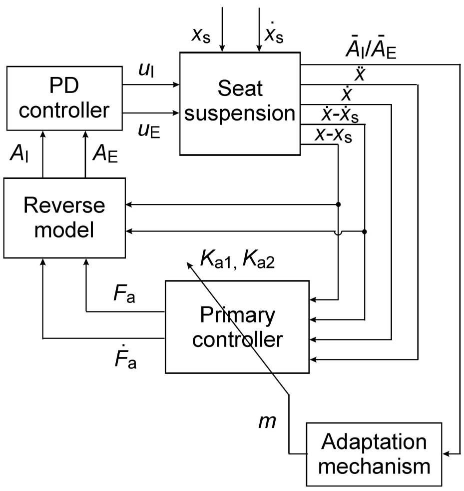

Fig. 2. Control system of the active seat suspension with adaptive mass recognizing

In this control system (Fig. 2), the desired active force ($F_a, \dot{F}_a$) is calculated using the primary controller. If the desired force is evaluated, then such force should be approximately reproduced by the active element applied in seat suspension. Therefore the actual effective areas (A_I, A_E) of proportional control valves are calculated using reverse model of the air spring. The actuating time of the force actuator is eliminated because the PD controller speed up the control signals (u_I, u_E). In addition, an adaptation mechanism evaluates the controller settings (K_{a1}, K_{a2}) as a function of the suspended mass (m). The suspended mass is recognized based on the mean values over time of the effective cross section areas of inlet and outlet valves ($\bar{A}_I, \bar{A}_E$). A detailed description of the each control system components are presented in the next subsections.

3.1. Simplified model of the air-spring

A simplified model of the air-spring relies on isothermal behaviour of air. According to this model, the actual value of air pressure p_{as} inside the chamber is defined as [3]:

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

where: V_{as} is the variable volume of the air-spring, R is the gas constant, T is the air temperature. Defining pressure-force relation of the air-spring $p_{as} = \frac{\delta_{as}}{A_{ef}} F_{as} + p_{as0}$ and its variable volume $V_{as} = A_{ef}\left(\frac{x - x_s}{\delta_{as}} + l_{as0}\right)$ the force of the air-spring can be calculated as follows:

$$\dot{F}_{as} = \frac{1}{x - x_s + \delta_{as} l_{as0}} \cdot \left(RT(\dot{m}_I - \dot{m}_E) - \left(F_{as} + \frac{A_{ef}}{\delta_{as}} p_{as0}\right)(\dot{x} - \dot{x}_s)\right) \quad (2)$$

where: δ_{as} is the reduction ratio of the air-spring, l_{as0} is the initial length of the air-spring, A_{ef} is the effective area of air-spring, p_{as0} is the initial pressure of the air-spring. The reduction ratio δ_{as} is defined as a proportion of the motion of the suspension system relative to the air-spring.

The mass flow rates $\dot{m}_I$ and $\dot{m}_E$ are calculated using the Mietluk-Awtuszko function (Kiczkowiak, 2005), that assumes the adiabatic air-flow, as follows:

– for inflating of the air-spring:

$$\dot{m}_I = 0.5787 A_I p_s \sqrt{\kappa}\sqrt{\frac{1}{RT}}\,\alpha\,\frac{1-\left(\frac{\delta_{as}}{A_{ef}}F_{as}+p_{as0}\right)/p_s}{\alpha-\left(\frac{\delta_{as}}{A_{ef}}F_{as}+p_{as0}\right)/p_s} \quad (3)$$

– for exhausting of the air-spring:

$$\dot{m}_E = 0.5787 A_E \left(\frac{\delta_{as}}{A_{ef}}F_{as} + p_{as0}\right)\sqrt{\kappa}\sqrt{\frac{1}{RT}}\,\alpha\,\frac{1-p_0/\left(\frac{\delta_{as}}{A_{ef}}F_{as}+p_{as0}\right)}{\alpha-p_0/\left(\frac{\delta_{as}}{A_{ef}}F_{as}+p_{as0}\right)} \quad (4)$$

where: A_I is the effective cross section area of inlet valve, A_E is the effective cross section area of outlet valve, p_s is the air pressure of power supply, p_0 is the atmospheric pressure, κ is the adiabatic coefficient, α is the parameter of the Mietluk-Awtuszko function.

3.2. Reverse model of the air-spring

The description of the air-spring force relating to the mass flow rate (Eq. (2)) can be rearranged in the following form:

$$\dot{m}_I - \dot{m}_E = \frac{1}{RT}\left(\begin{array}{c}\dot{F}_{as}(x - x_s + \delta_{as} l_{as0}) + \\ \left(F_{as} + \frac{A_{ef}}{\delta_{as}} p_{as0}\right)(\dot{x} - \dot{x}_s)\end{array}\right) \quad (5)$$

where: $\dot{m}_I - \dot{m}_E$ is the mass flow rate to achieve the desired air-spring force F_{as} at variable deflection $x - x_s$ and velocity $\dot{x} - \dot{x}_s$ of the air-spring. In order to obtain such mass flow rate, the effective cross section area of proportional valve should be calculated as follows:

– for inlet valve:

$$A_I = \frac{\dot{m}_I}{0.5787 A_I p_s \sqrt{\kappa}\sqrt{\frac{1}{RT}}\,\alpha\,\frac{1-\left(\frac{\delta_{as}}{A_{ef}}F_{as}+p_{as0}\right)/p_s}{\alpha-\left(\frac{\delta_{as}}{A_{ef}}F_{as}+p_{as0}\right)/p_s}} \quad (6)$$

– for outlet valve:

$$A_E = \frac{\dot{m}_E}{0.5787 A_E\left(\frac{\delta_{as}}{A_{ef}}F_{as}+p_{as0}\right)\sqrt{\kappa}\sqrt{\frac{1}{RT}}\,\alpha\,\frac{1-p_0/\left(\frac{\delta_{as}}{A_{ef}}F_{as}+p_{as0}\right)}{\alpha-p_0/\left(\frac{\delta_{as}}{A_{ef}}F_{as}+p_{as0}\right)}} \quad (7)$$

Assuming that the proportional directional control valves are utilized then the effective cross section areas: A_I and A_E are proportional to the electric input signals: u_{Iv} and u_{Ev}. In order to take into account the actuating times: t_I and t_E of the proportional control valves (for inlet and outlet valves, respectively), the electric input signals are subsequently formed by proportional-derivative (PD) controller as follows:

$$u_{Iv} = \frac{1}{k_I}\left(t_I\dot{A}_I + A_I\right), \qquad u_{Ev} = \frac{1}{k_E}\left(t_E\dot{A}_E + A_E\right) \tag{8}$$

where: k_I is the static gain of inlet valve and k_E is the static gain of inlet valve. The electric input signals for inlet and outlet control valves are finally limited in the operating ranges as:

$$u_I = \begin{cases} 0 & for \quad u_{Iv} < 0 \\ u_{Iv} & for \quad 0 < u_{Iv} \le u_{max} \\ u_{max} & for \quad u_{Iv} > u_{max} \end{cases} \tag{9}$$

$$u_E = \begin{cases} 0 & for \quad u_{Ev} < 0 \\ u_{Ev} & for \quad 0 < u_{Ev} \le u_{max} \\ u_{max} & for \quad u_{Ev} > u_{max} \end{cases} \tag{10}$$

where: u_{max} is the maximum input voltage that is responsible for the complete opening of proportional control valves.

3.3. Formulation of the primary controller

The full active system is used to determine desired force F_a that should be introduced into the suspension system actively (Fig. 3a). Subsequently, this desired force should be reproduced by the air-spring force F_{as} in hybrid suspension system (Fig. 3b).

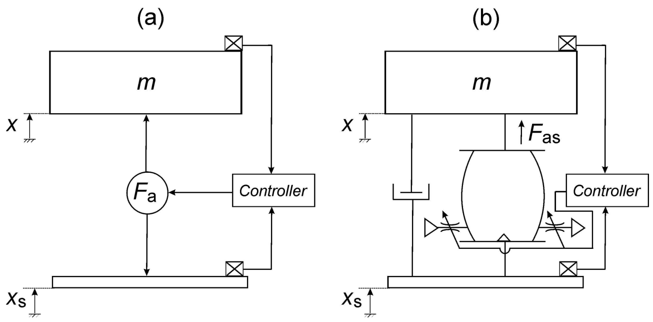

Fig. 3. Simplified seat suspension models: active (a), hybrid (b)

The relationships for the desired active force and its time derivative can be formulated in the following form (Maciejewski, 2012):

$$F_a = K_{a1}(x - x_s) + K_{a2}\dot{x} \tag{11}$$

$$\dot{F}_a = K_{a1}(\dot{x} - \dot{x}_s) + K_{a2}\ddot{x} \tag{12}$$

where: $x - x_s$ is the relative displacement of suspension system, $\dot{x}$ is the suspended mass velocity, $\dot{x} - \dot{x}_s$ is the relative velocity of suspension system, $\ddot{x}$ is the suspended mass acceleration, K_{a1} and K_{a2} are the primary controller settings to be designed.

3.4. Optimisation of the primary controller settings

In order to improve operator comfort the effective suspended mass acceleration $\ddot{x}_{RMS}$ should be small. At the same time it is necessary to ensure that the maximum relative displacement of suspension system $(x - x_s)_{max}$ is small enough to ensure that even very rough road profiles do not cause the deflection limits to be reached [2]. The trade-off between operator comfort and suspension deflection can be selected by the primary controller settings: K_{a1} and K_{a2} that are optimised for different requirements defined by the machine operators.

In this case the minimization of the effective suspended mass acceleration is proposed as:

$$\min_{K_{a1},K_{a2}} \ddot{x}_{RMS}(K_{a1}, K_{a2}) \tag{13}$$

using a constraint imposed on the maximum relative displacement of suspension system in the following form:

$$(x - x_s)_{max}(K_{a1}, K_{a2}) \le (x - x_s)_c \tag{14}$$

where: $(x - x_s)_c$ defines the constraint value. An appropriate selection of such value allows to choose the vibro-isolation properties of the suspension system.

The set of Pareto-optimal solutions, that are obtained for different mass loading: $50\,kg$, $75\,kg$, $100\,kg$, $125\,kg$, $150\,kg$, are presented in Fig. 4.

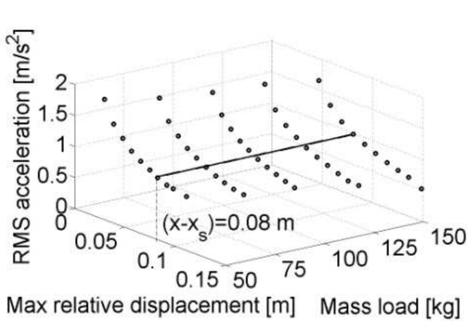

Fig. 4. Pareto-optimal point distribution for the active seat suspension obtaibed for different mass loading

The optimisation results (Fig. 4) show, that is possible to achieve nearly the same dynamic behaviour of active seat suspension for the same value $(x - x_s)_c = 0.08\,m$ of the maximum relative displacement of suspension system. In order to achieve such a system performance, the controller settings K_{a1}, K_{a2} have to be described as a function of suspended mass m. They are approximated using the exponential functions as follows:

$$K_{a1} = a_{a1}\exp(b_{a1}m) \tag{15}$$

$$K_{a2} = a_{a2}\exp(b_{a2}m) \tag{16}$$

where: a_{a1}, b_{a1} and a_{a2}, b_{a2} are the coefficients of the exponential functions to be evaluated by the least square approximation. The approximation results, that show the dependency of the controller settings: K_{a1} and K_{a2} for different m, are shown in Fig. 5.

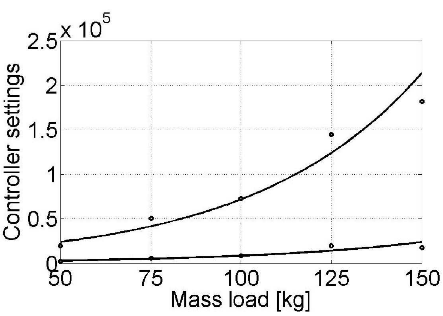

Fig. 5. Optimal controller settings

3.5. Adaptation mechanism

The load-dependent controller is proposed to control the active suspension system more efficient. The controller settings: K_{a1} and K_{a2} depend on the information of the suspended body mass and such value has to be known. This causes that an additional sensor could be employed in order to recognize the suspended mass m. However, the online available information of the air-spring inflation/deflation are helpful to recognize the mass load of the active pneumatic seat suspension. Using a principle that the mean values of inflated and exhausted air are equal ($\bar{\dot{m}}_I = \bar{\dot{m}}_E$), the corresponding Mietluk-Awtuszko functions (Eqs. (3) and (4)) can be compared in the following form:

$$\bar{A}_I p_s \frac{1-\bar{p}_{as}/p_s}{\alpha-\bar{p}_{as}/p_s} = \bar{A}_E \bar{p}_{as} \frac{1-p_0/\bar{p}_{as}}{\alpha-p_0/\bar{p}_{as}} \tag{17}$$

where: $\bar{p}_{as}$ are the mean value over time of the air-spring pressure. Determining the quotient $\bar{A}_I/\bar{A}_E$ of the effective cross section areas of inlet and outlet valves, the Eq. (17) can be rearranged as follows:

$$\frac{\bar{A}_I}{\bar{A}_E} = \frac{((\bar{p}_{as})^2-p_0\bar{p}_{as})(\alpha p_s-\bar{p}_{as})}{(\alpha\bar{p}_{as}-p_0)((p_s)^2-p_s\bar{p}_{as})} \tag{18}$$

Setting $f(\bar{p}_{as}) = 0$ produces the cubic equation in the following form:

$$a(\bar{p}_{as})^3 + b(\bar{p}_{as})^2 + c\bar{p}_{as} + d = 0 \tag{19}$$

with the coefficients: a, b, c, d of the cubic equation described as follows:

$$a = 1 \tag{20}$$

$$b = -\frac{\bar{A}_I}{\bar{A}_E}\alpha p_s - \alpha p_s - p_0 \tag{21}$$

$$c = -\frac{\bar{A}_I}{\bar{A}_E}\alpha (p_s)^2 + \frac{\bar{A}_I}{\bar{A}_E} p_s p_0 + \alpha p_s p_0 \tag{22}$$

$$d = -\frac{\bar{A}_I}{\bar{A}_E}(p_s)^2 p_0 \tag{23}$$

Solving the cubic equation (Eq. (19)) amounts to finding the roots of a cubic function. Defining a discriminant of the cubic equation as follows:

$$\Delta = \left(\frac{p}{3}\right)^3 + \left(\frac{q}{2}\right)^2 \tag{24}$$

where:

$$p = \frac{c}{a} - \frac{b^2}{3a^2}, \qquad q = \frac{2b^3}{27a^3} + \frac{d}{a} - \frac{bc}{3a^2} \tag{25}$$

Then for $\Delta < 0$ is possible to find each real roots of the cubic function (Eq. (19)) in the following form:

$$(\bar{p}_{as})_k = 2\sqrt{\frac{-p}{2}}\cos\left(\frac{\varphi+2(k-1)\pi}{3}\right) - \frac{b}{3a} \tag{26}$$

where:

$$\varphi = \arccos\left(\frac{-\frac{q}{2}}{\sqrt{\frac{-p^3}{27}}}\right) \tag{27}$$

and $k = 1, \ldots, 3$ is the number of root found. The body mass should be calculated for given root number of the cubic equation (Eq. 19)) as follows:

$$m = \frac{(\bar{p}_{as}-p_0)A_{ef}}{\delta_{as} g} \tag{28}$$

Using Eqs. (24)-(28) online calculation of the body mass based on the effective cross section areas of inlet and outlet valves: $\bar{A}_I$ and $\bar{A}_E$ is enabled. However, information of the actual cross section areas of inlet and outlet valves is not easily accessible. Therefore an adaptive load recognizing of the active suspension system is performed based on the control signals of inlet and outlet valves: $\bar{u}_I$ and $\bar{u}_E$. In this paper, a quotient of the effective cross section areas of inlet and outlet valves $\bar{A}_I/\bar{A}_E$ is assumed for the high performance pneumatic valves to be approximately equal to a quotient of their control signals multiplied by the corresponding static gains $k_I\bar{u}_I/k_E\bar{u}_E$ (i.e. $\bar{A}_I/\bar{A}_E \approx k_I\bar{u}_I/k_E\bar{u}_E$). The simulation results of recognized suspended mass as a function of the actual suspended mass, that are obtained for the root number $k = 3$ and for the simulation time of $300\ s$, are presented in Fig. 6a. Corresponding relative errors of the recognized mass are shown in Fig. 6b.

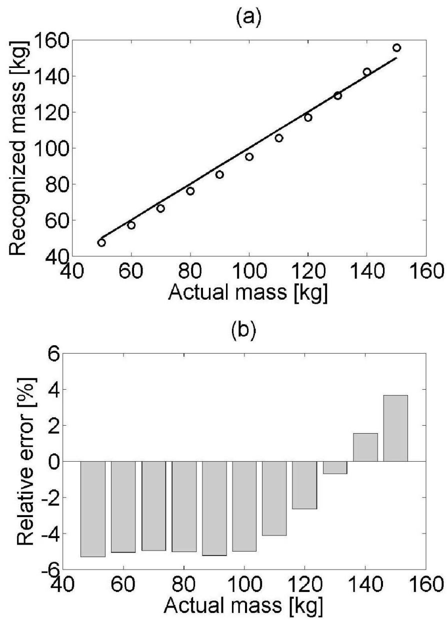

Fig. 6. Recognized suspended mass in function of the actual suspended mass (circles) and its linear dependency (solid line) (a), corresponding relative error of the recognized mass (b)

4. SIMULATION RESULTS

The transmissibility curves of the simulated conventional passive suspension system for variations of the mass loading $\pm 50\ \%$ are presented in Fig. 7a. As shown in this figure, the change of mass influences the dynamic behaviour of passive seat suspension. The suspension system loaded with a high mass ($150\ kg$) causes amplification of vibration at the resonance frequency, but for higher frequencies the best vibro-isolating properties are obtained. The system loaded with a low mass ($50\ kg$) has much lower amplification of vibration at resonance, but its vibration isolating properties at higher frequency range are poor. The dynamic behaviour of the active seat suspension (Fig. 7b) loaded by a high mass ($150\ kg$) and low mass ($50\ kg$) are much closer to each other than the corresponding behaviour of the passive seat suspension. The main improvement of the system robustness is observed in the frequency range 0.5-4

Hz. In that frequency range the air-flow effectively controls the air-spring force.

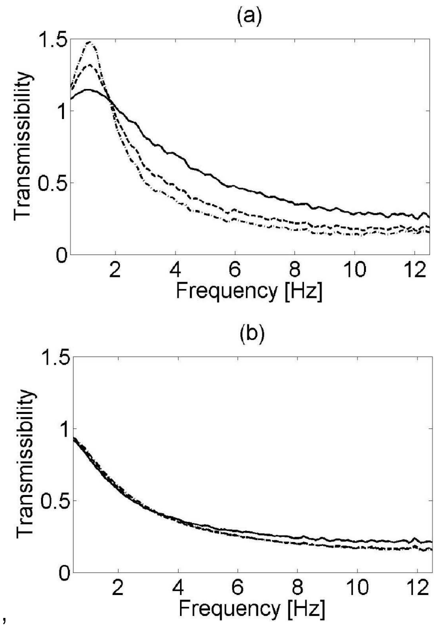

Fig. 7. Transmissibility curves of the passive (a) and active (b) seat suspension obtained for the suspended masses: 50 kg (solid line), 100kg (dashed line), 150 kg (dashed-dotted line)

5. CONCLUSIONS

The air-spring control system applied to the active seat suspension significantly improves the performance of machine operators' seat. The calculated reverse model of air-spring together with designed primary controller efficiently controls the active seat suspension dynamic behaviour. Moreover, the elaborated controller with adaptive mass recognizing ensures the desired system robustness to varying mass loading. In order to achieve the desired vibro-isolation properties of active seat suspension, an appropriate selection of the constraint value imposed on the maximum relative displacement of suspension system is required. Based on the chosen value, the controller settings are evaluated and their values define the vibration damping effectiveness of active seat.

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