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Active control of a seat suspension with the system adaptation to varying load mass

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ABSTRACT

This paper deals with the control system design of active seat suspension. The proposed control system structure is built basing on several various controllers. The primary controller is used to evaluate the actual value of the desired active force that should be generated in the suspension system. The secondary controller is employed to calculate the instantaneous value of a signal which controls the active element by means of its reverse model. The adaptation mechanism recognises the actual suspended mass in order to increase the effectiveness of vibration isolation. Additionally, the system robustness for different load masses is investigated using computer simulation and experimental research.

1. Introduction

The amplification of low frequency vibrations occurring in passive seats can be overcome with an active suspension mechanism. The mechanisms of this type employ a source of power to move the seat relative to its base. As the vehicle oscillates, the seat can be held relatively stationary in space under the control of an servomechanism. The studies of this problem have been of academic and industrial interest for a long time. At first, the passive suspensions have been investigated [1,2] and afterwards the semi-active [3] and active [4–6] systems have been developed. The conclusion drawn from these articles is that the semi-active and active suspensions enhance comfort considerably when compared to the passive systems. The concept of the semi-active and active suspensions can be successfully applied in road-vehicle suspensions, cabin suspensions in trucks or seat suspensions [7,8]. The main idea of these system is to use suspension control techniques for reducing the vibrations.

According to [9] the semi-active suspension system can be effectively applied to the passenger vehicle with improved both ride comfort and steering stability. Authors tested a car installed with four electro-rheological dampers which allowed to evaluate performance characteristics of the semi-active suspension system associated with skyhook controller. A three-dimensional vehicle model and the general seat suspension model was analysed using optimal dampers in the case of passive, semi-active and active vehicle suspensions [10]. Authors analysed the effects of vibrations and seat positions on comfort and road holding capabilities of 3-D road vehicles as observed in the variation of different parameters such as suspension coefficients, road disturbances and the seat position. By developing an objective function they generated an optimisation algorithm allowing calculation of the optimal suspension parameters. The optimal active suspension of the seat also has been obtained analytically for a human body represented alternatively by apparent mass weighted by standard frequency domain curves depicting discomfort levels [11].

The feedback control is intended to improve the dynamics of suspension systems, especially for a specific working conditions. The control element in semi-active and active vibration isolation systems is adjusted by means of the controller which uses information concerning the system states. However, there are some principal factors that influence the problem of vibration isolation of a whole body system. One of these problems is the selection of the appropriate model parameter like the damping and the stiffness and the problem of H-inf control for the active seat suspension systems via dynamic output feedback control [12]. In the paper [13] static-output-feedback controller design problem is investigated. The two-stage method is developed to determine the static-output-feedback gain matrix for the structural system even if actuator faults occur.

There are some works presented in the existing literature wherein active suspension systems are considered in relation to varying mass loading. A load-dependent controller design approach to solve the problem of the multi-objective control for active suspension systems is presented in the paper [14]. The paper

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Nomenclature

A_I	effective cross section area of the inlet valve, m^2
A_E	effective cross section area of the outlet valve, m^2
A_{ef}	effective area of the pneumatic spring, m^2
$a_{a1}, a_{a2}, b_{a1}, b_{a2}$	coefficients of logarithmic functions
a_s, b_s, c_s, d_s	coefficients of a cubic equation
F_a	desired active force, N
F_s	force of the pneumatic spring, N
SEAT	Seat Effective Amplitude Transmissibility factor
g	gravity constant, m/s^2
K_{a1}	proportional gain of the relative displacement feedback loop, V/m
K_{a2}	proportional gain of the velocity feedback loop, Vs/m
k_I	static gain of the inlet valve, m^2/V
k_E	static gain of the outlet valve, m^2/V
l_s	variable length the pneumatic spring, m
m	mass of the isolated body, kg
$\dot{m}_I$	mass flow rate for inflating of the pneumatic spring, kg/s
$\dot{m}_E$	mass flow rate for exhausting of the pneumatic spring, kg/s
p_a	atmospheric pressure, Pa
p_s	air pressure inside the pneumatic spring, Pa
p_{s0}	initial pressure of the pneumatic spring, Pa
p_z	air pressure of the power supply, Pa
R	gas constant, J/(kg K)
T	air temperature, K
t	computation time instant, s
t_I	actuating time of the inlet valve, s
t_E	actuating time of the outlet valve, s
u_I	control voltage of the inlet valve, V
u_E	control voltage of the outlet valve, V
u_{max}	maximum value of the control voltage, V
V_s	volume of the pneumatic spring, m^3
x	displacement of the isolated body, m
x_s	displacement of the excitation, m
$(x-x_s)_c$	constraint value imposed on the suspension travel, m
$(x-x_s)_{max}$	suspension travel, m
$(\ddot{x}_w)_{RMS}$	frequency weighted root mean square value of the simulated input acceleration, m/s^2
$(\ddot{x}_{sw})_{RMS}$	frequency weighted root mean square value of the measured seat acceleration, m/s^2
α	parameter of the Mietluk–Awtuszko function
δ_s	reduction ratio of active force acting in the vertical direction on the isolated body
κ	adiabatic coefficient

[15] shows an adaptive sliding controller for a non-autonomous suspension system with time-varying loadings. However, the usefulness and the advantages of the proposed controller design methodology are demonstrated via numerical simulations only. The objective of this paper is to suggest a new control strategy in order to increase vibro-isolating properties of the active seat suspension in response to varying load mass. In the paper [16], the first simulation results have been discussed, while the presented adaptive control is investigated experimentally using a laboratory method for measuring and evaluating the effectiveness of the seat suspension in reducing the vertical whole-body vibration.

The contribution of the paper consists in a better vibration control of seat suspension and a higher system robustness to the operator's mass variations. Using a unique vibration control system, that structure and individual components are presented in this paper, it is possible to achieve the desired system properties in view of the different requirements for the modern seat suspension systems.

2. Simulated input vibration to evaluate the SEAT factor and suspension travel

The International Standard [17] specifies the input vibrations for the purpose of determining the Seat Effective Amplitude Transmissibility (SEAT) factor [18]. The particular input vibration is used to evaluate the seat performance for the different types of earth-moving machinery. Each class (EM1–EM9) defines a group of machine having similar vibration characteristics. In this paper, only selected input vibrations are used to evaluate the seat suspension dynamic behaviour:

- EM3 – an excitation signal representative of wheel loaders.
- EM5 – an excitation signal representative of wheel dozers.
- EM6 – an excitation signal representative of crawler loaders and crawler dozers.

The power spectral densities of mentioned excitation signals are presented in Fig. 1. As shown in this figure, the magnitude of the generated input vibrations has to be within the tolerance of the target power spectral density function, that is defined in the paper [17].

The SEAT factor provides the first numerical assessment of the seat isolation efficiency and its value is calculated as follows [18]:

$$\text{SEAT} = \frac{(\ddot{x}_w)_{RMS}}{(\ddot{x}_{sw})_{RMS}} \tag{1}$$

where $\ddot{x}_{sw}$ is the frequency weighted root mean square value of the simulated input acceleration, $\ddot{x}_w$ is the frequency weighted root mean square value of the measured seat acceleration. The frequency weighting of acceleration signal shall be done in accordance with [19].

The SEAT factor value of 1 means that seating on the floor plate in the working machine cabin would produce the same vibration discomfort. If the SEAT factor value is greater than 1, the vibration discomfort is increased by the seat. If the SEAT factor value is less than 1, the useful vibro-isolation is provided by the seat.

The second numerical assessment of the seat dynamic performance is the suspension travel $(x-x_s)_{max}$. In this paper, the suspension travel is defined by the maximum relative displacement of suspension system and its value is calculated as follows [20]:

$$(x-x_s)_{max} = \max_t(x-x_s) - \min_t(x-x_s) \tag{2}$$

where x is the displacement of the seat and x_s is the displacement of the input vibration, t is the computation time instant.

There is no comfort criteria standards for the selection of the trade-off between the SEAT factor and the suspension travel $(x-x_s)_{max}$. On the one hand the SEAT factor should be minimised, but this is only the first objective of a system evaluation. The second objective is that the suspension travel has to be small in order to ensure, that even very rough road profiles do not cause the deflection limits to be reached [21]. An improvement in one objective requires a degradation of another, therefore these objectives conflict. Consequently, the design of seat suspension systems can be treated as an optimisation problem.