LABORATORY TESTS OF ACTIVE SUSPENSION SYSTEM

Jarosław Konieczny

AGH University of Science and Technology, Department of Process Control
Mickiewicz Av. 30 B-2 p.110/7, Krakow 30-059, Poland
phone: +48 12 6173032, fax: +48 12 4233076
e-mail: jaroslaw.konieczny@agh.edu.pl

Abstract

The main purpose of this study is to design an active vehicle suspension ensuring high vibration isolation efficiency and stability though the limited consumption of external energy. Active systems used in vehicle suspensions widely employ parallel (full active) structures. It shall be analysed and tested experimentally. The major part of the study summarises the laboratory experiments carried on a designed active vehicle suspension in the form of a quarter car model. Apart from the vibration isolation efficiency offered by the designed and engineered suspension, other considered performance indicators include the system's stability and external energy demand. A laboratory model of the suspension is described and laboratory tests are outlined that were performed to check the adequacy of various control algorithms. An attempt is made to evaluate the energy demand for the given structure and to determine the power ratings of the source supplying the active vibration isolation system. Inter alia quarter-vehicle full active suspension, vibration displacement transmissibility in the function of frequency, transmissibility function from the input to the output for various types of actuator's controllers, comparison of instantaneous power absorbed from the power-supplying unit for various types of actuator's controllers, comparison of instantaneous power intake from the supplying unit for various types of controllers are presented in the paper.

Keywords: *active vehicle suspension, control system , energy consumption, laboratory research*

1. Introduction

Vibration isolation efficiency becomes a key quality indicator of active controlled vehicle suspensions. However, high energy demand of these solutions precludes their wider use. The actual consumption of external energy by active suspension systems is chiefly determined by the parameter is associated with the roughness of the terrain, or, in other words, with the type of acting excitations. Suspension designers cannot influence the type of terrain, however the type of the applied controller for active vibration isolation systems determines the amount and the manner in which energy is transmitted to the system.

The experimental program outlined in this study uses an electro-hydraulic actuator, which is a very popular solution, yet satisfying the specified requirements in terms of generated force and displacement.

An extensive study of the literature on the subject reveals that kinematic structures widely applied in active suspension systems include parallel and series systems or full and slow active systems. The parallel structure, also referred to as broadband, demands the active system's operation in a wide frequency range (from 0 to 10-15 Hz). In the broadband "mode of operation' the energy demand of the active unit is considerable. Its main advantage, however, is that it does not require any extensions of the suspension strut. The difference between this solution and conventional passive systems is that the passive damper is replaced by a controlled actuator.

2. Active vehicle suspension model

The kinematic diagram of the analysed suspension is shown in Fig. 1. A quarter vehicle model of a suspension is adopted, with lumped parameters. This is a 2 DOF model, where the first DOF, associated with the mass m_1, represents the wheel with a tire. The wheel mass is unsprung. The second DOF, associated with the mass m_2, represents the car body with passengers. This mass is referred to as sprung mass. The main task is to minimise the vibrations of the mass m_2, despite the continuing excitations w induced by road unevenness.

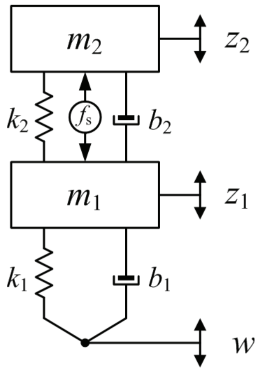

Fig. 1. Quarter-vehicle full active suspension

The following designations are used:

v - voltage controlling the servo-valve, directly affecting the force fs,
w - this input might be treated as excitation due to road unevenness,
z_2 - displacement of sprung mass (to be isolated from vibration).

The detailed synthesis of a nonlinear model of a full-active vehicle suspension with an electro-hydraulic actuator is presented elsewhere [1]. The nonlinear suspension model is compared with a linearised one and the parametric model obtained from identification procedure. The last model, i.e. the parametric model was developed using numerical analysis of data registered in the laboratory setup incorporating an physical model of the suspension.

2.1. Passive suspension

Dynamic properties of the open loop model of a full active structure can be determined on the basis of the model obtained from identification. The control matrix **B** in the model has two components: $\mathbf{B}_v$, $\mathbf{B}_w$, corresponding to two inputs to the system: control input v and excitation w. The model of a quarter vehicle suspension is governed by the equations:

$$\begin{aligned} \dot{\mathbf{x}} &= \mathbf{A}\mathbf{x} + \mathbf{B}_v v + \mathbf{B}_w w \,, \\ y &= \mathbf{C}\mathbf{x}. \end{aligned} \tag{1}$$

Characteristic equation for the matrix **A** is derived from the formula:

$$\begin{aligned} &\left| s\mathbf{I} - \mathbf{A} \right| = 0 \\ &s^6 + 177s^5 + 15713s^4 + 300095s^3 + 6612388s^2 + 14887015s + 8559185 = 0 \,, \end{aligned} \tag{2}$$

Solving Eq (2) yields the eigenvalues of the matrix **A**:

$$\lambda_1 = -1, \quad \lambda_2 = -1.439,$$
$$\lambda_3 = -7.495 + j\,20.570, \; \lambda_4 = -7.495 - j\,20.570,$$
$$\lambda_5 = -79.786 + j\,77.775, \quad \lambda_6 = -79.786 - j\,77.775.$$

Natural frequencies and damping ratios, computed for the eigenvalues of the matrix **A** are:

$$\mathrm{f}_{01} = 0.159 \text{ Hz}, \quad \xi 1 = 1,$$
$$f_{02} = 0.228 \text{ Hz}, \quad \xi_2 = 1,$$
$$f_{03} = f_{04} = 3.484 \text{ Hz}, \; \xi_3 = \xi_4 = 0.342,$$

f05 = f06 = 17.733 Hz, ξ5 = ξ6 = 0.716.

On the basis of the phenomenological model and other models reported in literature [2-6] the following physical quantities are selected as parameters of state of the given object: $x_1 = z_2, x_2 = \dot{z}_2, x_3 = z_1, x_4 = \dot{z}_1, x_5 = P_r = P_d - P_g$ (pressure difference in the actuator chambers). The derivative of work performed by the unit power-supplying the actuator is chosen as the sixth variable of state ($x_6 = P_zQ_z$). Hence, x_6 becomes the measure of power absorbed from the supplying unit.

Figure 2 shows the amplitude-frequency characteristics in relation to the excitation *w*. This is an equivalent of the transfer function defined as the ratio of vibration amplitudes at the output $y = z_2$ to that at the input *w*, expressed in dB, for the frequency range 0.1–100 Hz.

Characteristic (Fig. 2) are based on the simulations of the identified model under the actuator control voltage equal to zero (v = 0). The designed controller ought to reduce the vibration amplification in the neighbourhood of resonance frequency f_0 = 3.484 Hz and reduce their displacement transmissibility in the whole frequency range. From the standpoint of vibration control performance, the transmissibility characteristics is the key measure of quality of the vibration control system.

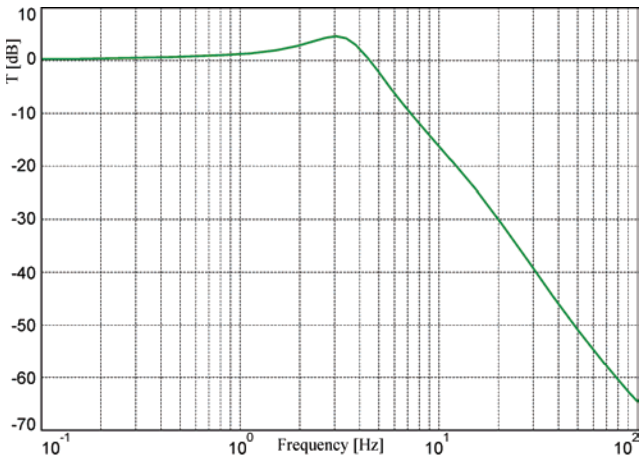

Fig. 2. Vibration displacement transmissibility in the function of frequency

2.2. Active suspension – controller selection

Several control algorithms are chosen on the basis of literature reports and then verified on a physical model. Most active control systems reported in literature utilise conventional PD and PID controllers, optimal controllers and neural networks. Optimal controllers for active vibration isolation include LQ and LQG controllers. Presently a great deal of attention is also given to robust and sliding mode controllers. Most research work in this field involves simulations instead of experiments. Other types of widely adopted controllers include fuzzy controllers, modal controllers and compensating controllers (feed-forward or predictive controllers).

Following a preliminary analysis, three types of controllers were selected to be verified in the laboratory conditions. The final assessment of the controller's adequacy depends on a great number of factors, including: hardware and software features, accessibility of measurement signals, actuator dynamics.

In this study we utilise the selected PID controller, a modal controller with the feedback from the vector of state and an optimal controller with a quadratic quality indicator LQ.

The PID controller was selected because of the simplicity of its synthesis, easy availability of dedicated computer systems and a number of examples quoted in literature on the subject. It is treated as the reference point in the analysis of further controllers.

Underlying the modal control approach is the principle that control of the system can be executed by controlling the system's modes. A pole (eigenvalue) placement method is an example of the modal control technique [7, 8]. This method enables the selection of natural frequencies and relative damping factors, which are key parameters in vibration control, so the Transmissibility of a closed-loop system can be controlled already at the stage of its synthesis [9, 10].

The synthesis of an optimal controller with the quadratic quality indicator requires the minimisation of a certain indicator of control quality. Dynamic optimisation problems reveal several major aspects which prompted the selection of the quadratic quality indicator from the group of optimal controllers. Of particular importance is the fact that a weigh factor is incorporated that limits the control signal and the state trajectory during the control action will remain optimal [11].

3. Experimental investigation

Experiments were run on a physical model of a full active suspension controlled by the controllers mentioned in earlier sections. The physical model of the suspension was fabricated in the laboratory facility for the testing of vibration reduction systems [12].

Actuator control and testing is supported by a dedicated program written in LabVIEW. The program enables the monitoring and recording of the following physical quantities:

z_2 - displacement of sprung mass,
z_1 - displacement of unsprung mass,
w - shaker displacement (excitations),
P_d - pressure in the lower chamber of the cylinder (active element),
P_g - pressure in the upper chamber,
P_z - actuator supplying pressure,
Qz - instantaneous flow rate between the supplying unit and the cylinder,
v - voltage controlling the servo-valve in the actuator system.

Simultaneously, the program executes the actuator control function. The controller assembly incorporates a recorder of state and the controller gain matrix. All controllers are tested in the identical laboratory conditions. Utmost care has to be taken to maintain the thermal conditions of the working fluid on the constant level.

The suspension system controlled by a selected controller is subjected to the applied harmonic excitations of fixed amplitude and frequency increasing in time. The amplitude of the excitation signal is 4 mm whilst frequency increases smoothly from 0.063 to 40 Hz within 60 s. This type of excitation signals allows the required quality indicators to be established. For each tested controller a series of time patterns is provided, enabling the particular systems' performance to be reliably evaluated.

The investigated control systems operate in real time with the predetermined looping time whereby the output signal and the subsequent input signals are duly synchronised. That implies that the control signal at the n-th moment is found on the basis of signals measured at the instant n – 1 and at still earlier time instants, depending on the particular requirements. The sampling time 1 ms is adopted in the testing of all control systems.

3.1. Comparison of various controllers

Figure 3 shows the transmissibility factor in the function of frequency for the sprung mass. It is readily apparent that for systems comprising the PID and modal controllers, their characteristics are quite similar in the pre-resonance and resonance range. For frequencies in excess of 6 Hz, the system comprising a modal controller transmits vibrations onto the sprung mass in a lesser degree. At the same time, comparison of instantaneous power demand levels (Fig. 4) implicates that high vibration isolation efficiency in the high frequency range does not affect the energy consumption. For medium frequencies the transmissibility function in the system comprising a LQ controller expresses fairly similar vibration isolation efficiency than for the two remaining controllers whilst the behaviour of a system with a LQ controller is superior at frequencies in excess of 10 Hz, at the same time featuring the lowest energy consumption. The higher transmissibility levels ensured by the LQ controller in the lowest frequency range (up to 5 Hz) results in the lowest energy consumption and making it independent of the excitation frequency.

The plots of instantaneous power in the function of time shown in Fig. 4 reveal that the largest fluctuations of power absorbed from the supplying source are registered for the system comprising a modal controller.

An overall quality indicator is found, expressing the vibration reduction efficiency. The indicator I_e is obtained in the form of the sum of squared error signals:

$$I_e = \frac{T}{T_c}\sum_{i=1}^{N} e^2(i), \tag{3}$$

where:
T - sampling time,
T_c - total time of measurement,
N - number of samples,
e - control error

To derive a uniform quality indicator characterising the suspension complete with a given controller it is required that energy used by the actuator be determined as the integral of instantaneous power absorbed from the source. In practical applications this energy is derived from the formula:

$$E = \frac{T}{T_c}\sum_{i=1}^{N} P(i). \tag{4}$$

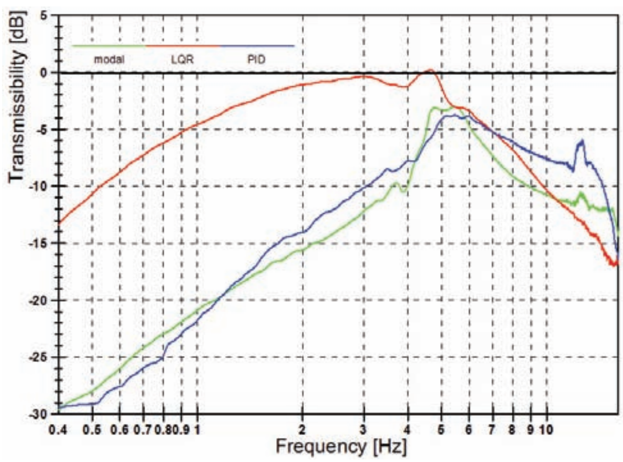

Fig. 3. Transmissibility function from the input (excitation) w to the output z2 for various types of actuator's controllers

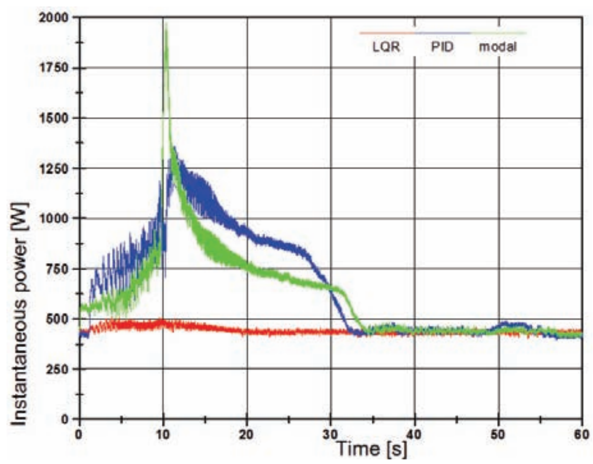

Fig. 4. Comparison of instantaneous power absorbed from the power-supplying unit for various types of actuator's controllers

The overall vibration reduction indicator and the total energy consumption by the system with a parallel-structure full active suspension are summarised in Tab. 1.

Tab. 1. Vibration reduction indices and energy consumption depending on the applied controller

Quality index /controller type	PID	modal	LQ
I_e [mm^2]	0.623	0.466	1.3
E [J]	664.73	620.88	442.67

It appears that in terms of energy consumption the best vibration reduction performance is offered by the system comprising an optimal controller. As far as the vibration reduction efficiency is concerned, the system with a modal controller seems most favourable.

4. System with limited energy consumption

Experiments performed to measure the power excess absorbed from the source supplying the actuator were repeated twice, though the energy parameters were constrained such that power absorbed from the source should not exceed the predetermined limit. Energetic parameters were constrained such that a part of the flux of the working liquid was re-routed from the hydraulic supply unit to the return branch (a oil tank). Since the instantaneous flow rates in the throttling valve depend on the pressure gradient on this valve, the exact, impassable limit cannot be precisely defined for this configuration of the hydraulic system.

In impulse systems, the pressure in the supplying line to the cylinder performing an oscillating motion can be stabilised through the use of a hydraulic accumulator. However, for large amplitudes of cylinder displacements, the pressure-stabilising systems becomes most complicated and requires huge accumulators to be incorporated. In the case analysed here, we observed the differences of pressure supplying the actuator. Hence the predetermined limit of the instantaneous power consumption becomes and arbitrary parameter fluctuating in a narrow range. The analysis of instantaneous power consumption in the function of time for various types of actuator controllers (Fig. 4) enabled us to select two power limit levels: 450 and 750 W. These limits were chosen as averaged instantaneous power levels in the time interval equal to 60 s.

4.1. First degree of the restriction

Figure 5 shows the comparison of instantaneous power intake for various types of actuator's controllers, the power limit being 750 W.

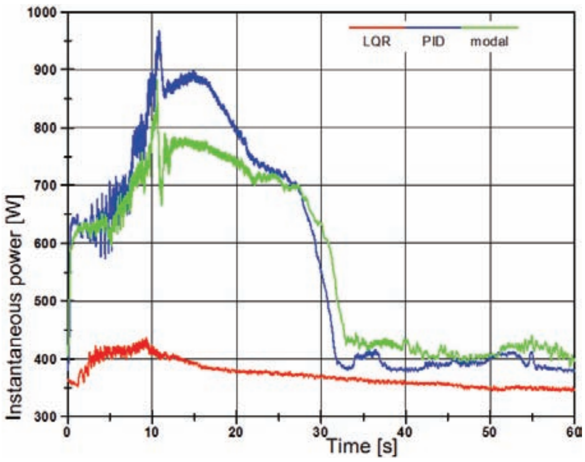

Fig. 5. Comparison of instantaneous power intake from the supplying unit for various types of controllers, the power limit being 750 W

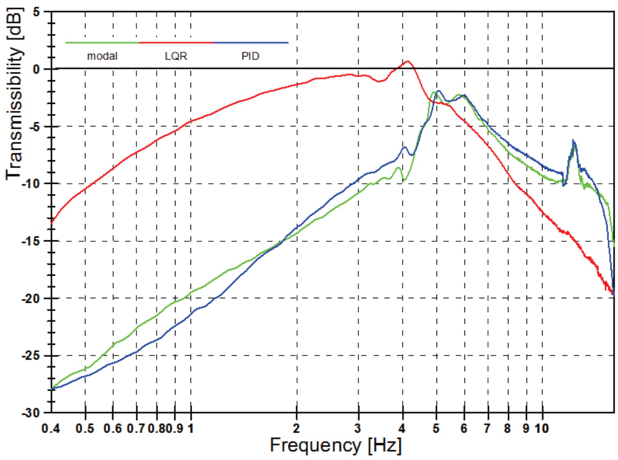

Fig. 6. Transmissibility function from the input w to the output z2 for various types of controllers, the instantaneous power limit being set to be 750 W

It is readily apparent that because of this restriction the power intake level was two times lower at the time instant when the resonance frequency was reached. Tab. 2 summarises the vibration isolation efficiency data and energy consumption depending on the type of applied controller, the power limit being set to be 750 W.

This constraint being imposed, the power consumption was observed to decrease in relation to systems with no restrictions placed on them, no matter what type of applied controller. The restriction level is set to be larger than instantaneous power intake by the system comprising a LQ controller and with no effective power restrictions. Nevertheless, the reduced energy consumption by 70 J was observed also in a system comprising such controller. In other cases the reduction of energy consumption would account for 90 J (in a system with a PID controller) and 54 J, in a system comprising a modal controller.

Tab. 2. Uniform vibration control efficiency indicators and energy consumption depending on the type of applied controller, the power limit being set to be 750 W

Quality index /controller type	PID	modal	LQ
I_e [mm^2]	0.675	0.635	1.168
E [J]	574.96	566.81	372.92

Of particular interest are the effects of the restriction placed upon the instantaneous power intake by the active element on the transmissibility factor of the suspension. Results are shown in Fig. 6.

It appears that transmissibility characteristics for systems comprising a PID and modal controller are similar. In the system with a PQ controller, vibrations are enhanced at 4 Hz and then the system performs correctly at higher frequencies, like in all cases when no restrictions were placed on the power-supply parameters.

4.2. Second degree of the restriction

In further analysis the power limit is set to be 450 W. That is a major restriction in relation to previous experiments. This power limit is set on level roughly equal to the average power consumption by a system comprising a LQ controller. Unfortunately, such harsh restriction placed upon the power level caused that the system comprising a PID controller arrived at the working point near its stability limit.

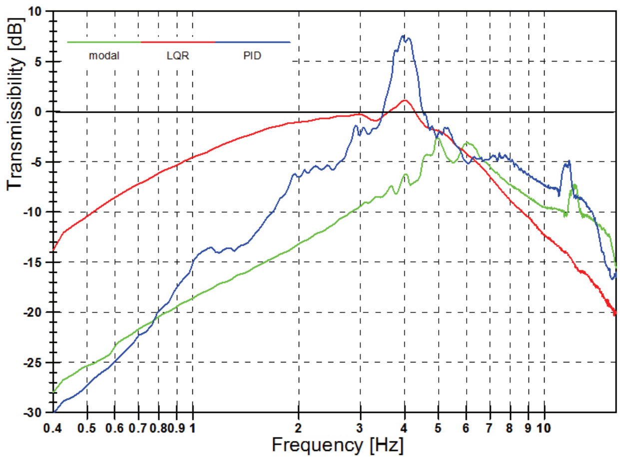

Fig. 7. Comparison of transmissibility function (input w-output z2) for various types of controllers, the power limit being 450 W

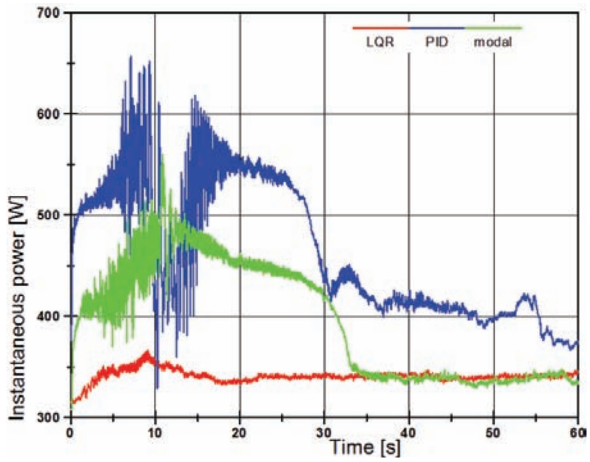

Fig. 8. Comparison of instantaneous power intake for various types of the actuator's controllers, the power limit being 450 W

The range of unstable operation of a system comprising a PID controller is also revealed on the transmissibility characteristics in Fig. 7, showing the transmissibility functions for the systems comprising a PID, modal or LQ controller. Unlike the system with PID controllers, restriction of the instantaneous power intake (450 W) did not cause any major deviations of the transmissibility characteristics for the systems with the modal controller and LQ controller.

Comparison of instantaneous power intake in the function of time for systems with various controllers, the power limit being set to be 450 W is shown in Fig. 8; the values of quality indicators obtained for various controllers are compiled in Tab. 3.

Tab. 3. Overall vibration reduction efficiency indicators and energy consumption depending on the type of employed controller, the power limit being set to be 450 W

Quality index /controller type	PID	modal	LQ
I_e [mm^2]	3.402	0.615	1.260
E [J]	463.56	395.90	341.05

5. Conclusions

Because of the imposed restriction, the value of the energy consumption by the system comprising a modal controller approaches the energy level consumed by the system with a LQ controller. At the same time, the difference between vibration isolation efficiencies, particularly in the low frequency range, remains unchanged. Placing the restriction upon the instantaneous power level (450 W) caused the reduction in energy consumed by the actuator controlled by the modal controller by about 225 J. Therefore, power consumed in this case accounts for about 60% of that consumed by the system with no imposed restrictions. As regards the active system controlled by the LQ controller, adopting this restriction led to the reduction of energy consumption to about 77% of that consumed by the system with no imposed restrictions.

For a suspension controlled by a PID controller, placing too harsh restriction upon the instantaneous power intake from the supplying source led to the major deterioration of the transmissibility features, making the system operation unstable. Minor restrictions of power intake lead to reduced energy consumption by the system and does not lead to any major changes of the transmissibility patterns. When the modal controller is employed, the transmissibility function from the excitation upon the sprung mass will change only slightly, though proportionally to the applied restriction.

For the LQ controller the transmissibility features seemed to deteriorate at near-resonance frequencies, when the harsher restriction is placed upon the power intake from the supply source. For frequencies in excess of 5.5 Hz imposing the limitation of the power level improved the vibration isolation efficiency, which might be attributable to the fact that the stricter power restriction leads to the reduced dynamics of the active element, which encourages weaker vibrations of the unsprung mass. Hence high-frequency vibrations are handled already by the first DOF.

Acknowledgements

The research work has been supported by the Ministry of Science and Education within the project No. N N502 213938 in years 2010-2013.

References

[1] Konieczny, J., *Modelling of the electrohydraulic full active vehicle suspension*, Engineering Transactions / Polish Academy of Sciences Institute of Fundamental Technological Research, ISSN 0867-888X, Vol. 56, issue 3, pp. 247-268, Warsaw 2008.

[2] Hansen, C.H., Snyder, S.D., *Active control of noise and vibration,* E & FN SPON, London 1997.

[3] Hrovat, D., *Survey of advanced suspension developments and related optimal control applications,* Automatica, Vol. 33, No. 10, pp. 1781-1817, 1997.

[4] Yildirim, S., *Vibration control of suspension using a proposed neural network*, Journal of Sound and Vibration, Vol. 277, pp. 1059–1069, 2004.

[5] Yu, F., Crolla, D.A., *An optimal self-tuning controller for an active suspension*, Vehicle System Dynamics, Vol. 29, pp. 51–65, 1998.

[6] Zaremba, A., *Optimal active suspension design using constrained optimization*, Journal of Sound and Vibration 207 (3), pp. 351–364, 1997.

[7] Meirovitch, L., *Dynamics and control of structures*, John Wiley and Sons, New York 1990.

[8] Takahashi, Y., Rabins, M. J., Auslander, D. M., *Control and Dynamic Systems*, WNT, Warsaw 1976.

[9] Kowal, J., Konieczny, J., *Active control of vibration with eigenvalue placement controller*, Proceedings of Inter-noise 2005 International Congress and Exposition on Noise Control Engineering, Rio de Janeiro 2005.

[10] Kowal, J., Konieczny, J., *A Pole placement controller for active vehicle suspension*, Archives of Control Sciences, Vol 15(LI), No. 1, pp. 97–116, Gliwice 2005.

[11] Kwakernaak, H., Sivan, R., *Linear Optimal Control Systems*, John Wiley & Sons, New York 1972.

[12] Konieczny, J., Pluta, J., *Laboratory dynamics and control of structures with fluid elements*, Hydraulics and Pneumatics (Hydraulika i Pneumatyka), ISSN 1505-3954, R. 28 Vol. 5 pp. 32–39, Wrocław 2008.