On Testing of Vehicle Active Suspension Robust Control on An One-Quarter-Car Test Stand

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Abstract: - Between 2010 and 2014, the research team of Josef Bozek´s Research Center of Combustion Engines and Automobiles at the Czech Technical University in Prague, Czech Republic, has developed innovations in vehicle suspension technology. The research team has designed a unique innovative suspension system that uses a linear electric motor as a controlled actuator. We have undertaken many experiments on energy management in the system. In order to verify various control strategies and to test different ways of energy consumption optimization we designed and constructed a unique one-quarter-car test stand on which we have tested different control strategies In order to realize simulation and practical experiments at the test stand it is necessary to find a proper experimental road disturbance signal to excite the active suspension system The disturbance signal is applied on a one more linear motor that is placed under a wheel of the one-quarter-car model to excite the active suspension system The paper deals with the way and results of experimental verification of vehicle active suspension behavior when robust control is applied and also with energy management strategy that we used in the system. A modified H-infinity controller that enables to set energy management strategy is mentioned in the paper. At the close of the paper some experiments taken on the one quarter-car model and their evaluation is discussed.
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Key-Words: - Active suspension, one-quarter–car model, robust control, verification, experiments

1 Introduction

The suspension system is an important aspect of car design because it influences both the comfort and safety of passengers. Two major performance requirements of any automotive suspension system are to provide a comfortable ride and good handling when random disturbances from road unevenness and variable cargo act upon the running vehicle. Passenger comfort can be understood as an attenuation of sprung mass acceleration or as sprung mass vertical displacement minimization, while good handling can be characterized as attenuation of unsprung mass acceleration.

Effort devoted to passive suspension design is ineffective because improvements to ride comfort are achieved at the expense of handling and vice versa. Instead, the best result can be achieved by active suspension [1], [3], i.e. by an additional force (Fig.1) that can act on the system and simultaneously improve both of these conflicting requirements. Another important goal of the control design is to maintain stability and robustness of the closed loop system [2], [9].

In most active suspension systems, the biggest disadvantage consists in energy demands. Regarding linear electric motors, this drawback can be minimized or even eliminated because under certain circumstances there is a possibility to recuperate energy, accumulate it and use it later for the active suspension when necessary. This way, it is possible to reduce the posted claims on an external power source as much as possible. In next paragraphs the proposed strategy how to control the energy distribution in the suspension system is described. In order to regenerate electric power from the vibrations excited by road unevenness a new energy-regenerated active suspension system has been designed. The research team modelled and simulated the active system to show the performance improvement and performed many experiments with the actuator prototype on a new test stand we designed.

All suspension systems are designed to meet various specific requirements. In suspension systems, mainly two most important points are supposed to be improved – vibrations absorbing (videlicet to reach maximal passenger comfort) and attenuation of the disturbance transfer to the road (videlicet car handling). The first requirement could be understood as an attenuation of the sprung mass acceleration or as a peak minimization of the sprung mass vertical displacement. The second one is characterized as an attenuation of the force acting on the road or – in simple car models – as an attenuation of the unsprung mass acceleration. The goal is to satisfy both these contradictory requirements. Satisfactory results can be achieved when an active suspension system generating variable mechanical force acting between the sprung and unsprung masses is used.

Such an actuator can be a linear electric motor [1]. In comparison with traditional actuators that use revolving electro-motors and a lead screw or toothed belt, the direct drive linear motor enables contactless transfer of electrical power according to the laws of magnetic induction. The gained electromagnetic force is applied directly without the intervention of mechanical transmission. Linear electric motors are easily controllable and for features like low friction, high accuracy, high acceleration and velocity, high values of generated forces, high reliability and long lifetime, their usage as shock absorbers seem to be ideal.

2 Problem Solution

2.1 One-Quarter-Car Model

We used a traditional one-quarter-car model to design a suspension controller and to simulate the system behaviour. The basic configuration of the model is shown in Fig. 1.

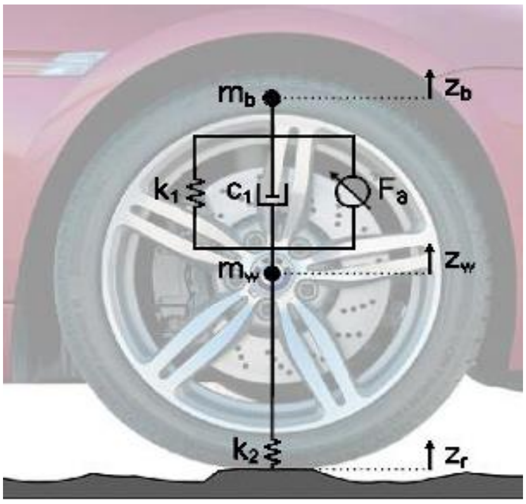


Fig. 1: One-quarter-car model

In Fig.1:

F_a active suspension force [N]
m_w unsprung mass (wheel) [kg]
m_b sprung mass supported by each wheel and taken as equal to a one quarter of the total body mass [kg]
k_2 stiffness of the tyre [N/m]
$z_r(t)$ road displacement (road disturbance) [m]
$z_b(t)$ displacement of the sprung mass [m]
$z_w(t)$ displacement of the unsprung mass [m]
k_1 stiffness of the passive suspension [N/m]
b_1 damping quotient of the passive suspension [Ns/m]

The model involves unsprung (wheel mass) and sprung (taken as one ideal quarter of the car body mass) masses, a conventional passive suspension (a spring and a damper), stiffness of the tire, and linear electric motor as an actuator placed in parallel to the traditional passive suspension (Fig.1).

2.2 Test Stand

We used the same configuration for real experiments and verification. Mechanical construction of the test stand is obvious from Fig. 2. Under the wheel there is placed another linear electric motor that uses an experimental input signal described in next paragraphs to generate road displacement (road deviations) under the running wheel. The modified H-infinity controller mentioned bellow has been developed via Matlab and implemented into dSpace and connected to the test stand system. For more details see [5].

Fig. 2: Test stand

3 Linear Electric Motor

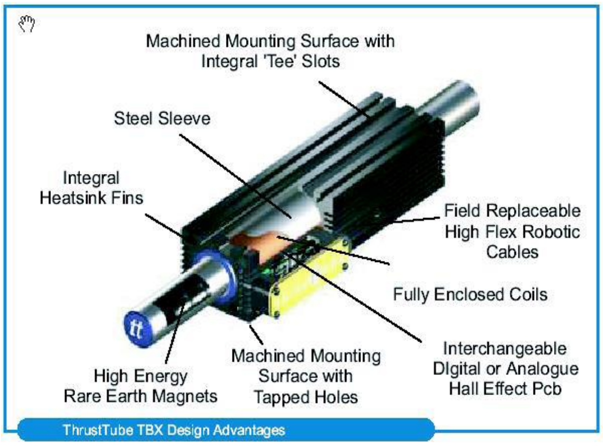


Fig. 3: Linear electric motor - basic structure

Fig.3 shows the basic principle and structure of the linear motor used as an actuator in the designed active suspension system. The appreciable feature of linear motors is that they directly translate electrical energy into usable mechanical force and motion and back. They are linear shaped.

Linear motor translator movements reach high velocities (up to approximately 4 m/s), accelerations (up to g [m/s^2] multiples) and forces (up to 8 kN). The electromagnetic force can be applied directly to the payload without an intervention of mechanical transmission.

3.1 Linear Motor Model

In order to verify control algorithms a linear motor model including a power amplifier in Matlab Simulink has been created. The model enables to demonstrate the conversion of electrical energy to mechanical energy.

In the model, it is assumed that the magnetic field of the secondary part with permanent magnets is sinusoidal, the phases of the primary part coils are star-connected, and a vector control method is used to control the phase current. Here, PWM voltage signal is substituted by its mean value to shorten (about 10 times) the simulation period (inaccuracies caused by such a substitution can be neglected).

The linear motor model is shown in [1]. In the model, an input vector is given (see Fig. 4) by the instantaneous position [m] (necessary to compute the commutation current of the motor coils), instantaneous velocity [m/s] (the induced voltage [V] in the coils depends on the position [m] and velocity [m/s]) and desired force [N] .

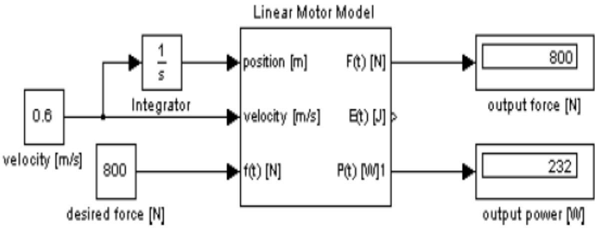
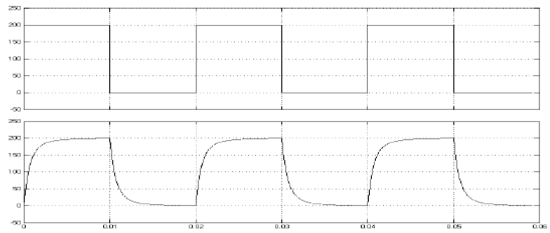


Fig. 4: Linear motor input-output model

The designed model function, we verified comparing dynamics on the simulation model and the real linear electric motor, respectively. The simulation parameters correspond to catalogue parameters of TBX3810 linear motor fy Thrust-tube.

For example, responses caused by changes of the desired force in the form of a right-angular signal and corresponding real time responses has been compared in Fig.5. and b, respectively. The upper signals in both figures represent desired force and the lower ones real output forced.

a)



b)

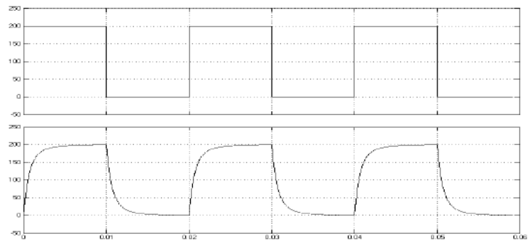


Fig. 5: a) Simulated responses
b) Real time responses

Figs. 5a) and 5b) represent simulated and real time responses, respectively (rightangular force signal : 0→200 [N], power supply of 150 [V], velocity: 0 [m/s]).

For the automotive suspension system, the application of the synchronous three-phase linear motor TBX 3810 produced by Copley Controls Cooperation (technical parameters: peak force 2027N, peak current 21.8A, continuous stall force 293.2N, electrical time constant 1.26ms, continuous working voltage 320V$_{ac}$, maximum phase temperature 100^{o}C) has been designed by the research team.

Comparing the time responses in Fig.4 and Fig.5a) and b) it can be seen a very good matching level of the model and real motor behaviour.

It results from many experiments we made in [4] with TBX3810 linear motor that the designed model describes the real linear motor equipped with necessary auxiliary circuits very authentically and enables to verify control algorithms developed to control the linear motor as an actuator in the active suspension system credibly.

4 Energy Balance

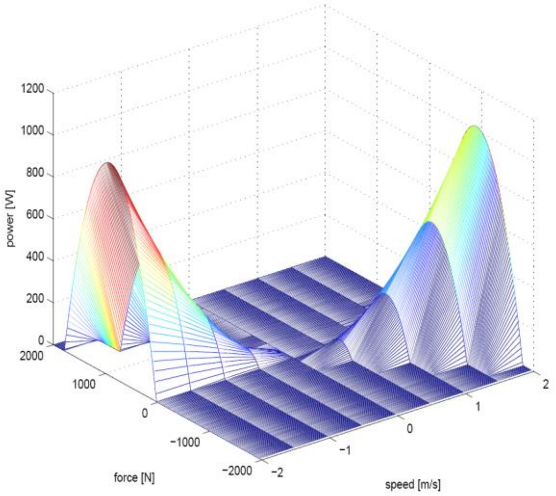


Fig. 6: Force-velocity profile of the recuperated energy of the used linear motor

In fact, there is some non-linearity in the recuperation process and that is why the energy management control is fairly difficult. The 3D plot in Fig. 6 represents a force-velocity profile of the recuperated energy. It shows how much recuperated (and only recuperated) energy can be obtained under the given forces and velocities. In the plot, when the recuperated energy is equal to zero or bigger it is necessary to supply energy into the system.

Regarding to this characteristic surface this gives important information of the requirements on the control system. Optimization objectives are equal to maximization of the recuperated energy (with necessary trade-offs).

5 Road Displacement Simulation

In order to realize simulation and practical experiments at the test stand it is necessary to find a proper experimental signal to excite the active suspension system. Although the simplified suspension model seems to be linear there are many nonlinear parts in the system. Now the question is what signal to generate for experimental testing in order to reach a true model of the uneven road under the wheel. We can define two types of input signals regarding objectives:

- to prove results of simulations and pre-calculations
- to test real behavior on the road

Let us start with the first objective i.e. to verify simulation results. White noise signal is supposed to be the best disturbance signal mainly for of its full frequency spectrum. But it should be noted that the whole system is nonlinear and that is why white noise cannot be used for testing. Moreover it is not possible to generate easily white noise on the test stand.

Unfortunately, neither step input signal simulating road bump up or bump down is not realizable by the test stand for the right-angular slopes of the signal.

To simulate road profile we used a random signal approximation (see Fig.7). Such a signal is described by the following equation:

$$z_r = \sum_{i=1}^{n} \sqrt{\frac{\dot{\omega}_i}{\pi \cdot v_x}} \Big\{ Re\Big(\frac{b_o}{-\omega_i^2 + a_1 j\omega_i + a_o}\Big) \cdot \cos(\omega_i t + \alpha) + $$

$$+ Im\Big(\frac{b_o}{-\omega_i^2 + a_1 j\omega_i + a_o}\Big) \cdot \sin(\omega_i t + \alpha) \Big\} \qquad (1)$$

where

$b_o = 0.121 \cdot v_x$

$a_o = 2.249 \cdot v_x$

and v_x represents car velocity.

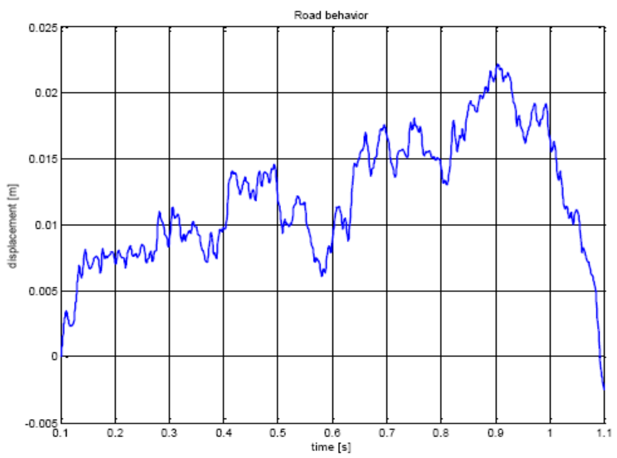


Fig. 7: Random signal approximation used to simulate road displacement

6 Controller

Design of an appropriate controller is a complex problem. The research team tested commonly used high authority control concepts, such as Linear

Quadratic Gaussian control, feedforward control concept, fuzzy control [5] as well as H-infinity robust control [4]. Most of the proposed concepts do not consider robustness of the designed controller, although this aspect is fairly important and well-founded. For example, changes in ambient conditions can lead to changes in the dynamics of the vibration system which can result in substantial deterioration of the feedback controller performance.

A robust controller is necessary for the suspension system because many system parameters often vary in a wide range. Especially cargo mass changes occur for every single drive. For this reason H-infinity theory has been chosen for robust controller design [4], [6] via Matlab toolbox procedures.

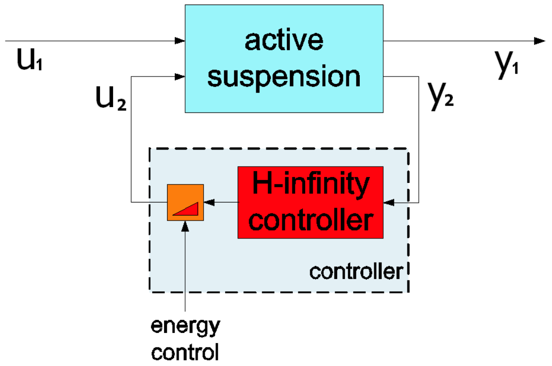


Fig. 8: Modified controller structure

Nevertheless the standard H-infinity controller cannot handle energy consumption. We did some modifications of the controller on the base of the realized experiments. An additional input to control energy demands is supposed to be connected to the controller. Then master controller can use this input to keep energy balance. The general modified controller structure is shown in Fig. 8. Energy management is controlled by an external signal depending on car and road parameters, i.e. on energy accumulator (super-capacitors) current capacity and the road surface, respectively [5].

First way to control energy consumption in the suspension system is analysis of the driving conditions and cyclic re-computing of the control signal in real time. For necessary high sampling frequency (over 1 kHz) and controller performance we rejected this approach.

The second possibility of energy consumption control is control via controller deterioration. Then the designed controller is reliably robust and the active suspension system relatively stable.

Let us assume two types of driving conditions:

- the terrain (surface) the car is going on is very rough and uneven and there is enough energy stored in the accumulator (supercapacitors). Then the controller works in “comfort setting” mode, the linear electric motor consumes energy from the accumulator and the suspension performance is preserved.

. ● the terrain (surface) under the car wheels is relatively smooth and there is not enough energy stored in the accumulator systems (super-capacitors) because of the situation described above. The external signal provides this information to the controller in order to deteriorate its performance and to reduce the linear motor energy consumption. The deterioration is stated by the desired force attenuation. In this case the controler works in “energy setting” mode.

If the force is attenuated too much the active suspension system works only as the passive suspension (connected in parallel to the active suspension) while the linear electric motor works as a generator generating electrical energy to be stored in supercapacitors. Of course, in such a situation the suspension performance is deteriorated (down to the passive suspension level in the worst case).

7 Quantification

7.1 Car Stability

The first requirement in the active suspension system is to improve car stability and reach “road friendliness” that can be characterized as the attenuation of the tire pressure, or more precisely the attenuation of the unsprung mass force acting on the road. To gain a measurable parameter, the following RMS function has been introduced:

$$J_{stab} = \sqrt{\int_0^T (z_w - z_r)^2 dt} \qquad (2)$$

where z_w represents wheel displacement and z_r road displacement.

7.1 Passenger Comfort

Another important requirement in the active suspension system is to improve passenger comfort. This requirement can be formulated as the sprung mass acceleration attenuation when the RMS function is defined as:

$$J_{comf} = \sqrt{\int_O^T G_w * \ddot{z}_b^2 dt} \qquad (3)$$

where $\ddot{z}_b$ represents body acceleration, G_w is a weighting function for human sensitivity to vibrations and * denotes convolution.

7 Experiments

We used random signal approximation stated in (4) and plotted in Fig. 8 has been taken as an input signal (road profile) for experiments taken on the active suspension test stand. Two main objectives must be taken into account - passenger comfort improvement and energy consumption (see Fig. 9). In Fig.9, the corresponding body displacement for “energy” and “comfort” controller settings to the road profile input are displayed. Body (sprung mass) displacement can be taken as an indicator of the passenger comfort.

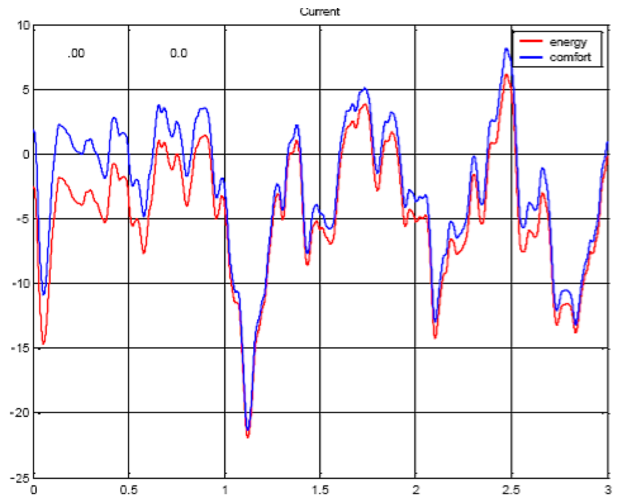


Fig. 9: Body displacement for “energy” and “comfort” controller settings

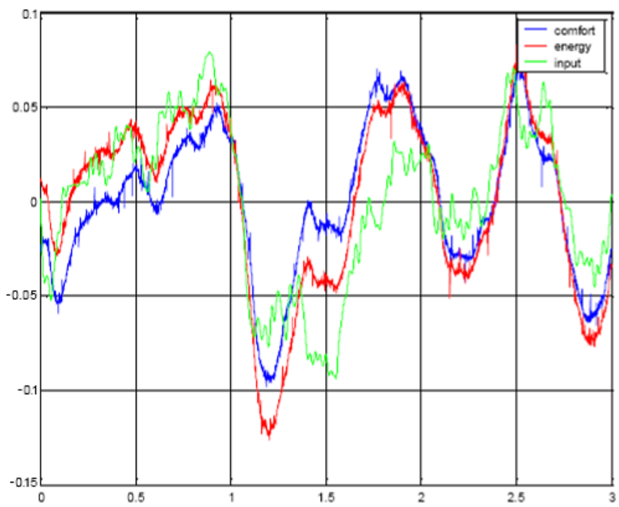


Fig. 10: Demands on electric current as an indicator of energy demands for “energy” and “comfort” controller settings

For “comfort” controller setting, passenger comfort is maximized with no respect to energy consumption while for “energy” controller setting the controller performance is deteriorated and energy recuperated with no respect to passenger comfort.

Fig.10 .displays two curves- energy demands for “comfort” controller setting and energy demands for “energy” controller setting. Both curves were measured when excited with the same road profile input signal (green curv) plotted in Fig .9. Negative values of energy in Fig.10 represent recuperated energy.

Let us show the results as mean values of comfort and energy indicators stated in the previous paragraph. Table I involves mean values for the body displacement as absolute values of the defined body displacement indicator and also as a percentage of its improvement.

Table 1
Body displacement mean values

	Body displacement mean value [m]	**Percentage**
For Comfort setting	0,0386	100%
For Energy setting	0,0474	123%

Table 2
Power mean values

	El.Current mean value [A]	**Percentage**
For Comfort setting	2,689	100%
For Energy setting	1,598	59%

In Table I, the first table row shows that for “comfort” controller setting the comfort indicator, i.e. body displacement mean value (0,0386m) is taken as 100 %. In the second row, when the controller was deteriorated, the comfort was devaluated up to 0,0474 m, i.e. to 123 %.

Table II contains mean values of power indicator i.e. electric current mean values. Lower value corresponds to lower energy demands. It is sufficient to express power values by values of electric current because the voltage supply of the motor remains constant. Similarly, the first line shows that for standard controller setting the “comfort” indicator (2,698 A) is taken as 100 %.

In the second line, when the controller was deteriorated the energy indicator is devaluated down to 1,598 A, i.e. to 59 %. Briefly, controller deterioration causes comfort devaluation to 123 % while energy consumption decreases to 59 %.

8 Conclusion

We developed a new method for direct real-time energy control in the car suspension system with respect to reduction of energy consumption. Experiments that we took on the designed test stand verified validity of simulations and showed that it is possible to change energy demands according to the road surface mode of the energy storage in the car supercapacitors. The method can be extended to general plants with considerable energy demands in which the decreasing actuator signal in a given range can preserve system stability.

Various active suspension controllers we developed via Matlab implemented into dSpace and connected to the test stand system. As we discussed in [5] and [6], the most satisfactory responses of the suspension system have been reached using H-infinity robust control.

8 Acknowledgement

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