Active Suspension System with Linear Electric Motor

Katerina Hyniova, Antonin Stribrsky, Jaroslav Honcu and Ales Kruczek
Department of Control Engineering, Faculty of Electrical Engineering
CTU Prague
Karlovo namesti 13, Prague 2, 12135
CZECH REPUBLIC
stribrsk@fel.cvut.cz, hyniova@fel.cvut.cz, honcu@fel.cvut.cz, kruczea@ fel.cvut.cz

Abstract: - Suspension system is an important part of the car design, because it influences both the comfort and safety of the passengers. In this paper an active suspension using linear electric motor is designed. This article is also focused on experiments with active suspension – developing an appropriate input signal for the test bed and evaluation of the results. Second important aspect of the active suspension design is energy demand of the system. Modification of the standard controller which allows changing amount of energy required by the system has been designed. Performance of the modification was verified taking various experiments.

Key-Words: - robust control, active vehicle suspension, linear motors, energy control

1 Introduction

Increased competition on the automotive market has forced companies to research alternative strategies to classical suspension systems. The basic function of the vehicle suspension is to support the weight of the car, maximize the friction between the tires and the road surface, provide steering stability with good handling and ensure sufficient comfort of the passengers. In order to improve handling and comfort performance instead of a conventional static spring and damper system, semi-active and active systems are being developed. There are, of course, numerous variations and different configurations of suspension. In experimental active systems, the force input is usually provided by hydraulic actuators. As an alternative approach to active suspension system design electromechanical actuators are being studied. Such actuators would provide a direct interface between electronic control and the suspension systems.

Nowadays, the theoretical research of the research team concerning the active suspension of mechanical vibrations and improving ride comfort and handling properties of vehicles is concentrating on various suspension innovations.

In most active suspension systems, the biggest disadvantage consists in energy demands. Regarding linear electric motors, this drawback can be eliminated because under certain circumstances there is a possibility to recuperate energy, accumulate it and use it later when necessary. This way, it is possible to reduce or even eliminate the demands concerning the external power source. In the next chapters also the proposed strategy how to control the energy distribution is going to be described. In order to regenerate electric power from the vibrations excited by road unevenness a new energy-regenerative active suspension for vehicles has been proposed, then the active system has been modelled and simulated to show the performance improvement and the performance tests of the actuator prototype testing stand have been carried out.

All suspension systems are designed to meet specific requirements. In suspension systems, usually two most important features are expected to be improved - disturbance absorbing (i.e. passenger comfort) and attenuation of the disturbance transfer to the road (i.e. car handling). The first requirement could be presented as an attenuation of the damped mass acceleration or as a peak minimization of the damped mass vertical displacement. The second one is characterized as an attenuation of the force acting on the road or - in simple car model - as an attenuation of the unsprung mass acceleration. It is obvious that there is a contradiction between these two requirements. With respect to these contradictory requirements the best results can be achieved using active suspension systems generating variable mechanical force acting in the system using a linear electrical motor as the actuator. Compared to traditional drives using rotational electro-motors and lead screw or toothed belts, the direct drive linear motor exhibits the property of contact-less transfer of electrical power according to the laws of magnetic induction. The electromagnetic force is applied directly without the intervention of a mechanical transmission. Low friction and no backlash resulting in high accuracy, high acceleration and velocity, high force, high reliability and long lifetime enable not only effective usage of modern control systems but also represent the important attributes needed to control vibration suspension efficiently.

2 The Suspension Model and Test Bed

A simple model has been developed for controller design and simulations. The simplest car model is a one-quarter-car model. The basic schema is presented in Fig.1
.

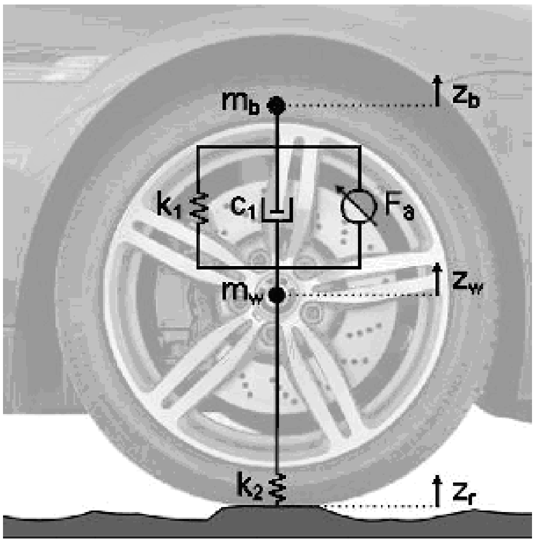

Fig.1 One-quarter-car model

The motion equations for the one-quarter-car model illustrated in the Fig.1 can be derived to describe the system. Therefore the equations are [2]:

$$m_b \ddot{z}_b = F_a - k_1(z_b - z_w) - c_1(\dot{z}_b - \dot{z}_w) \quad (1)$$
$$m_w \ddot{z}_w = -F_a + k_1(z_b - z_w) + c_1(\dot{z}_b - \dot{z}_w) - k_2(z_w - z_r)$$

where:
z_r . . . road vertical displacement,
z_w . . . axle vertical displacement,
z_b . . . car body vertical displacement,
F_a . . . power of linear power source,
m_b . . . body mass = 250kg,
m_w. . .wheel mass and unsprung mass = 35kg,
c_1 . . . damping constant = 1160Nsm^{-1},
k_1. . . stiffness constant of spring = 15kNm^{-1}.
k_2. . . stiffness constant of tires = 115kNm^{-1},

2.1 State-space model

The state and input variables has been chosen as follows:

$$x_1 = z_b - z_w$$
$$x_2 = z_w - z_r$$
$$x_3 = \dot{z}_b$$
$$x_4 = \dot{z}_w$$
$$u_1 = \dot{z}_r$$
$$u_2 = F_a$$

Then the conversion from the differential equations to the state-space matrices (2) is as follows.

$$\dot{x} = Ax + B_1 u_1 + B_2 u_2 \quad (2)$$

2.2 Simulation schema

Simulations have been performed for a one-quarter car model, a half car model and a full car model. Because of the one-quarter car character of the test bed, only one-quarter car model behavior could be verified.
A reduced schema of the full car model is shown in Fig.2.

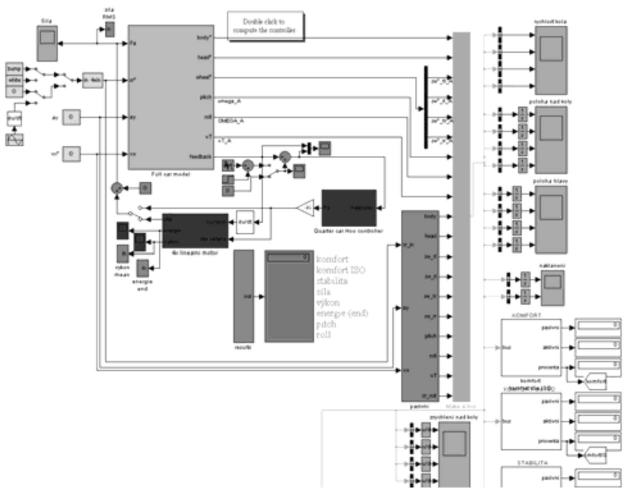

Fig.2 Full-car model (reduced schema)

A power source of displacement has generated the input signal i.e. road deviations using experimental signal described in next paragraphs. Mechanical configuration of the test bed is obvious in Fig.3.
As will be mentioned later the controller has been developed using MATLAB software. Resulted controller had been implemented into dSpace a connected to the test bed system.

Fig.3 Test bed

3 Linear Electric Motor

Linear electric motor has been used as an actuator generating required forces. Fig.4 represents the basic principle and configuration of the linear motor. The beauty of linear motors is that they directly translate electrical energy into usable linear mechanical force and motion, and vice versa. The motors are produced in synchronous and asynchronous versions. Compared to conventional rotational electro motors, the stator and the shaft (translator) of direct-drive linear motors are linear-shaped. One can imagine such a motor taking infinite stator diameter.

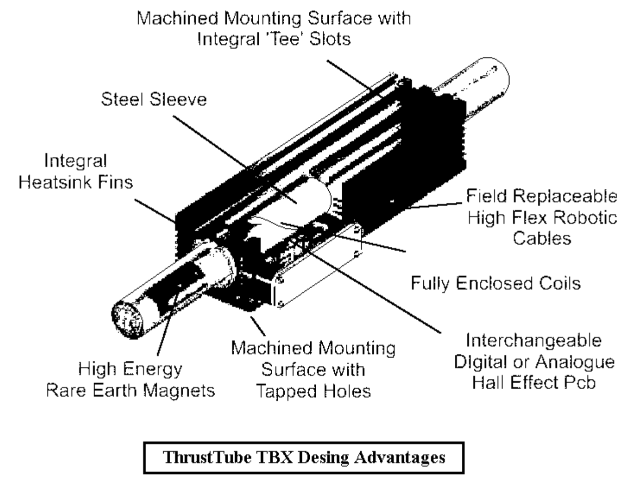

Fig.4 Linear motor - basic design (adapted manufacturer spreadsheet)

Linear motor translator movements take place with high velocities (up to approximately 200m/min), large accelerations (up to g multiples), and forces (up to kN). As mentioned above, the electromagnetic force can be applied directly to the payload without the intervention of a mechanical transmission, what results in high rigidity of the whole system, its higher reliability and longer lifetime. In practice, the most often used type is the synchronous three-phase linear motor.

It is necessary to answer one important question - if it is more advantageous to include the model of the linear electric motor in the model for active suspension synthesis or if it should be used only for simulations.

Comparing advantages and disadvantages of the model inclusion, it can be said that the closed-loop provides more information so that better control results can be achieved. Unfortunately, there are also some significant disadvantages in such a solution. The first one insists in the rank of the system (and consequently the rank of the controller which increases up to 5) and the second one is that the D matrix in the state space description of the motor model does not have full rank and that is why implementation functions are limited or too complicated. On the base of this comparison the linear motor has not been included in the model for active suspension synthesis.

There is another important question whether the linear motor model could be omitted and a linear character of the desired force could be supposed. The answer is “yes”. Both the mechanical and the electrical constants are very small – just about 1ms. Moreover it will be shown that the robustness of the H_∞ control design has been verified using numerous simulation results and experiments.

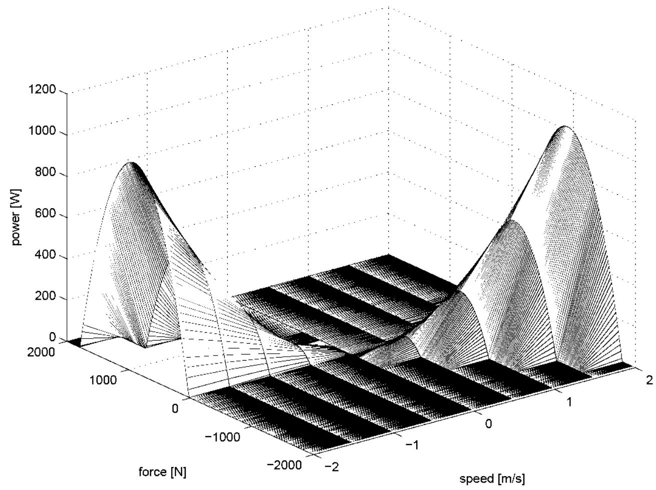

Fig.5 Recuperation range in linear motor

4 Experimental Signal

The most important step of the experiment design is the choice of a proper experimental signal representing a real road surfce. Although the simplified suspension model seems to be linear the truth is that there exist many nonlinear parts. It implies the results will depend on inputs.
Now the question is how to choose the appropriate signal for experimental testing. We should define two types of input signals to conclude two objectives:

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

Let's begin with the first – to verify simulation results. The best signal is probably white noise, because we can observe full frequency spectrum. But it should be noted it isa nonlinear system and even white noise is not sufficient. Moreover it is not possible to generate easily white noise on the test bed because there is none analytical expression of the signal.
A bump has been chosen as a signal which can be generated by the test bed. This signal allows observing both directions – up and down. It is not possible to generate infinite slope and during experiments the following signal has been used (Eq.3). Different magnitudes of signal should be tested, because the system is nonlinear. Magnitudes has been chosen according to mechanical dimensions of the suspension system:

$$\dot{z}_r(t) = 0.5\pi \sin 20\pi (t - 0.1) \qquad (3)$$

Thus this signal is used to verify the quality of the simulation model, but does not give the usability of the controller at a real road situation.
More important results will be obtained using an input signal which is similar to the real road behavior. Deterministic random signal has been used to simulate it. The input signal for simulation can be described by thefollowing equation:

$$z_r = \sum_{i=1}^{n} \sqrt{\frac{\dot{\omega}}{\pi \cdot v_x}} \{ \mathrm{Re}(\frac{b_o}{-\omega^2 + a_1 j\omega + a_o}) \cdot \cos(\omega t + \alpha_i) + \mathrm{Im}(\frac{b_o}{-\omega^2 + a_1 j\omega + a_o}) \cdot \sin(\omega t + \alpha_i) \}$$

$$b_o = 0.121 \cdot v_x \qquad (4)$$
$$a_o = 2.249 \cdot v_x$$
$$a_1 = 30.36 \cdot v_x$$

where v_x in Eq.2 represents the car velocity and α_i are randomly generated angles.

Thus resulted signal is obtained as a superposition of the sinusoids with deterministic "random".In this case 128 random angles have been calculated. The used signal is plotted in Fig.6.

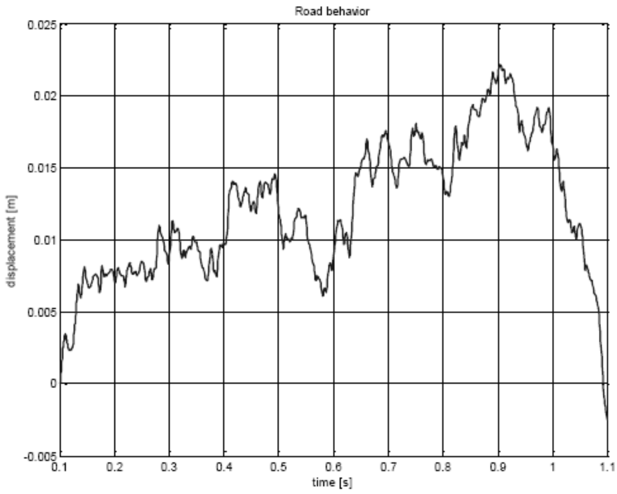

Fig.6 Random signal

5 Controller

The controller for active suspension we have designed using H_∞ theory. The standard H_∞ control scheme is shown in Fig. 7. In this case the controller is robust enough when system parameters vary in a wide range.When the open loop transfer matrix from u_1 to y_1 is denoted as T_{y1u1}, then the standard optimal H_∞ controller problem is to find all admissible controllers $K(s)$ such that $\|T_{y1u1}\|_\infty$ is minimal, where $\|\cdot\|_\infty$ denotes the H_∞-norm of the transfer function (matrix). For more information, see [1].
The H_∞ controller is stated minimizing the $\|T_{y1u1}\|$-norm. In addition, it is possible to shape open loop characteristics to improve performance of the whole system.
For the active suspension system the performance and robustness outputs should be weighted. The performance weighting has to include all significant measures as comfort and car stability (body speed, suspension displacement, actuator force, etc). For the linear electric motor in the position of an actuator, an additional weight should be added to control maximum force, energy consumption and robustness of the system.

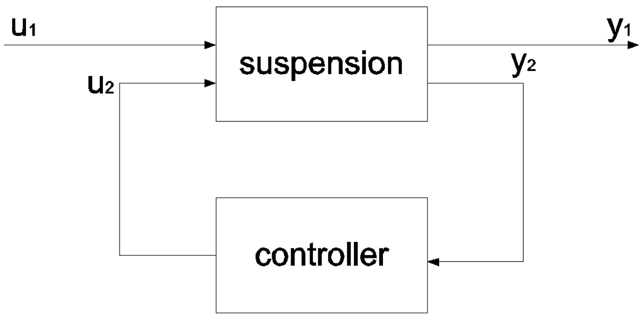

Fig.7 H_∞ control scheme

The plant augmentation must be done[2].

5.1 Plant augmentation

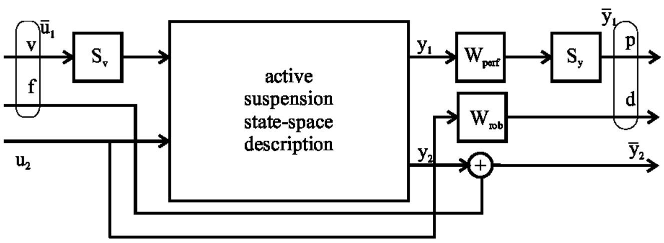

Fig.8 Augmentation scheme

The schematic diagram of a plant augmentation is shown in the Fig.8. The first input signal v represent the road disturbance and is scaled by factor S_v. The first group (port) includes the input f too, which represents a disturbance signal acting on the measured feedback. Then the second input is desired value for the actuator. And as for outputs, the first output y_1 consists of two parts:

(1) nominal system output (states in our case) weighted by MIMO function $W_{per\ f}$ and scaled by constant matrix S_y and
(2) actuating signal weighted by W_{rob}. The second output is formed as a sum of nominal output y_2 and disturbance input f.

As for robustness function W_{rob}, to avoid some singularity problems in zero of nominal transfer function from u_2 to y_2 the additive uncertainty has been used. Whereas the function W_{rob} doesn't mean the conventional robustness stability function, but W_{rob} is supposed to be total weighting by reason of linear motor nonlinearities, feedback branch disturbances and other uncertainties.
At first, it is necessary to choose the weighting functions, then what kind of feedback will be established. In this design output feedback has been used. Suspension speed has been chosen as a measured output (y_2), because for a linear motor control the speed has to be measured as well.
Moreover signals, which affect the important characteristics, have to be weighted in controlled output (y_1). Influence of weighting functions and constants is mainly evident, moreover it has been checked by simulations. So the first output is as follows:

(1) $x_1 \equiv z_b - z_w$ ... weighted by constant, improve steady-state deviation value.
(2) $x_2 \equiv z_w - z_r$ ... weighted by constant, improve process of deviation stabilization.
(3) $x_3 \equiv \ddot{z}_b$ ... weighted by function, shape the sprung mass speed. It is a function ($W_{per\ f1}$), because the different shaping at each frequency is needed according to the human sensibility. This weighting imply the car comfort improvement character.
(4) $x_4 \equiv \ddot{z}_w$ … weighted by function, shape the unsprung mass acceleration. It is a function ($W_{per\ f2}$), because this variable influences the vehicle stability and thus there should be the possibility to change the vehicle properties in the frequency domain.
(5) W_{rob} ... finally the function W_{rob} is weighted by constant to find the optimal value for car requirements satisfaction.

The appreciative reader could notice that accelerations has been weighted instead of speeds. In the frequency domain the only difference between speed and acceleration is in one integrator, thus in the slope of magnitude frequency characteristic (phase is practically not interesting in H_∞ case). Therefore to ovoid some problems for frequency tending to infinity, the sprung and unsprung speed has been weighted (see complementary weighting function $1/W_{per\ f1}$ in Fig.9).
So the states are used as controlled output (y_1) and the variable $\dot{z}_b - \dot{z}_w$ is used as measured output (y_2).
Then obtaining the matrices C_1, C_2 and $D_{11} \dots D_{22}$ to complete the state-space model of the nominal plant is simply now and thus the nominal state-space (Eq.2) are completed by:

$$y_1 = C_x + D_{11}u_1 + D_{12}u_2 \qquad (5)$$
$$y_2 = C_x + D_{21}u_1 + D_{22}u_2$$

The function *tf2ss* in MATLAB is used for transformation of the weighting transfer function to the statespace description in controller canonical form. Dynamics of the all weighting function and weighting constants are put together and one multiple input multiple output system is obtained.

First the functions $W_{per\,f1}$ and $W_{per\,f2}$ are converted to one MIMO system $W_{per\,f}$ .and then it is transformed into state-space description:

$$W_{per\,f}(s) \rightarrow W_{per\,fA}, W_{per\,fB}, W_{per\,fC}, W_{per\,fD}$$
$$W_{rob}(s) \rightarrow W_{robA}, W_{robB}, W_{robC}, W_{robD}$$

Therefore it is possible to augment the nominal plant. The states of the augmented plant are as follows:

$$\bar{x} = \begin{bmatrix} x \\ x_{Wperf} \\ x_{Wrob} \end{bmatrix}$$

From the state-space equations mentioned above and the schematic diagramin Fig. 8 it can be written in the matrix form:

$$\dot{\bar{x}} = \begin{bmatrix} A & 0 & 0 \\ W_{perfB}C & W_{perfA} & 0 \\ 0 & 0 & W_{robA} \end{bmatrix} \begin{bmatrix} x \\ x_{Wperf} \\ x_{Wrob} \end{bmatrix} + \\ + \begin{bmatrix} BS_v & 0 \\ W_{perfB}D_{11}S_v & 0 \\ 0 & 0 \end{bmatrix} \bar{u}_1 + \begin{bmatrix} B_2 \\ W_{perfB}D_{12} \\ W_{robB} \end{bmatrix} u_2$$

$$\bar{y}_1 = \begin{bmatrix} S_y W_{perfD} C & S_y W_{perfC} & 0 \\ 0 & 0 & W_{robC} \end{bmatrix} \bar{x} + \\ + \begin{bmatrix} S_y W_{perfD} D_{11} S_v & 0 \\ 0 & 0 \end{bmatrix} \bar{u}_1 + \begin{bmatrix} S_y W_{perfD} D_{12} \\ W_{robD} \end{bmatrix} u_2 \qquad (6)$$

$$\bar{y}_2 = \begin{bmatrix} C_2 & 0 & 0 \end{bmatrix} \bar{x} + \begin{bmatrix} D_{21}S_v & 1 \end{bmatrix} \bar{u}_1 + \begin{bmatrix} D_{22} \end{bmatrix} u_2$$

Let:

$$\begin{bmatrix} A & 0 & 0 \\ W_{perfB}C & W_{perfA} & 0 \\ 0 & 0 & W_{robA} \end{bmatrix} = \bar{A}$$

$$\begin{bmatrix} x \\ x_{Wperf} \\ x_{Wrob} \end{bmatrix} = \bar{x}$$

$$\begin{bmatrix} BS_v & 0 \\ W_{perfB}D_{11}S_v & 0 \\ 0 & 0 \end{bmatrix} = \bar{B}_1$$

$$\begin{bmatrix} B_2 \\ W_{perfB}D_{12} \\ W_{robB} \end{bmatrix}' = \bar{B}_2$$

$$\begin{bmatrix} S_y W_{perfD} C & S_y W_{perfC} & 0 \\ 0 & 0 & W_{robC} \end{bmatrix} = \bar{C}_1$$

$$\begin{bmatrix} S_y W_{perfD} D_{11} S_v & 0 \\ 0 & 0 \end{bmatrix} = \bar{D}_{11}$$

$$\begin{bmatrix} S_y W_{perfD} D_{12} \\ W_{robD} \end{bmatrix} = \bar{D}_{12}$$

$$\begin{bmatrix} C_2 & 0 & 0 \end{bmatrix} = \bar{C}_2$$

$$\begin{bmatrix} D_{21}S_v & 1 \end{bmatrix} = \bar{D}_{21}$$

$$\begin{bmatrix} D_{22} \end{bmatrix} = \bar{D}_{22}$$

Then (6) can be simplified as:

$$\begin{aligned} \dot{\bar{x}} &= \bar{A}\bar{x} + \bar{B}_1\bar{u}_1 + \bar{B}_2 u_2 \\ \bar{y}_1 &= \bar{C}_1\bar{x} + \bar{D}_{11}\bar{u}_1 + \bar{D}_{12} u_2 \\ \bar{y}_2 &= \bar{C}_2\bar{x} + \bar{D}_{21}\bar{u}_1 + \bar{D}_{22} u_2 \end{aligned} \qquad (7)$$

5.2 Weighting functions

Weighting function $W_{per\,f1}$ shapes the attenuation of the sprung mass acceleration. Human being is most sensitive in the frequency range between 4 and 8Hz, thus between 25 and 50rad/s. Then function $W_{per\,f1}$

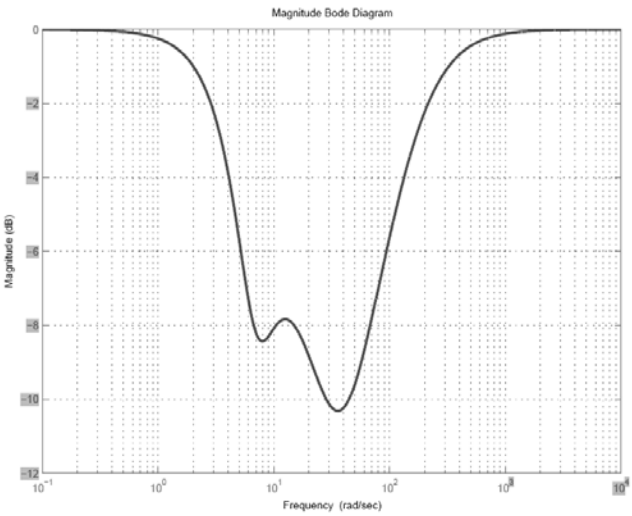

Fig.9 Weighting function $1/W_{per\,f1}$

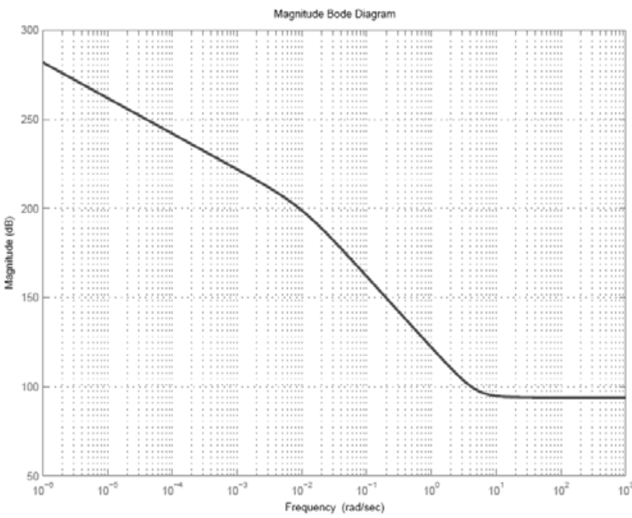

Fig.10 Weighting function $1/W_{rob}$

has to be a band-pass filter. Complementary weighting function $1/W_{per\,f1}$ is shown in Fig.9, where small correction at frequency 7rad/s can be seen. This correction has been obtained by trial and error method to achieve the best acceleration shaping.
After simulation tests, weighting function $W_{per\,f2}$ had to be considered constant.
Last weighting function is W_{rob}. This function should, as has been mentioned above, respect the motor non-linearity and some inaccuracies and disturbances in the feedback loop. Complementary function $1/W_{rob}$ is plotted in Fig.10.

6 QUANTIFICATION

Some quantitative measures have to be defined to evaluate the results achieved by the closed loop system and to compare the active and passive systems.

6.1 Car stability

First requirement in the active suspension system is to improve car stability and “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 get a measurable parameter, the following RMS function can been introduced:

$$J_{stab} = \sqrt{\int_0^T (z_w - z_r)^2 dt} \qquad (8)$$

where z_w represents wheel displacement and z_r road displacement.

6.2 Passenger comfort

Second 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:

$$J_{comf} = \sqrt{\int_0^T G_w * \ddot{z}_b{}^2 dt} \qquad (9)$$

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

7 ENERGY CONTROL

7.1 Energy control principles

The H_∞ controller has been designed using appropriate weights to optimize minimum of the energy consumption with respect to the performance.
In the car, where the working conditions change according to the various drive situations it is very difficult (if possible) to say in general what level of performance is sufficient enough and how much energy can be obtained. It would be optimal to find a possibility of real-time control of the energy consumption. The energy management is supposed to be controlled by an external signal depending on the car and road parameters, i.e. on the energy accumulator capacity and on the road surface, respectively.
First possibility consists in the analysis of the driving conditions and cyclic re-computing of the control signal in real-time. While the time requirements of the H_∞ controller design are too high (sampling period has to be less than 1ms!) and moreover the performance of the H_∞ controller cannot be guaranteed for all operating conditions this approach has been rejected.
The second possibility is to control the energy consumption by controller deterioration. Then the designed H_∞ controller is reliably robust and the active suspension system is relatively stable.
Let´s assume two driving conditions:

- the terrain/surface the car is driving on is very rough and uneven and there is enough energy stored in the accumulator system - then the controller works in the standard mode, the motor consumes energy from the accumulator and the suspension performance is preserved.
- the terrain/surface the car is driving on is relatively smooth and there is not enough energy

stored in the accumulator system because of the situation described above. The external signal provides the information to the H_∞ controller to deteriorate its performance and to reduce the energy consumption. The deterioration is stated by the desired force attenuation.

If the force is attenuated too much then the active suspension system works similarly to the passive suspension and the linear electric motor works as a generator producing energy for the accumulator system. Of course, the suspension performance is deteriorated now (to the passive suspension level in the worst case). The influence of these controller modifications to the suspension system performance we will be discussed in the section bellow.
The principle of the proposed energy management strategy is illustrated in Fig.11. The H∞ controller is extended by the variable gain block controlled by the external signal (energy management control input) [3].

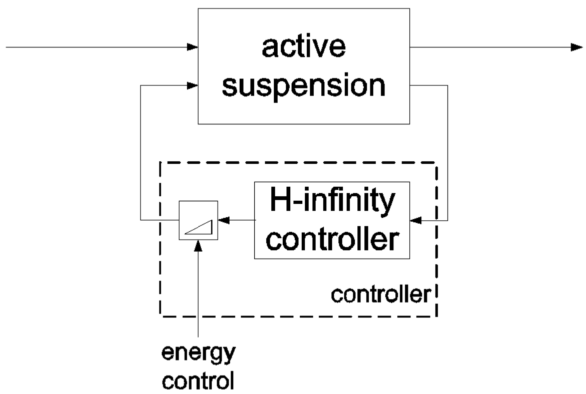

Fig.11 Energy control scheme

7.2 Energy management analysis

In paragraph 7.1, the energy management has been discussed as an extension of the H_∞ controller abilities. Now the influence on the performance and robustness will be presented. The H_∞ controller is deteriorated by the desired force attenuation using the input coefficient that is given by a superior controller.
At first robustness tests have to be done to find the range of the input coefficient in the energy management block. To test robustness the direct numerical method has been chosen because the rank of the closed system is relatively small (4 for plant + 6 for weights = 10 for system, 10 for system + 10 for controller = 20 total for closed loop rank). Hence the poles have been tested for stability for a given input coefficient range.

The stability test in graphical form is shown in Fig.12 and Fig.13. In the figures, closed loop poles are plotted for the input coefficient range of (-0.5 ÷ 1.7). Zoomed surroundings of the stability region from Fig.12 is shown in Fig.13. The original H_∞ pole placement is presented by *, pole placement for stable region by # and unstable region by ■, respectively.

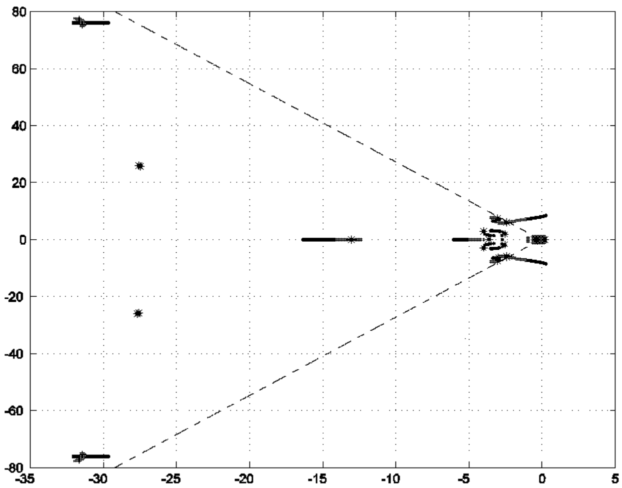

Fig.12 Pole plot in energy control

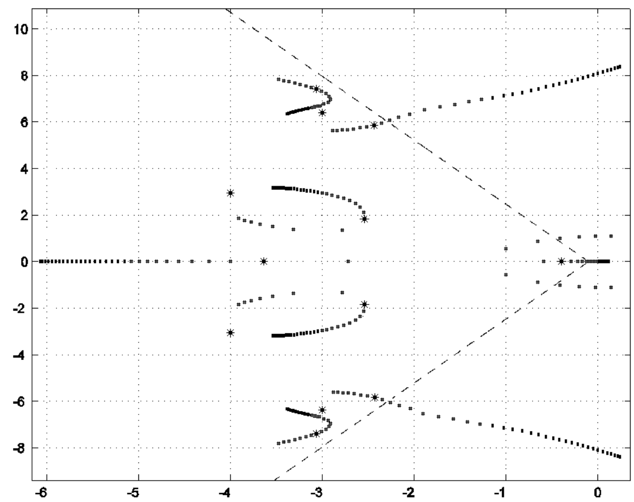

Fig.13 Pole plot in energy control (zoom)

On the base of the test mentioned above, we have stated the maximum and minimum stable input coefficients. To achieve stability the coefficient must not exceed the range of (0.000 ÷ 1.613).
The coefficient range should be determined to achieve also certain robustness. That is why we have chosen the pole region of relative damping 1.4 and maximum real part -0.1 as a condition. In Fig.13, the selected region is represented by the dashed line.

According to the previous section it does not have any sense to set the input coefficient greater than one. The resulting input coefficient range that satisfies the defined conditions is as follows:

- minimum: 0.512
- maximum: 1.000

At the end, the influence of the input coefficient on the active suspension performance has been tested.
The quantitative measures we have compared using passive suspension performance. The random road disturbance we have used as a first test input and the driving over a bump as a second input. The comparison for minimum and maximum input coefficients and their influence on the active suspension system performance is summarized in Table 1. The percentage values are computed as relative improvements of the active system compare to the passive suspension.

Table 1
Influence of input coefficient on performance

coefficient	0.512	1
H_∞ norm	0.455	0.359
comfort	20.13%	29.89%
stability	8.92%	12.83%
energy	-71.1J	127.6J

8 Results

Deterministic random signal described above has been used for experiments on the test bed. Two important things should be supervised during experiments – suspension comfort improvement and energy consumption.
At first the body (sprung) mass displacement is plotted as an indicator of the comfort improvement. There are three curves in Fig.14 – the input signal (road displacement) and body displacement for two levels of the energy demands.

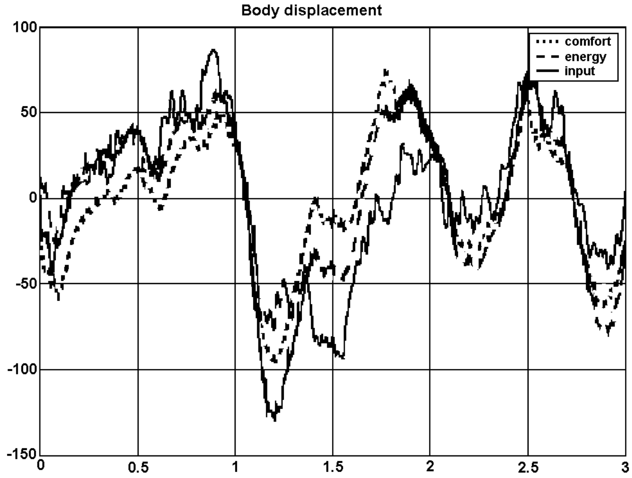

Fig.14 Sprung mass displacement

Fig.14 shows two curves - energy demand for standard energy consumption (say “comfort setting”) and energy demands for lower consumption (say “energy setting”). Negative values represent the recuperated energy.

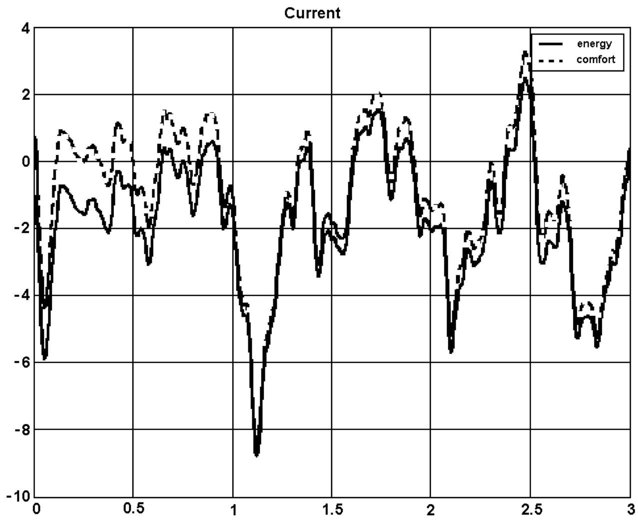

Fig.15 Energy demands (electric current)

Let’s show the results as mean values. Table 2 involves mean values for the road and body displacements as absolute values and as a percentage of the improvement.

Table 2
Displacement mean values

	Mean value	Percentag
Body displacement – comfort	38.6	100%
Body displacement – energy	47.4	123%

Table 3 involves mean values of energy. Lower value corresponds the lower energy demand.

Table 3
Power mean values

	Mean value	Percentage
Comfort setting	2.689	100%
Energy setting	1.598	59%

9 Conclusion

In this paper the H_∞ controller for active suspension with linear electric motor has been used for experiment on the test bed. The experiment signal for real road simulation has been developed and then it has been used for experiments. The method for the direct real-time energy control with respect to reduction of the energy consumption has been used. Experiments verified validation of the simulations and showed that it is possible to change energy demands according to the road situation and status of the energy storage in the car (battery or super-capacitor). The method can be extended to general plants with considerable energy demands, where the decreasing actuator signal in a given range can preserve the system stability. Thus this controller with linear motor as an actuator can be used in any suspension system.

Acknowledgement

This research has been supported by the MSMT project No. 1M6840770002 ”Josef Bozek’s Research Center of Combustion Engines and Automobiles II” and MSMT projects INGO No. LA296 and No. LA299.

References:

[1] K. Zhou and J. C. Doyle: *Essentials of Robust Control*, Prentice Hall, 1998

[2] Kruczek, A. and Stribrsky, A.: H_∞ Control of Automotive Active Suspension with Linear Motor, *In Proccedings of the 3rd IFAC Symposium on Mechatronic Systems*, Sydney, Australia, 2004

[3] Stribrsky, A., Hyniova, K., Honcu J. and Kruczek, A.: Energy Recuperation in Automotive Active Suspension Systems with Linear Electric Motor, I*n Proccedings of the 15th Mediterranean Conference on Control and Automation*, Athens, Greece, 2007

[4] Koulocheris, D., Vrazopoulos, H.,Dertimanis, V.: A Hybrid Evolution Strategy for vehicle suspension optimization, *WSEAS Transactions on Systems,* Issue 1, Vol.3, 2004

[5] Honcu, J., Hyniova, K., Stribrsky, A., Kruczek, A.: Linear Motors as a New Way in Vehicle Suspension System. *In Preprints of the 3rd WSEAS International Conference on Computing, Communication and Control Technologies,* Orlando, Florida, USA, 2005

[6] Libin, L., Quiang, L.:Vibration Analysis Based on Full Multi-Body Model for the Commercial Vehicle Suspension System, *In Proceedings of the 6th WSEAS International Conference on Signal Processing, Robotics and Automation,* Corfu Island, Greece, 2007

[7] Maxche, A.R.: Theoretical and Experimental Dynamic Response Analysis of a Road Vehicle Suspension System Using an Electromagnetic Damper, *In Preprints of the International Conference on Advances in Machine Design & Industry Automation*, Pune, India, 2007