The Virtual Prototype of a Mechatronic Suspension System with Active Force Control

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Abstract: - The paper presents the virtual prototype of an automotive suspension system with force generator actuator. The active suspensions use sensors to measure the accelerations of sprung mass and unsprung mass, the analog signals from the sensors being transmitted to the controller, which communicates with the force actuator. In this way, the passenger comfort and car stability are simultaneously improved. The study is developed for a quarter-car model, which contains the guiding & suspension system of a front wheel. The virtual prototype of the active suspension is a control loop composed by the multi-body mechanical model connected with the dynamic model of the actuator and with the controller model. The digital platform used in the study integrates MBS (Multi-Body Systems) and DFC (Design for Control) software solutions. The numeric simulations have been performed considering a step input signal, the control strategy aiming to decrease the overshoot, without adversely affecting other performance indexes.

Key-Words: - Suspension, Active force control, Mechatronic system, Virtual prototype.

1 Introduction

The suspension mechanisms are vehicle-specific systems that serve a dual purpose - contributing to the car's roadholding/handling and braking for good active safety and driving pleasure, and keeping vehicle occupants comfortable and reasonably well isolated from road noise, bumps, and vibrations. Generally, the suspension system is an assembly of linkages (the guiding mechanism), translational and torsion springs. dampers/shock absorbers. bumpers and rebound elements, which connect the wheels to car body.

The modern design process of the suspension systems involves conceptual and functional design, digital mockup, virtual prototyping and testing. The conceptual design has as main objective to establish the best product concept, by using data picked by the science, technology, economy or market, while the functional design involves identifying, modeling and evaluating the operational performances of the suspension systems, and the deviations from the imposed characteristics, with other words the mode in which the suspension mechanism responses to the design requirements (objectives and constraints).

Determining the real behavior has become a priority in the functional design of the automotive systems, including the suspension systems. Various scientific papers reveal a growing interest on analysis methods for multi-body systems (MBS) that allow the selfformulating algorithms, having in view to develop powerful virtual prototyping environments [1-6]. An important advantage of this kind of simulation consists in the possibility of make virtual measurements in any point or area and for any parameter. In this way, we can take decisions on any design changes without going through physical prototype building and testing.

In the field of automotive design & development, the virtual prototyping technique is used in different types of applications, such as: suspension design (predicting suspension characteristics, optimizing suspension design, loads analysis, packaging studies); vehicle dynamics (modeling tire-roadway interaction, simulating linear-range and emergency handling maneuvers, predicting vehicle stability, braking studies, predicting chassis behavior on acceleration, assessing durability); engine design (valve train dynamics, timing chain design and simulation, crankshaft loads prediction); power train engineering (transmissions, transfer cases, differentials, drivelines, linkage design, predicting shift linkage precision and driver effort, analyzing transmission gear rattle and shifting smoothness, predicting bearing loads); body hardware engineering (door, trunk, and hood latch design, windshield wiper simulation and refinement, seat mechanism design, and so on).

The steps to create a virtual model mirror the steps to create a physical prototype. During the build phase, virtual prototypes are created of both the new product concept and any target products which may already exist in the market. One of the most important axioms for successful functional virtual prototyping is to simulate as test. Testing of hardware prototypes has traditionally involved both lab tests and field tests in various configurations, which are very expensive. With virtual prototyping, it is enough to create virtual equivalents of the lab and field tests. To validate the virtual prototype, the physical and virtual models are tested identically, using the same testing and instrumentation procedures. Refining the virtual prototype involves the fidelity of the model. By replacing the rigid components with flexible counterparts, adding frictions, and representing the automatic systems that control the operating performance of the mechanical system, respectively, can make the improvement of the virtual prototype.

Generally, a virtual prototyping platform (fig. 1) includes the following software solutions [7]: CAD -Computer Aided Design (ex. CATIA, PROENGINEER, SOLIDWORKS); MBS - MultiBody Systems (ex. ADAMS, SD-EXACT, PLEXUS); FEA - Finite Element (ex. NASTRAN/PATRAN, Analysis COSMOS, ANSYS); DFC - Design for Control (ex. MATLAB, EASY5, MATRIXx). The MBS software is the main component of the virtual prototyping platform, and it allows analyzing, optimizing, and simulating the system under real operating conditions. The CAD software is used for creating the geometric (solid) model of the system. This model contains data about the mass & inertia properties of the rigid parts. The part geometry can be exported from CAD to MBS using standard format files, such as STEP or Parasolid.



The FEA software is used for modeling flexible bodies in mechanical systems. Integrating flexibilities into model allows to capture inertial and compliance effects during simulations, to study deformations of the flexible components, and to predict loads with greater accuracy, therefore achieving more realistic results. The flexible body characteristics are defined in a finite element modeling output file (MNF - Modal Neutral File). The information in a MNF includes location of nodes and node connectivity, nodal mass and inertia, mode shapes, generalized mass and stiffness for modal shapes. The MBS model transmits to FEA the motion & load states in the mechanical system, which can be defined using a FEA Loads format file.

The modeling & analysis of the suspension systems in MBS concept involves the following steps: defining the suspension mechanism as multi-body system (modeling the kinematic elements / parts as bodies or composite restrictions); establishing the coordinate frames of the bodies (global frame - for the ground, and local frames - one for each mobile part, respectively); defining the geometric model of the suspension mechanism, for each body in the local coordinate system; defining the connection matrices between the local coordinate frames and the global coordinate frame; establishing the global coordinates of the characteristic points that define the structural model of the suspension mechanism; defining the equation system that describe the geometric and kinematic constraints from the suspension mechanism; formulating the motion's differential equations using different formalisms (ex. Newton - Euler, Lagrange); defining the mass and inertia characteristics of the bodies; establishing the reaction forces and torques from the suspension mechanism; solving the algebraic and differential equation system.

The paper approaches the design and simulation of an active suspension system, by using the modeling & analysis capabilities of the virtual prototyping technique. Generally, an active suspension controls the vertical movement of the wheels via an onboard system rather than the movement being determined entirely by the surface on which the car is driving. The system therefore virtually eliminates body roll and pitch variation in many driving situations including cornering, accelerating, and braking. Our study aims a tradeoff between passenger comfort, i.e. minimizing car body travel, versus suspension travel as the performance objective. The idea is to transform the conventional passive suspension into an active one, using a force generator actuator, controlled by feedback, between the chassis and wheel assembly.

The suspension system is approached in mechatronic concept, integrating the mechanical structure and the electronic control system at the virtual prototype level. In fact, the virtual prototype is a control loop composed by the multi-body mechanical model connected with the dynamic model of the actuator and with the controller model. The mechanical model of the suspension mechanism was developed by using the MBS environment ADAMS, while for the control system design we used the DFC software solution EASY5.

The main contribution of our work, comparative with other related papers that discuss the same issue, is related to design & control philosophy, considering the road disturbance as a perturbation to be eliminated. No less important is the tuning algorithm for the PID controller design, based on parametric technique, which is a collection of procedures and statistical tools for planning experiments and analyzing the results.

2 Suspension Model

A classic (passive) suspension system consists of an energy dissipating element, which is the damper, and an energy-storing element, which is the spring. Since these two elements cannot add energy to the system this kind of suspension systems are called passive; the advantages are simplicity and costs. If there a force actuator is placed in parallel to passive system, an active suspension system is obtained. The active suspensions use sensors to measure the accelerations of sprung mass and unsprung mass, the analog signals from the sensors being transmitted to the controller, which communicates with the force actuator. In this way, the passenger comfort and car stability can be simultaneously improved [8-13]. However, the applications of these advanced suspensions are constrained by the cost, packaging, weight, and reliability.



Fig. 2

In this paper, the study is developed for a quarter-car model, which contains the guiding & suspension system of the left front wheel. The virtual model of the passive suspension system, which has been designed by using the MBS environment ADAMS of MSC, is shown in figure 2. A four-bar mechanism is used for the wheel suspension. The suspension mechanism uses two control arms to hold the wheel carrier and control its movements. The lower and upper wishbones connect to the car body using compliant joints (i.e. bushings). Spherical joints constrain the upright parts to the upper and lower control arms. Tie rod attaches to the steering mount part and to the wheel carrier through spherical joints. Revolute joints connect the wheel carriers to the tire mount part. The upper and lower struts of the damper are connected through cylindrical joints, and to the adjacent parts through spherical joints. The spring is disposed between the lower strut of the damper and car body.

The active suspension system is obtained from the conventional passive suspension by using a force generator actuator, controlled by feedback, which is mounted in parallel with the passive spring & damper group (fig. 3). For the quarter-car model, the car body equilibrium is assured with a translational joint to ground (i.e. fixed part), in the median plane of the vehicle, along the vertical axis. The simplified model of the quarter-car passive suspension system is shown in figure 4.a, while the equivalent active suspension system is one in figure 4.b; these models consider only two mobile parts (bodies), namely the sprung and unsprung masses.







Fig. 4

The sprung mass m_2 represents the quarter car body, while the unsprung mass m_1 represents the wheel assembly (including the guiding mechanism). The spring k_2 and damper c_2 represent a passive spring and shock absorber that are placed between the car body and the wheel assembly, while the spring & damper group k_1 - c_1 serves to model the tire. The variables z_2 , z_1 , and z are the car body travel, the wheel travel, and the road disturbance, respectively. The force actuator represents the active component of the suspension system, being disposed between the sprung and unsprung masses.

According with the Newton-Euler formalism, the differential dynamic equations for the active suspension system can be written in the following form:

$$m_{2}\ddot{z}_{2} = -c_{2}(\dot{z}_{2} - \dot{z}_{1}) - k_{2}(z_{2} - z_{1}) + u,$$

$$m_{1}\ddot{z}_{1} = c_{2}(\dot{z}_{2} - \dot{z}_{1}) + k_{2}(z_{2} - z_{1}) - u + c_{1}(\dot{z} - \dot{z}_{1}) + k_{1}(z - z_{1}),$$
(1)

where u is the control signal (i.e. the force generated by the actuating system). The equations can be rewritten using the Laplace transform,

$$(m_2 s^2 + c_2 s + k_2) Z_2(s) - (c_2 s + k_2) Z_1(s) = U(s), [m_2 s^2 + (c_1 + c_2) s + (k_1 + k_2)] Z_1(s) - (c_2 s + k_2) Z_2(s) = (2) = (c_1 s + k_1) Z(s) - U(s).$$

The passive suspension system can be modeled with the same equations, considering u=0. The inputs in the dynamic model are the road disturbance (z) and the control signal (u), while the outputs are represented by the car body travel (z_2) and the wheel travel (z_1). For control system synthesis, we have considered that the performance index of the system refers to the difference between output travels (z_2 - z_1). The system is a linear one, being possible to apply the overlapping effects principle: the output z_2 - z_1 is the combined effect of the input signals z and u. In these terms, from equations (2) we have obtained the transfer functions,

$$G_U(s) = \frac{Z_2 - Z_1}{U}, G_Z(s) = \frac{Z_2 - Z_1}{Z}.$$
(3)

where $G_Z(s)$ is the transfer function of the passive suspension system, while $G_U(s)$ defines the ratio of the output to the input for the force actuator. Considering the equations (2) and (3), we obtained:

$$G_{U}(s) = \frac{(m_{1} + m_{2})s^{2} + c_{1}s + k_{1}}{(m_{2}s^{2} + c_{2}s + k_{2}) \cdot [m_{1}s^{2} + (c_{1} + c_{2})s + (k_{1} + k_{2})] - (c_{2}s + k_{2})^{2}},$$

$$(4)$$

$$G_{Z}(s) = \frac{-m_{2}c_{1}s^{3} - m_{2}k_{1}s^{2}}{(m_{2}s^{2} + c_{2}s + k_{2}) \cdot [m_{1}s^{2} + (c_{1} + c_{2})s + (k_{1} + k_{2})] - (c_{2}s + k_{2})^{2}}.$$

The numeric values of the parameters correpond to a domestic motor vehicle, as follows: $m_1 = 31 \text{ kg}$, $m_2 = 415 \text{ kg}$, $k_1 = 225000 \text{ N/m}$, $k_2 = 51000 \text{ N/m}$, $c_1 = 3500 \text{ Ns/m}$, $c_2 = 3200 \text{ Ns/m}$. In this way, there are the following transfer functions:

$$G_U(s) = \frac{0.03467 \, s^2 + 0.02721 \, s + 17.49}{s^4 + 223.8s^3 + 11490 \, s^2 + 120600 \, s + 1296000},$$

$$G_Z(s) = \frac{-112.9s^3 - 7258 \, s^2}{s^4 + 223.8s^3 + 11490 \, s^2 + 120600 \, s + 1296000}.$$
(5)

For the current stage of our research, the active suspension was designed considering the road disturbance as a perturbation to be eliminated by the control system (which generates the signal u). Accordingly, the basic scheme of the controlled suspension system is shown in figure 5, the detailing being done in the next chapter of the paper.

3 Control System Design

The control block diagram of the active suspension system (fig. 6) was developed by considering the basic scheme shown in figure 5. The control system model is designed in the concurrent engineering concept by using the DFC (Design for Control) software solution EASY5 of MSC Software. This is an engineering software program used to model, design, and simulate dynamic systems, which cover a broad range of engineering systems, including, mechanical, electrical, hydraulic, pneumatic, thermal, gas dynamics, powertrain, vehicle dynamics, digital/analog control systems and much more. Models are assembled from simple modeling blocks, such as summers, dividers, lead-lag, filters, integrators, advanced application-specific and components provided from specialized libraries.

The input and output plants of the mechatronic system have been modeled in ADAMS/Controls (which is a plugin to ADAMS/View) for connecting the mechanical model and the actuating & control system model. The communication data are saved in a specific file for EASY5 (*.inf); ADAMS/Controls also generates a command file (*.cmd) and a dataset file (*.adm) that are used during simulation. In the control diagram, the

"ADAMS Mechanism" block represents the MBS model of the suspension system, and it was created based on the information from the "inf" file.

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MSC ADAMS block provides an interface link to the mechanical model of the suspension system. The combined model is executed in the co-simulation mode (the EASY5 solver integrates the EASY5 equations, while the ADAMS solver integrate the ADAMS equations). The two solvers exchange input and output data at a rate determined by the communication interval parameter. The state of the co-simulation mode is the vector of independent states of the ADAMS model. The physical meaning of these states depends on how your ADAMS model was constructed and may change during the course of a simulation. The user-defined inputs and outputs are the ADAMS variables specified in the Plant Output and Input elements in the ADAMS model.

There are the following input parameters for the cosimulation mode: communication interval - the time between exchanges of data between the EASY5 and ADAMS solvers; output interval - determines how often animation data is sent to the ADAMS window; EASY5 interpolation order - for linear interpolation to be used by the EASY5 solver on the values of the inputs from the ADAMS model between communication times, or to maintain ADAMS inputs at constant value between communication times; ADAMS interpolation order - to use linear extrapolation on inputs to the ADAMS model between communication times, or to use constant previous value of the inputs between communication times; ADAMS solver - to use the Fortran solver, or the C++ based solver; communication mode - to use the default pipes based communication method, or the TCP/IP communication.

Unlike passive suspension, which is an open loop, the active suspension is a closed loop system. Sensors continually monitor body movement and vehicle ride level, the analog signals being transmitted to a controller. Regarding the controller, different solutions are used / presented in literature, such as classical PID (Proportional Integral Derivative) controllers, or more advanced FLC (Fuzzy Logic Controllers) techniques [14-20]. In the synthesis of a control system, the main problem is not only the selection of the controller, but also the tuning of the specific parameters, to verify certain given specifications for the controlled process. From this point of view, because three gains will tune a PID system, the design process involved is easy to describe, while the design of a fuzzy rule-based system is more complex, involving input selection, membership function definition, and rule definition.

The PID controller is a generic control loop feedback mechanism, which attempts to correct the error between a measured process variable and a desired set-point by calculating and then outputting a corrective action that can adjust the process accordingly. The proportional gain determines the reaction to the current error, the integral value determines the reaction based on the sum of recent errors, and the derivative value determines the reaction based on the rate at which the error has been changing. The transfer function of the PID controller is given by the next equation:

$$v(t) = K_{p} e(t) + K_{i} \int_{0}^{t} e(\tau) d\tau + K_{d} \frac{d e(t)}{d t}, \qquad (6)$$

which can be written using the Laplace transform,

$$V(s) = \left[K_p + \frac{K_i}{s} + K_d s\right] E(s),$$
(7)

where K_p is the proportional term, K_i - the integral term, K_d - the derivative term, e - the position error, τ - the generalized force, V - the Laplace transform for the command signal, E - the Laplace transform for the position error. The output signal of the controller (v) will be sent to the plant, and the new output will be obtained. This new output will be sent back to the sensor again to find the new error signal (e); the controller takes this new error signal and computes its derivative and its integral again.

There are several methods for tuning a PID controller. The most effective methods generally involve the development of some form of process model, and

then choosing P, I, and D based on the dynamic model parameters. To obtain the desired response, there is the following sequence: add a proportional control to improve the rise time, add a derivative control to improve the overshoot, add an integral control to eliminate the steady-state error. In our research, the tuning of the PID controller is made by using the optimization capabilities of the virtual prototyping technique. For performing this study, we used ADAMS/Insight, part of the MSC ADAMS suite of which allows designing sophisticated software. experiments for measuring the performance of the mechanical & mechatronical systems. It also provides a collection of statistical tools for analyzing the results of the experiments so that we can better understand how to refine and improve the system.

Design of experiments (DOE), also called experimental design, is a collection of procedures and statistical tools for planning experiments and analyzing the results, which can be performed for identifying the varying several design variables effects of simultaneously, having as goal to identify which variables and combinations of variables most affect the behavior of the mechanical/mechatronical system. In this paper, the DOE technique was applied to determine the optimal values of the tuning parameters of the PID controller (the proportional term, the integral term, and the derivative term), in order to assure the imposed performance indexes of the active suspension system.

Experimental design has been performed in five stages: modeling the purpose of the experiment; choosing the set of factors for the active suspension that we are investigating (i.e. the tuning parameters of the PID controller); determining the values for each factor, and planning a set of trials in which we vary the factor values from one trial to another; executing the runs and recording the performance of the system at each run; analyzing the changes in performance across the runs. The runs are described by the design matrix, which has a column for each factor and a row for each run. The matrix entries are the levels for each factor per run.

The control system of the active suspension is one in figure 6, for which the control block model of the PID controller is shown in figure 7, where sum is the output from the SJ (Summing Junction) block in figure 6. The derivative state for input to the PID block is consistent with the proportional state. The PID block automatically creates the integrated state of the proportional input for use as the integrated input. In these terms, we used an implicit differential equation (DIFF) to get the time derivative. follows: sum derivative as \rightarrow DIFF 1 DIF1(.suspension.DIFF 1), where = [DIF(.suspension.DIFF 1) - sum], suspension being the model name. Afterwards, the suspension model was exported in ADAMS/Insight for creating the experiment.

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Fig. 8

The first step required to creating the designed experiment is to select the factors to include in the design matrix (the proportional, integral, and derivative terms). We select factors from the Candidates list, and then promote them to the Inclusions list (fig. 8). Promoting candidates to inclusions causes them to become part of the design matrix. After promoting the design factors, we have defined parameters for them in the factor form, as follows: the nominal value, and the value range. The nominal value of the factor is also known as the center point, the model inputs varying relative to this value. The value range (field) of the factor can be defined using the absolute minimum and maximum values, or relative / relative percent to the nominal value.

After promoting and modifying the factors, there is necessary to promote the responses for the experiment. The specific parameters in the response form refer to the desired target, or the approximate lower and upper limits, as well as the operation in the optimization form (e.g. minimize, maximize, less/greater then or equal). In our case, the responses refer to the settling time, overshoot and steady-state error (see figure 8).

The next step is to set the design objective and design type for the experiment, mainly the investigation strategy and the DOE design type. For the suspension system in study, we used DOE Screening strategy of full-factorial type. This method identifies the factors and combinations of factors that most affect the behavior of the system, picking only high and low values of the setting range. The full factorial algorithm is the most comprehensive of the design types and uses all of the possible combinations of levels for the factors. The total number of runs is m^n , where m is the number of levels and n is the number of factors (in our case, n=3 – the tuning factors of the PID controller).

After configuring the investigation strategy, we created the design space and the work space. The design space is a matrix with the rows representing the run and the columns representing the factor settings; the settings are in a normalized representation. The work space is a matrix with the rows indicating the runs and the columns identifying the factor settings and resulting responses' values. ADAMS/View will run a simulation for each trial defined in this matrix; finally, the simulation results will appear in the design matrix.

In the final tuning stage, the effective optimization of the active suspension system / PID controller has been performed by updating design objectives (responses) settings. Because there are three responses, a multiobjective optimization was performed, using the target and weight values for adjusting the relative importance of the responses. Based on these, ADAMS/Insight computes a cost for each response and combines these into an overall cost, and then minimizes the overall cost. The response cost is the difference between the response value and the target value multiplied by the weight. The target value acts as the desired value, and the weight scales the response relative to other responses. During the optimization (tuning) process, ADAMS/Insight automatically adjusts the factor values so that the resulting responses come as closely as possible to the specified target values.

The method used to solve the optimization problem is OptDes GRG (Generalized Reduced Gradient), which is a conventional gradient-based optimizer. The solver settings refer to the relative tolerance for convergence, the maximum number of steps to perform, the relative amount to perturb variables during differencing, and the method for computing derivatives using finite differences. As differencing method we have selected Forward, which perturb above the nominal value only, and use the slope as the derivative.

Finally, the optimal values of the factors will result in a simulation that meets the design requirements, as follows: $K_p=K_d=2e5$, $K_i=0$; therefore, to adjust the suspension there is necessary, and sufficiently, a PD controller. These values, which assure the minimum cost for the objectives, are then transferred to the controller model for performing the co-simulation of the mechatronic suspension system in ADAMS and EASY5; the two solvers exchange input and output data at the rate determined by the communication interval.

4 Simulation Results

The numeric simulations have been performed considering a step input signal, with amplitude of 0.1 meters, like the wheel would climb over a bump. For a comfortable suspension system, the settling time should be less than 2 seconds, while the maximum acceptable value for the overshoot is 5% [21]. Referring to the passive suspension, there are good values for the settling time and the steady-state error, but the overshoot is too high - $\sigma \approx 31\%$ (fig. 9). The vertical displacements of the sprung (z_2) and unsprung (z_1) masses are shown in figure 10. From these results we can extract the following: the car body oscillates (transitory regime) on a period less than 1 second; the maximum elongation of the oscillations (which are measured relative to the wheel axis) is around 3 cm.

Fig. 9

Fig. 10

In these conditions, the control system of the active suspension aims to decrease the overshoot, without adversely affecting other performance indexes. Following the control strategy for the active suspension, the overshoot of the output signal become very small $(\sigma \approx 1.5\%)$, while the stabilization time is around 1 second (fig. 11). Therefore, the results are in the recommended field for a comfortable suspension, and this demonstrates the viability of the control strategy. For generating this behavior, there are necessary the displacement & force signals of the actuator shown in figure 12.

The final aspect was to verify the robustness of the control system (i.e. the capability to operate with the imposed indexes, or closed-by these values, when one or more parameters of the physical model are changing). In our research, the mass of the car body (more precisely, the quarter-car body mass, m_2) is the variable parameter for verifying the robustness performance of the system (the vehicle can carry different payloads, which modify the mass/weight of the car body). The study was performed considering the variation field of the sprung mass $m_2 \in [415, 800]$ kg; the control system was initially designed for $m_2 = 415$ kg, the robustness being verified for $m_2 = 800$ kg. In these terms, considering the equations 4, there are obtained the new transfer function of the passive suspension $G_{U}(s)$, and the ratio of the output to the input for the force actuator $G_Z(s)$, as follows:

Fig. 12

$$G_U(s) = \frac{0.03351 s^2 + 0.1411 s + 9.073}{s^4 + 220.1s^3 + 10240 s^2 + 62550 s + 672500},$$

$$G_Z(s) = \frac{-112.9s^3 - 7258 s^2}{s^4 + 220.1s^3 + 10240 s^2 + 62550 s + 672500}.$$
(8)

The verification of the robustness is made considering the control system model shown in figure 6. The results obtained for the new loading conditions are shown in figure 13, corresponding to the passive suspension, while the behavior for the active suspension system is presented in figure 14. The control strategy leads to a comfortable suspension system even for the limit value of the sprung mass, $m_2 = 800$ kg. This result demonstrates the robustness of the control system.

Fig. 14

5 Conclusions & Future Researches

The application is a relevant example regarding the implementation of the virtual prototyping instruments in the design process of the active suspension systems. One of the most important advantages of this kind of simulation is the possibility to perform virtual measurements in any point or area of the suspension system, and for any parameter (e.g. motion, force). Virtual prototyping-based simulation tools allow realizing the projected reductions in cycle times while maintaining and increasing the performance, safety, and reliability. This helps us to take quick decisions on any design changes without going through expensive hardware prototype building and testing.

The future researches in the field will be focused on the verification of the controller behavior through frequency analysis methods. We intend to determine the resonance magnitude and the corresponding frequency, for evaluating the filter character of the system. Generally, the road profile can be modeled as irregular system, which can be decomposed into sinusoidal signals with amplitudes of increasingly smaller as the frequencies are becoming higher. Dangerous are those signals for which the dominant components (with high values) have the frequency equal with the resonance frequencies. At the same time, we intend to extend the research for more complex suspension models, such as half-car and full-car models, and for other control strategies & controller types (including fuzzy logic & fuzzy neutral logic techniques). We also intend to continue the research by modeling the mechanical structure with finite elements, for identifying the eigenshapes and eigenfrequencies of the system, which are useful to avoid the resonance phenomenon due to the action of the external dynamic loads.

The active suspension system is in manufacturing stage, and it will be tested by using an experimental stand (fig. 15), creating a real perspective for the research in the field. The testing installation is based on hydraulic linear actuators (MTS HTC - Hydraulic Testing Components), including the control digital system Flex Test GT Controller and the applications Basic Test Ware & Multi-Purpose Test Ware. This will allow a relevant comparison between the virtual prototype analysis and the data achieved bv measurements, in the validation process of the virtual prototype. The results of the experimental study will be presented in a future paper.

Fig. 15

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