Dynamic Optimization of the Controller for the Active Suspension System of a Race Car

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Abstract. This work deals with the dynamic optimization of the control system for the active suspension system of a single-seater race car. The suspension mechanism in study is an innovative one, in its design starting from the requirement to eliminate the contradictory variations of some movement parameters in the case of the classic suspension with four-bar mechanism. The suspension system is approached in a mechatronic concept, by using a virtual prototyping platform that integrates MBS (Multi-Body Systems) and DFC (Design for Control) software solutions. The optimization aims at determining the tuning parameters of the controller so that to minimize the wheel track variation.

Keywords: Race Car · Suspension · Control System · Optimization · Dynamics

1 Introduction

The suspension role is to protect the car against shock, vibration and harmful oscillations caused by road bumps. Current vehicles use the following types of suspensions: passive suspensions, which are made of elastic and dissipative elements such as springs and dampers (the dynamic behavior is given by the characteristics of these elements, and it cannot be changed during operation) [1–3]; semi-active suspensions, which are composed of elastic elements and controlled dampers (the dynamic behavior is changed by adjusting the viscosity coefficient, without introducing external forces into the system) [4–9]; active suspensions, which are composed of elastic and dissipative elements along with which there are actuation systems (the dynamic behavior is changed by the external forces introduced by the actuators) [10–16].

This paper deals with the optimal design of the control system for the active suspension system of a race car (Formula Student). More specifically, it is about transforming a traditional passive suspension into an active one, so as to solve one of the well-known problems with a commonly used passive suspension, namely the contradictory variations of the wheel track and camber angle [2, 3]. The study is carried out through the use of a virtual prototyping platform, by integrating the MBS (Multi-Body Systems) mechanical model (developed in ADAMS) and the DFC (Design for Control) actuation system (conceived in EASY5). Important advantages are obtained by such an approach in mechatronic concept, as stated in [17–19].

I. Dumitru et al. (Eds.): ICOME 2022, AHE 15, pp. 18–26, 2023.
https://doi.org/10.2991/978-94-6463-152-4_3

2 Active Suspension System Setup

The traditional suspension system (for both front and rear wheels) used for race cars is the one with a four-bar mechanism (Fig. 1). This mechanism has also been commonly used in passenger cars, but has been replaced by the McPherson strut suspension solution. At the four-bar mechanism used for the suspension of passenger cars, the spring & damper assembly is arranged in a vertical plane (Fig. 1,a), usually between the upper control arm of the mechanism and the car body, which is not possible for race cars due to the limited available space. Under these conditions, there is used the solution with the arrangement of the spring & damper assembly in a horizontal plane, the forces being transmitted through a push-rocker group (Fig. 1,b).

From a structural (and also kinematic) point of view, the two variants of the suspension system based on four-bar mechanism each have one degree of mobility, which corresponds to the vertical travel of the wheel (Y_K). Although it is a simple constructive solution, the four-bar suspension mechanism has the disadvantage of contradictory variations of the wheel track and camber angle (the decrease of one of these variations leads to the increase of the other, this resulting in a non-linear dynamic behavior).

The decoupling of the two contradictory variations can be achieved by using a suspension mechanism with degrees of mobility, either in passive or active suspension. To design such a suspension, it was started with a five-bar mechanism, as shown in Fig. 2, which implies a supplementary degree of mobility (for example at the level of the movement of the upper arm) by reference to the four-bar suspension.

The control of the second degree of mobility in the basic five-bar mechanism can be done both mechanically (so passive suspension) and electronically (active suspension).

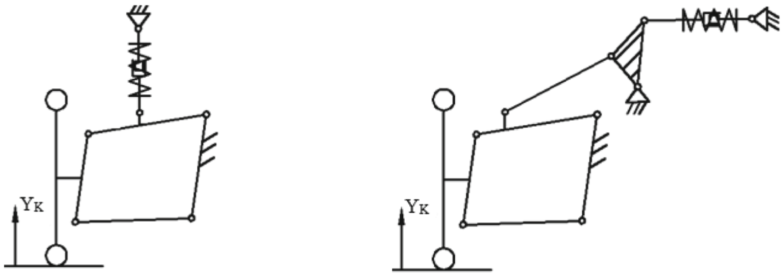

Fig. 1. The four-bar suspension mechanism: setups for passenger (a) and race (b) cars.

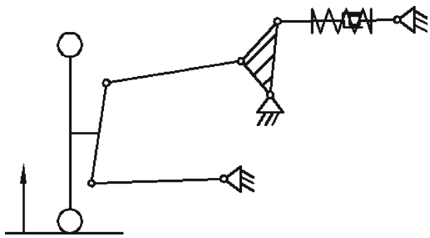

Fig. 2. The bi-mobile suspension system based on a five-bar mechanism.

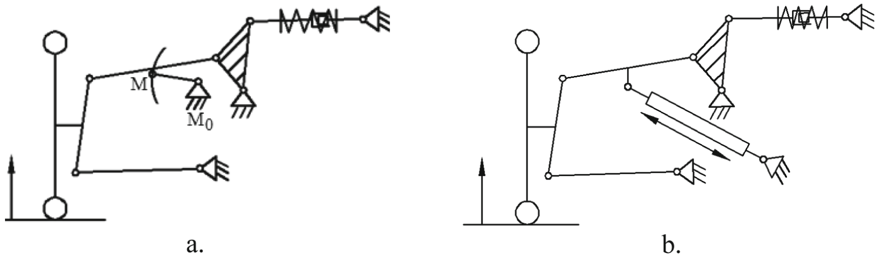

Fig. 3. The mono-mobile suspension systems derived from the five-bar mechanism: passive (a) and active (b).

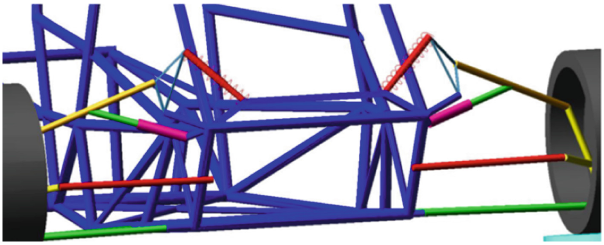

Fig. 4. The MBS model of the front axle suspension system (half-car model).

In the pure mechanical suspension (Fig. 3,a), the movement of the upper suspension arm is realized through a rocker mounted between the upper rod (in M) and the car body (in M_0). The arrangement of this rocker is made so that the trajectory of the M point will ensure the cancellation (or at least the minimization) of the wheel track or camber angle variation, as the case may be. This solution was discussed in detail in a previous work of the authors [20].

In the active suspension system, the control of the second degree of mobility can be realized by adding an actuating element (actuator) that pulls/pushes the upper rod of the five-bar mechanism (Fig. 3,b), thus cancelling, as the case may be, the variation of the wheel track or of the camber angle. The MBS dynamic model of the suspension system, which is shown in Fig. 4 (corresponding to the front axle of the race car), was developed by using ADAMS/View, the general preprocessing/modeling module in ADAMS software package. In the following, the study focuses on the optimal design of the control system, in terms of controller synthesis (tuning).

3 Control System Design

In order to design the control system for the active suspension of the race car, the virtual prototyping platform used in the work integrates a DFC software solution (namely, EASY5), which exchanges information (export & import) with the MBS software (ADAMS), meaning that the data results from the MBS model is an entry into the DFC model and vice versa. The communication between the MBS and DFC models is

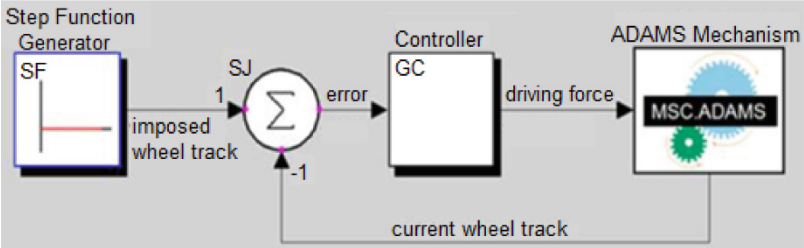

Fig. 5. Single-loop control scheme of the active suspension.

managed by using ADAMS/Controls, which is a plug-in for ADAMS/View. The simulation algorithm involves, in addition to designing the MBS model of the suspension mechanical device, the following steps:

- modeling the input and output plants (the outputs describe the variables transmitted to the DFC application, while the inputs describe the variables returned in MBS);
- transfer and configure the MBS interface block into the DFC model;
- designing the control system block diagram;
- synthesis of the control element (controller);
- co-simulation of the mechatronic system.

For the control system, a series of layouts can be designed, with one or more loops (corresponding to the number of monitored/controlled parameters). In the single-loop control schemes, the position of the system is controlled, while in the two-loop schemes, in addition to position, a speed parameter usually occurs. In a more general case, three parameters can be controlled (position, speed and current). Obviously, single-loop schemes are the simplest, while multi-loop schemes ensure superior system behavior (stability, robustness) but at a higher complexity and cost.

For the present work, a single-loop control scheme was chosen, the controlled parameter being the wheel track variation, which must be cancelled/minimized. In these terms, the general control scheme designed in EASY5 (corresponding to one of the actuating elements, that is, the suspension of one of the wheels) is shown in Fig. 5, the blocks involved in this scheme having the following meanings [21]:

- RF – ramp function generator, used to model the input signal (imposed wheel track variation, in this case to be null);
- SJ – summing junction block, used to compare the imposed measure (branch “1”) with the measured/current (branch “-1”);
- GC – block used to model the controller;
- ADAMS Mechanism – ADAMS interface block, which integrates the MBS model of the suspension system (i.e. the mechanical device conceived in ADAMS/View).

By the SJ block, the imposed wheel track variation is compared with the variation achieved by the active suspension system, the output from this block being, in fact, the error that must be minimized by the control system. This is an input to the controller,

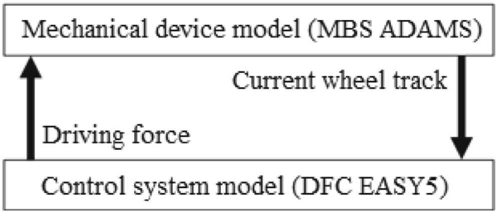

Fig. 6. Communication scheme in the mechatronic system.

which generates the driving force for the MBS mechanical model developed in ADAMS. In order to ensure the communication between the mechanical and control models, the input & output variables were defined, and respectively the functions by which these variables are called, as shown in Fig. 6.

For the input variable, representing the driving force developed by the linear actuator, the time function has a null value, and this because the variable is to receive its value from the control application. Subsequently, this variable was assigned as a function for the force applied to the actuator piston, using the predefined function VARVAL - Variable Value, by which the value of the state variable is returned. For the output state variable, the time function returns the current value of the wheel track, for whose modeling the predefined function DX (Distance Along X) was used (X is the transverse axis of the car).

Based on these state variables, the input (PINPUT - Plant Input) and output (POUTPUT - Plant Output) plants of the controlled process were then defined. The next step was to generate the files for the control application (EASY5), using the ADAMS/Controls module. The information about the input and output plants are saved in a file with the extension "inf" (specific for EASY5); at the same time, a command file "cmd" (for ADAMS/View) and a data file "adm" (for ADAMS/Solver) are generated, which will be used during the co-simulation [22]. The configuration of the ADAMS interface block in EASY5 involves selecting the "inf" file generated by the ADAMS/Controls export, and the execution mode (in this case, co-simulation).

From the point of view of the control element, several variants in the PID (Proportional-Integral-Derivative) family were tested, in order to identify the simplest controller that ensures a proper behavior of the suspension system. Besides the general PID controller, the following derived/simplified variants were also tested: PI (Proportional-Integral), PD (Proportional-Derivative), and P (Proportional).

The diagram of the general PID controller in EASY5 is shown in Fig. 7, the intervening parameters having the following meaning (the notations from EASY5 were used): REF_GC - controller input (output from the summing junction block SJ, i.e. the error); S_Feedback - feedback signal; GKP - amplification factor (proportional); GKF - amplification factor on the feedback line; GKI - integral factor (integration time); TC1 - derivative factor (derivation time - used to calculate the approximate derivative of the error signal); TC2 - time constant for damping the feedback (used to prevent an implicit contour); S_Out_GC - controller output (i.e. the driving force transmitted to the MBS model); s - Laplace transform [21].

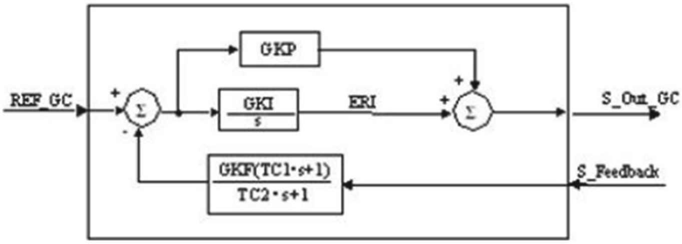

Fig. 7. Schematic of the PID controller in EASY5 [21].

In the following, the problem of tuning the PID controller will be discussed, the presented algorithm being then adapted for the particular situations of the derived controllers (PI, PD and P). The purpose of tuning the controller is to determine the optimum values of the specific factors involved in the transfer function equations, that is, as the case may be, the proportional factor (GKP), the integral factor (GKI) and the derivation time (TC1), so as to obtain the imposed performance indices.

4 Results and Conclusions

The tuning of the PID controller can be achieved by different methods, which include the root location method, frequency methods and others [1]. In the present work, the tuning of the controller is regarded as an optimal design process, similar to that used to optimize the mechanical device of the passive suspension system [20], which will be conducted in ADAMS. There are the following specific data for the optimization process: the design variables - the controller's tuning factors (P-I-D, P-I, P-D, or P, as the case may be); the design objective - the positioning error, as a difference between the imposed value of the wheel track and the current/measured one; the monitored value of the design objective - the root mean square (RMS) during simulation; the optimization goal - to minimize the monitored value of the design objective. Therefore, the optimization problem is a mono-objective one, without design constraints.

In order to have access to the parametric optimization procedure included in ADAMS, the control system model was transferred from EASY5 to ADAMS. For this, the model is exported from the EASY5 interface through the External System Library (ESL) format, specifying also the system parameters that will later be identified in ADAMS as design variables (in this case, GKP, GKI and TC1). Once imported into ADAMS, in the form of a general state equation, the parameterized model of the control system, coupled with the MBS model of the suspension system, becomes available for optimization. For each design variable, there is defined an initial value as well as a range of variation (set by minimum and maximum boundaries).

The suspension system was tested in passing over bumps regime, the vertical travel of the wheel being controlled by an $Y_K = f(t)$ motion restriction, simulating the wheel passing over a 50mm (±25 mm) obstacle/bump, which was transposed in the form of a sinusoidal function, $Y_K = 25\cdot\sin(\text{time})$. The simulation was performed over a period of 2π ($\cong$6.28) seconds.

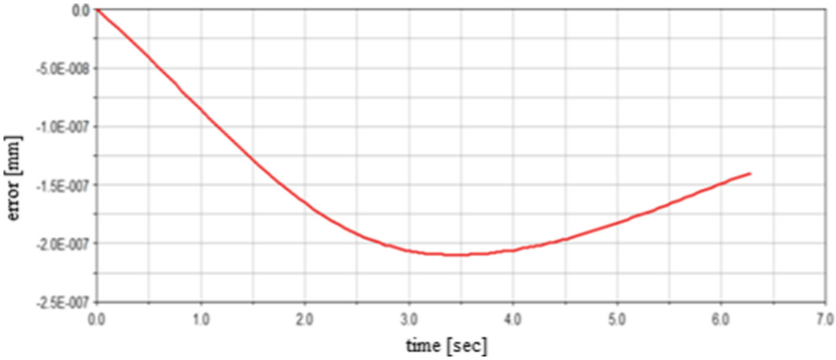

Fig. 8. The error obtained from the optimization of the PID controller.

The effective optimization was carried out by using OptDes-GRG, an algorithm provided with ADAMS/View [23]. In this way, the optimal values of the design variables (i.e. the tuning parameters of the controller) resulting from the optimization process were obtained, as follows: P (GKP) = 1e + 09, I (GKI) = 1000, D (TC1) = 9.6887e + 05. With these values, the time history variation of the error (the difference between the imposed and measured values of the wheel track) is shown in Fig. 8, the root mean square during simulation being practically insignificant (RMS = 1.6435e-007), which proves the viability of the adopted optimization procedure.

In a similar way, the optimization of the simplified types of controllers (derived from PID) was performed, with the following results: PI controller: P = 1e + 09, I = 3.9504e + 05, RMS = 2.6552e-007; PD controller: P = 1e + 09, D = 1e + 06, RMS = 2.4282e-007; P controller: P = 1e + 09, RMS = 8.0484e-007. Based on the obtained results, all the types of investigated controller ensure a proper behavior of the active suspension system. Under these conditions, the simplest (so cheap) variant, namely the proportional controller (P), is considered optimal.

The two variants of suspension systems derived from the 5-bar mechanism (shown in Fig. 3) are subject to a recently granted patent [24]. It should be noted that the passive suspension variant (Fig. 3,a) has already been developed and implemented on the university race car (Formula Student), the experimental tests to which it was subjected (both static and dynamic) proving its good performance [20].

References

1. Alexandru, C., Alexandru, P.: A comparative analysis between the vehicles' passive and active suspensions. International Journal of Mechanics 5(4), 371–378 (2011).
2. Țoțu, V., Alexandru, C.: Multi-criteria kinematic optimization of a front multi-link suspension mechanism using DOE screening and regression model. Applied Mechanics and Materials 332, 351–356 (2013).
3. Țoțu, V., Alexandru, C.: Optimal design of the front suspension mechanism used for a race car. Mechanisms and Machine Science 18, 243–253 (2014).
4. Alexandru, C.: A study on the semi-active suspension systems used for motor vehicles. Journal of Research and Innovation for Sustainable Society 2(1) 16–25 (2020).
5. Anaya-Martinez, M., Lozoya-Santos, J.J., Felix-Herran, L.C., Tudon-Martinez, J.C., Ramirez-Mendoza, R.A., Morales-Menendez, R.: Control of automotive semi-active MR suspensions for in-wheel electric vehicles. Applied Sciences 10(13) 4522(1–28), 2020.
6. Hemanth, K., Kumar, H., Gangadharan, K.V.: Vertical dynamic analysis of a quarter car suspension system with MR damper. Journal of the Brazilian Society of Mechanical Sciences and Engineering 39(1), 41–51 (2017).
7. Lopez, I., Nijmeijer, H.: Prediction and validation of the energy dissipation of a friction
8. damper. Journal of Sound and Vibration 328, 396–410 (2009).
9. Talib, M.H.A., Darus, I.Z.M.: Intelligent fuzzy logic with firefly algorithm and particle swarm optimization for semi-active suspension system using magneto-rheological damper. Journal of Vibration and Control 23(3), 501–514 (2017).
10. Alexandru, C., Alexandru, P.: The virtual prototype of a mechatronic suspension system with active force control. WSEAS Transactions on Systems 9(9), 927–936 (2010).
11. Alexandru, C., Alexandru, P.: Control strategy for an active suspension system. Proceedings of World Academy of Science, Engineering and Technology 79, 126–131 (2011).
12. Formentin, S., Karimi, A.: A data-driven approach to mixed-sensitivity control with application to an active suspension system. IEEE Transactions on Industrial Informatics 9(4), 2293–2300 (2013).
13. Gohari, M., Tahmasebi, M.: Active off-road seat suspension system using intelligent active force control. Journal of Low Frequency Noise, Vibration and Active Control 34(4), 475–490 (2015).
14. Hyniova, K., Stribrsky, A., Honcu, J., Kruczek, A.: Active suspension system with linear electric motor. WSEAS Transactions on Systems 8(2), 278–287 (2009).
15. Konoiko, A., Kadhem, A., Saiful, I., Ghorbanian, N., Zweiri, Y., Sahinkaya, M.N.: Deep learning framework for controlling an active suspension system. Journal of Vibration and Control 25(17), 2316–2329 (2019).
16. Wanichanon, T., Thidrasamee, C., Pornrattanaprasert, T., Mahachotikul, P.: Stability analysis of an active vehicle suspension system. International Journal of Geomate 19(76), 110–117 (2020).
17. Alexandru, C.: Virtual prototyping platform for designing mechanical and mechatronic systems. Product Design, IntechOpen, 67–88 (2020).
18. Alexandru, P., Macaveiu, D., Alexandru, C.: A gear with translational wheel for a variable transmission ratio and applications to steering box. Mechanism and Machine Theory 52, 267–276 (2012).

19. Tarniță, D., Catană, M., Tarniță, D.N.: Contributions on the modeling and simulation of the human knee joint with applications to the robotic structures. Mechanisms and Machine Science 20, 283–297 (2014).
20. Țoțu, V., Alexandru, C.: Multi-criteria optimization of an innovative suspension system for race cars. Applied Sciences 11(9), 4167(1–25) (2021).
21. *** EASY5 user guide. MSC Software (2005).
22. *** Getting started using ADAMS/Controls. MSC Software (2005).
23. *** Getting started using ADAMS/View. MSC Software (2005).
24. Țoțu, V., Alexandru, C.: Suspension mechanism for light and single seat motor vehicles (Mecanism de suspensie pentru autovehicule ușoare și monoposturi). Patent no. RO130228 (2021).