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A New Semi-Active Suspension System for Racing Vehicles

The purpose of this paper is to enhance a passive suspension system with an electro-mechanical device in order to improve performance in terms of comfort, handling, and safety. The main goal is to develop a variable geometry suspension system of simple construction, small in size, which requires reduced energy for its implementation, and that is installable without substantial changes to the original passive suspension system through retrofitting operations. The device will be applied to a vehicle whose geometry is inspired by an open-wheel racing vehicle provided with a push-rod suspension. By means of a kinematic analysis, we evaluated geometry and kinematic properties of the suspension system, followed by CAD modeling and subsequent dynamic analysis. The kinematics of the system is analyzed by using the Lotus Suspension Analysis (LSA) software, while the multibody mechanical model is realized in the SimMechanics MATLAB Environment. Numerical simulations show the effectiveness of the proposed method.

Keywords: *Suspension, Retrofitting, Vibration, Variable-Geometry Suspension, Control.*

1. INTRODUCTION

In industrial applications, the design process of complex mechanical systems, such as for example a three-dimensional multi-body system, requires nonconventional methods for the analysis of the approximation present in a general design solution [1-10]. To this end, the complex dynamic behaviour of mechanical systems constrained by kinematic joints can be effectively studied within the multi-body system framework [11, 12]. Multi-body systems are mechanical systems composed of continuum bodies and kinematic joints which can be rigid and/or deformable [13-15]. Nonlinear force elements and time-varying control actuators are often applied to the bodies and the joints that form a general multi-body system [16-18]. Several examples of mechanical systems that can be modelled using the multi-body approach can be found in different engineering applications [19-22]. In general, it is well-known that the dynamic behaviour of a multi-body system is governed by a set of differential-algebraic equations of motion [23-25]. Therefore, advanced analysis approaches and computational procedures are necessary for performing reliable dynamic simulations [26, 27]. Furthermore, the design of effective control strategies for rigid-flexible multi-body systems is particularly challenging and, consequently, requires new computational approaches capable of handling the inherent nonlinear behaviour of a general multi-body system.

In automotive systems, the role of suspensions is to guarantee the contact between tire and road in order to assure good ride performance and road holding according to comfort-performance compromise. The suspension system determines critical components and parameters such as, for example, the height of the roll centre and the half-track change [28]. Since various safety properties and the cost of a vehicle are determined by the suspension geometry, the selected geometry has a great influence on the control design [29]. In this respect, active suspension systems allow for reaching the best compromise, depending on road conditions and type of trajectory, but they present some disadvantages as high energy consumption, high components prices, high weight and big dimensions. However, semi-active suspensions allow for obtaining a good compromise between performance and costs, leading to performances comparable to the active suspensions and reducing the energy required for its operation. Furthermore, compared to other variable geometry alternatives, variable geometry suspensions offer advantages such as an inherent fail-safe behaviour [30]. Another important example of application of the variable-geometry suspension is the steering of narrow vehicles [31].

In this paper, we propose a new concept of suspensions based on the retrofitting of a passive suspension without changing the spring stiffness coefficient or the viscous characteristics of the damper but simply varying the direction and the stroke of the suspension. Moreover, this variation, when opportunely controlled, allows for enhancing suspension performances. Thus, the main challenge is to develop a control system which can exalt the suspension system capabilities. Consequently, this kind of suspensions, named Variable-Geometry Suspension (VGS), is used to actively change geometrical parameters as roll centre, toe angle, wheel tilting and steering angle and orientation of

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shock absorber [32-36]. VGS suspensions allow to keep unchanged almost all component of native passive suspensions system. In our system, we added a crank actuated by a servomotor between the chassis and the upper strut mount. The first applications of this idea date back to the second half of the 1960s, when the English motorcycle manufacturer named Velocette, that introduced a device on their bikes that allowed the upper strut mount to move along a special guide following a curved path [37]. This solution enabled to manually vary the wheel-damper handling relation and, therefore, the dynamic behaviour of the system. In this paper, we want to investigate the use of such device on a racing car, developed at the Department of Industrial Engineering (DIIN) of the University of Salerno (UNISA).

We organised this paper in the following way: in section 2, we reported the description of the retrofitted device and the kinematic study performed using Lotus Software. In section 3, we showed the mathematical model used for evaluating the optimal motion laws for the left and right crank. In section 4, we evaluated the dynamic behaviour of the multi-body model of the racing vehicle when using the feed forward law. Due to the obvious simplifications of the mathematical model with respect to the physical model, we evaluated a feedback control law in order to compensate this difference. Finally, we present our conclusions. This paper is the starting point of future experimental activities that will be conducted on this class of suspensions for racing vehicles.

2. SYSTEM DESCRIPTION

The original passive suspension system of interest for this investigation is inspired by an open-wheel formula student racing car. The main features of the suspension system are double-wishbone layout and shock-absorber actuated through a push rod system connected to the lower wishbone [38]. In the configuration considered in this paper, the passive system has been retrofitted with a mechanical link placed between the shock-absorber eye and the attachment point on the chassis. This link is actuated through a servo-motor. Varying the mechanical link angle, one can change the spring and damper orientation as shown in figure 1.

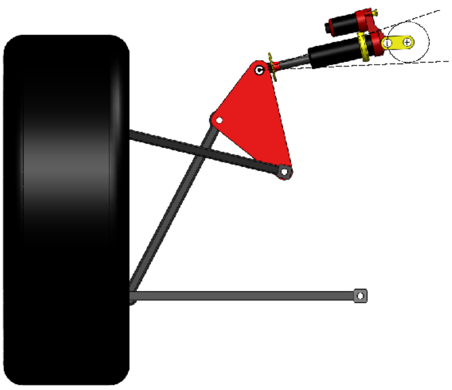

Figure 1. Original system retrofitted with new mechanical link to chassis

The change of inclination of the suspension system affects the way in which the elastic force and the damping force act [39, 40]. Therefore, actuating the link leads to a change in ride height and motion ratio.

2.1 Kinematic analysis of the original passive suspension system

To design the passive suspension system for the racing vehicle, we first performed a kinematic study. This preliminary study is necessary to define the geometry of passive suspension system that will be the starting point for our retrofitting activity. The software LSA (Lotus suspension analysis) allows us to verify the kinematic performance of the system in the function of chosen suspension layout as shown in figure 2.

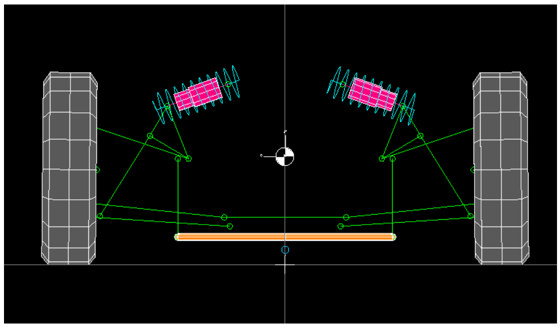

Figure 2. Kinematic simulation implemented in LSA

By means of the kinematic analysis, one can achieve all the dimensions that characterise the system and that allow for modelling the system through the use of SolidWorks CAD software in order to obtain all the physical properties of the system under analysis. The output of LSA software are plots that show the variation of selected kinematic parameters during characteristic movements of vehicle (roll, bump and rebound, steering). One must ensure to minimise the variation of parameters and that their trend is as linear as possible. In this way, the driver will find predictable the vehicle behaviour in order to improve lap times.

Figure 3 shows the variation of height of the roll centre from the ground. It is desirable that this height doesn't become negative during bump and rebound motions. Moreover, figure 4 shows the variation of camber angle during bump and rebound motion. Furthermore, the analysis carried out with Lotus software, allowed us to evaluate toe angle and motion ratio. Such relations have not been reported due to their small change in values.

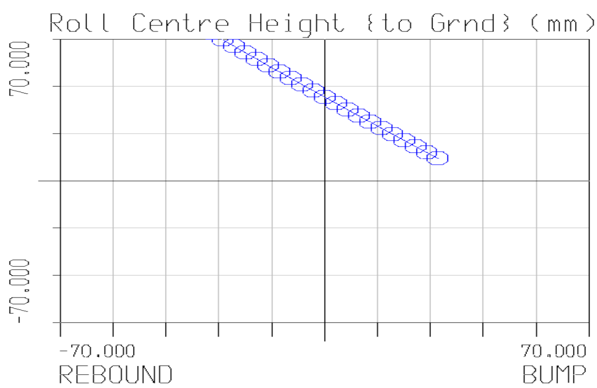

Figure 3. Roll center height to ground variation in bump and rebound for passive system

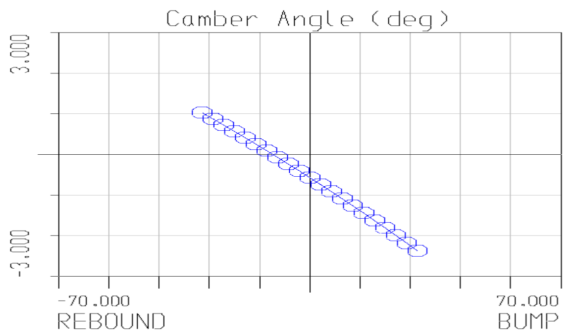

Figure 4. Camber angle variation in bump and rebound for passive system

2.2 Dynamic analysis of the half-car model

For evaluating the dynamic behaviour of our half race car model, we created a multi-body model by importing the CAD model in Simulink's Simscape environment [41].

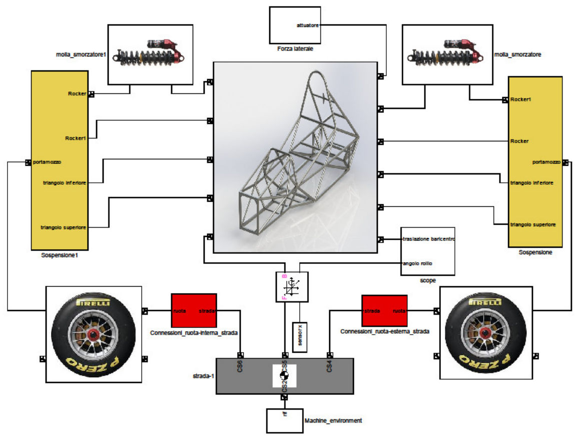

Figure 5. Multi-body model of system, Simulink interface

This environment allows to study the dynamic behaviour by using the mass and inertia properties drawn from the CAD model. Figure 5, shows the scheme of the half car multi-body model created.

3. SIMPLIFIED MATHEMATICAL MODEL

To evaluate the law of optimal motion of the two cranks, we studied the behaviour of the one degree of freedom simplified system shown in figure 6. For this model, we made the assumptions that left and right suspension device always acts in the vertical direction for varying crank positions.

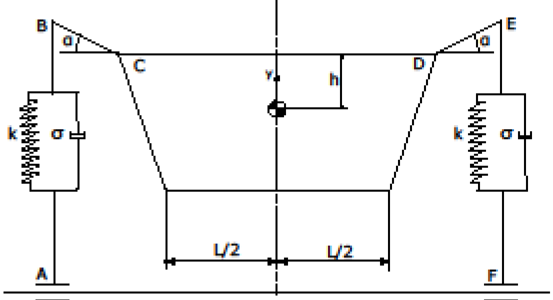

Figure 6. Physical model of retrofitted system

By using the Lagrange equation for deriving the equation of motion, one can write:

$$\frac{d}{dt}\frac{\partial E_c}{\partial \dot{q}} - \frac{\partial E_c}{\partial q} + \frac{\partial D}{\partial \dot{q}} + \frac{\partial V}{\partial q} = Q. \quad (1)$$

where E_c is the system kinetic energy, V is the system potential energy, D is the power dissipated by the system damping elements, and Q is the generalised forced vector of the system external forces. We obtained the following nonlinear equation of motion of the one degree of freedom system:

$$m\ddot{z}(t) + R\left(\dot{z}(t), \dot{\alpha}(t)\right) + K\left(z(t), \alpha(t)\right) = 0. \quad (2)$$

where m is the translating mass of the system, with R we reported the damping forces, function of vertical velocity $\dot{z}(t)$ and of the rotational velocity $\dot{\alpha}(t)$ of the left and right crank, while with K we denoted the elastic forces, function of the vertical position y of the chassis, and the angular position $\alpha(t)$ of the cranks. The profile of the road has been approximated by a sine wave of wavelength L, wave period T and amplitude Y as reported in (3),

$$y(t) = Y \sin\left(\frac{2\pi v}{L} t\right). \quad (3)$$

introducing with v the vehicle speed. By indicating with z_A and z_F the vertical position of the lower ends of the suspensions related to ground, we were able to evaluate the optimal angular position α and angular velocity $\dot{\alpha}$ for the two cranks reported in the following equations:

$$\alpha(t) = \sin^{-1}\left(\frac{-2hk + 2kL_r + kz_A(t) + kz_F(t)}{2kR}\right). \quad (4)$$

$$\dot{\alpha}(t) = \frac{k\left(\dot{z}_A + \dot{z}_F\right)}{2kR\sqrt{1 - \left(\frac{-2hk + 2kL_r + k\left(z_A(t) + z_F(t)\right)}{4k^2R^2}\right)^2}}. \quad (5)$$

In figures 7 and 8 we reported the vertical displacement and acceleration, respectively, of the chassis for the same harmonic displacement signal for z_A and z_F. As can be observed from the plots, the optimal relation evaluated for the angular position and velocity of the left and right crank allows four counteracting the forces generated by the vertical motion of point A and F. In this way, the system becomes no longer forced. This is the goal that we want to replicate for the multibody model of the racing vehicle.

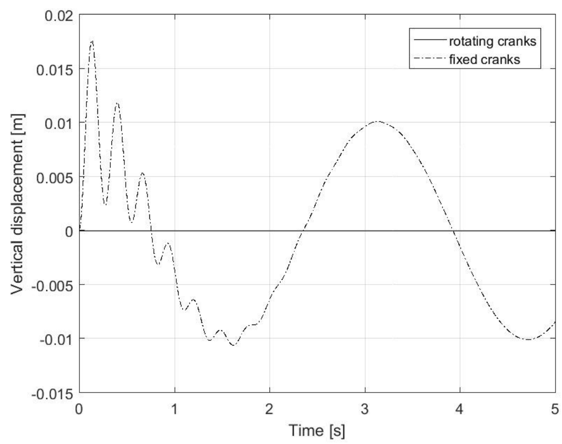

Figure 7: Vertical displacement of the 1 DOF system

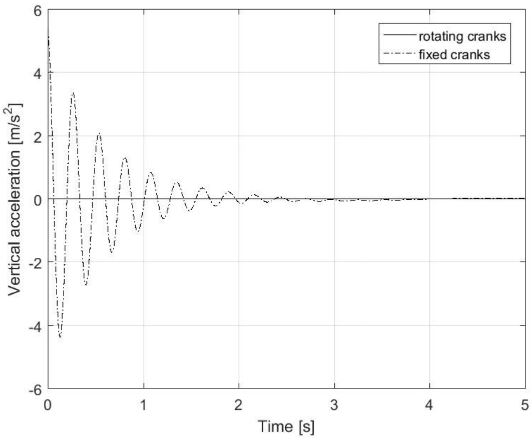

Figure 8. Vertical acceleration for the 1DOF system

4. THREE-DIMENSIONAL MULTIBODY MODEL

For evaluating the performance of the semi-active system, we decided to give more details to our multi-body model in order to obtain a behaviour as faithful as possible. In figure 9, we reported the multi-body model created for our numerical investigation. In red it is possible to see the linear actuators, whose task is to transmit the motion between actuator, simulating the road surface, and wheel, while the chassis has been hidden to highlight the retrofitted suspension system.

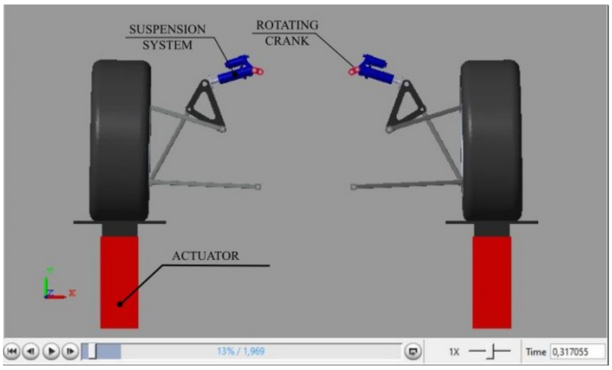

Figure 9. Multi-body half model of the race car

In blue we reported the passive suspension system while in orange we indicated the rotating cranks actuated by servomotors. To increase the accuracy of the model, the wheels have the ability to detach themselves from the motion actuators in the positive direction of the vertical axis, while they are constrained by a hard-stop along the negative direction. Furthermore, the motion law used to excite the system is a harmonic signal. The biggest difference, compared to the mathematical model, is the layout of the suspension that is substantially horizontal respect to the vertical suspension reported in figure 6. Such a difference results in a different system response to that of the mathematical model. Therefore, to the optimal relation, we will need to couple a closed loop control law. Figure 10 shows the closed loop control system used for the multi-body system.

Referring to the scheme, the regulator receives the difference $\delta x(t)$, that is the difference between the output of multi-body system and mathematical system, and produces an output $\delta u(t)$ that is the feedback (closed-loop) control input computed using a PID control strategy that added to the feedforward (open-loop) control law evaluated on the mathematical model will tend to minimise the error $\delta x(t)$.

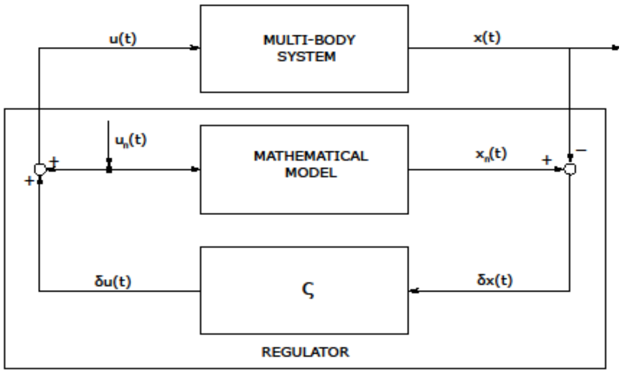

Figure 10. Closed loop control scheme

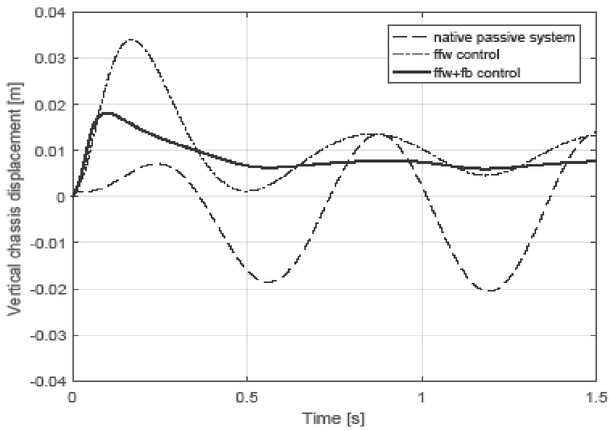

Figure 11. Vertical Chassis Displacement Comparison

In figure 11, we present for comparison the response of the multibody system for fixed angular position for the left and right crank with the response of the system under feedforward control law (FFW), and feedback control law (FB) for the rotating cranks under the same input signal reported in (3).

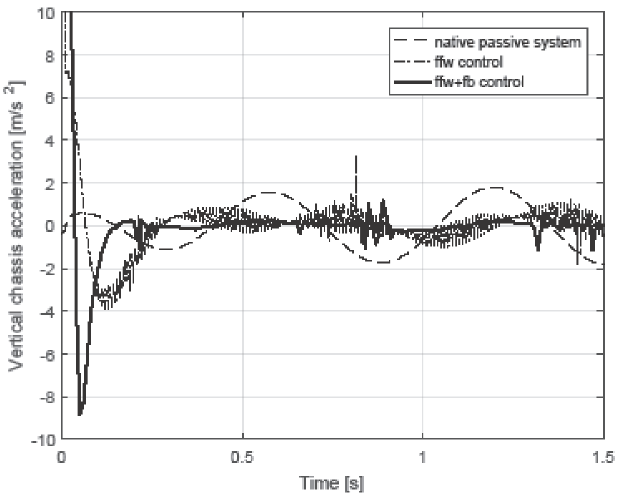

Figure 12. Vertical chassis acceleration comparison

In terms of displacement, the different steady state values assumed by the system depend on different configurations assumed by mechanical link, that can change the ride height. From the plot of figure 11, it is possible to notice how the feedforward control law helps to reduce the amplitude of oscillations, but the ideal zero amplitude cannot be reached because of existing differences between the adopted mathematical model and multibody system. This gap is reduced, as limit is nullified for a theoretical analysis, when the feedforward control law is coupled to the feedback control law. In this case, the amplitude of oscillations is strongly reduced. Furthermore, in terms of accelerations in both cases, there is an amplitude reduction compared to passive system as shown in figure 12.

5. CONCLUSIONS

The research conducted by the authors is focused on the development of new methodologies for performing accurate analytic modelling [42-45], numerical parameter identification based on experimental data [46-48], and optimal control optimisation for the dynamic models of rigid-flexible multi-body mechanical systems by using the interdependencies between multi-body dynamics, system identification, and control theory [49-52]. In particular, the purpose of this work was to design a device with characteristics comparable to active suspension without incurring in high energy consumption for their management. To do so, we decided to retrofit a passive suspension system by introducing two cranks driven by electro-mechanical devices. The vehicle chosen for our study is inspired by a formula student racing car. We carried out a kinematic analysis on the passive suspension with the aid of Lotus Software. Such analysis allowed us to understand the operating characteristics of the suspension and, at the same time, to define its geometry. Thereafter, it was possible to draw the half car model for subsequent dynamic analysis by using Solidworks CAD software. We evaluated the optimal motion laws for the two cranks by using a one degree of freedom mechanical system. The feed-forward control laws have been tested by using a multi-body model developed in SimScape, that is the multi-body environment of the Matlab software. Despite the differences between the multi-body model and the simplified model, the use of the optimal motion law for the two cranks produced a good reduction of the chassis displacement. Furthermore, by adding a regulator to the optimal motion laws, we were able to significantly reduce the oscillations of the chassis, while maintaining the accelerations under control. This preliminary research represents the first step towards further studies on the performance of semi-active suspension system obtained by means of retrofitting techniques. In future works, the numerical results of the simulations conducted on full car models will be compared to experimental results conducted on a retrofitted 1/8 scale car model.

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РАЗВОЈ И ИМПЛЕМЕНТАЦИЈА СИСТЕМА УПРАВЉАЊА КОД РЕКОНСТРУИСАНЕ ЦНЦ МАШИНЕ ПРИМЕНОМ АРДУИНО ПЛАТФОРМЕ

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Рад се бави развојем контролера без повратне спреге који је примењен код Ардуино платформе у циљу поновног коришћења постојеће ЦНЦ машине за извођење једноставних производних операција. ЦНЦ машина о којој је реч у овој студији је штампач Objet Quadra Tempo 3D. Циљ овог рада је да се једноставном реконструкцијом изврши конверзија машине, која се сматра застарелом због високих трошкова одржавања, коришћењем јефтиних компоненаната из постојећих залиха и отвореним софтвером у циљу смањења електронског и индустријског отпада. Микроконтролер ArduinoMega 2560 је искоришћен за управљање драјверима степер мотора машине. Овај микроконтролер омогућава једноставно управљање аналогним и дигиталним уређајима. Целокупна реконструкција је извршена у циљу додавања и одузимања производних операција. Активности приказане у овом раду обављене су помоћу инсталираног електро-вретена за обраду дрвета и поликарбоната. Коришћење јефтиних компонената омогућило је трансформацију 3Д принтера у ЦНЦ глодалицу која може да обрађује материјале као што су дрво и поликарбонати.