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DESIGN, MODELING, AND CONTROL OF ACTIVE HYDRAULIC SUSPENSION SYSTEM FOR VEHICLES

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ABSTRACT

Suspension systems have been extensively used in automobiles for better ride comfort and stability. Long rides on irregular roads and infrastructure problems like uncomfortable seating have a very bad impact on human body. To reduce this factor, an active suspension system that uses hydraulic piston-cylinder arrangement to actuate necessary force is designed. The proposed hydraulic system is controlled by a PID controller, which controls the hydraulic pump rpm and valve opening. The response of the active system, simulated with 3 different road profiles, is then compared against the response of a passive suspension system. Finally, ADAMS/MATLAB co-simulation was carried out to ascertain the validity of the system with the non-linear model modeled in ADAMS. It is found that the proposed active suspension system has more effective capacity to reject the road disturbances..

***Keywords*:** Hydraulics, Matlab/Simulink, PID, Quarter Car, Suspension

NOMENCLATURE

M Sprung mass (quarter-car)
k Spring constant of passive suspension system
B Damping constant of passive suspension system
M_w Unsprung mass (wheel)
k_w Spring constant of the wheel
r Road disturbance in vertical direction
z Vertical displacement of the sprung mass
z_w Vertical displacement of the unsprung mass
F Force generated by active suspension system

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1 INTRODUCTION

When passengers evaluate the ride quality of a vehicle, many factors like acoustic quality, comfort of seats, climate control system, etc. come into play making it a subjective opinion. A suspension system is used to provide passengers with a comfortable ride primarily by isolating them from vibrations and shock, improving mobility and facilitating vehicle control [1]. A good suspension system rejects track level variations like bumps and road noises, keeping the passengers well isolated from these disturbances. A conventional solution is to use passive components like springs and dampers to mitigate the effects of these disturbances. For better ride comfort and ride safety, however, semi active or active suspension systems are preferred.

Semi active suspension systems are identical to passive one but with some control of damping coefficient is achieved by switching the characteristics of dampers. Consequently, this gives the possibility of the damper reaction forces.However, since it is a semi-active system, no active force can be applied and therefore, total roll and pitch elimination is impossible [2]. An active suspension is one including an actuator that can supply active force, which is regulated by a control algorithm using data from sensors attached to the vehicle. An active suspension is composed of an actuator and a mechanical spring and a damper [3].This system acquire the ability to reduce acceleration of sprung mass continuously as well as to minimize suspension deflection, which results in progression of tire grip with the road surface, thus, brake, grip control and vehicle maneuverability can be considerably improved [4]. In this paper, we propose a PID controlled active suspension system modeled with ADAMS and MATLAB/Simulink with hydraulic actuation for generation of required force. Finally, the controller is deployed on a non-linear system designed in ADAMS for the validation of the proposed technique.

2 Modeling of the System

Various models like quarter car, half car, full car, etc have been used to simplify the dynamics of vehicle suspension systems. Figure 1 represents a quarter car model. The quarter-car model has two degrees of freedom, describing the vertical vibrations of a sprung mass and an unsprung mass. The sprung mass represents the vehicle body and the unsprung mass represents the wheel. A passive suspension system comprising a spring-damper system connects the two masses. The wheel properties in the radial direction are modelled by the spring that links the unsprung mass to the road. [5]

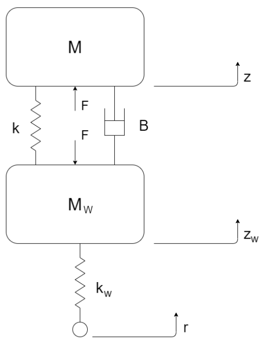


Fig. 1: Quarter Car Model

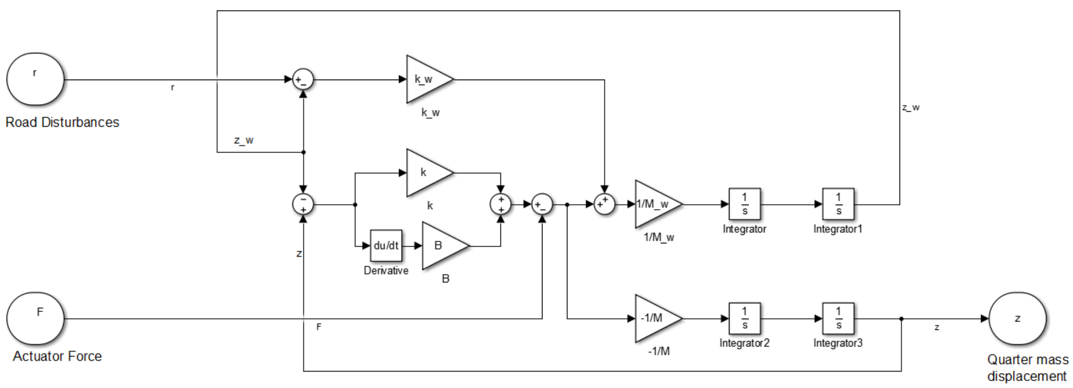


Fig. 2: Block diagram representation of quarter car model in Simulink

2.1 Mathematical Modelling of the Vehicle

Following equation governs the dynamics of the unsprung mass:

$$M_w\ddot{z}_w = k_w(r - z_w) + k(z - z_w) + B(\dot{z} - \dot{z}_w) - F \tag{1}$$

Similarly, for the sprung mass:

$$M\ddot{z} = k(z_w - z) + B(\dot{z}_w - \dot{z}) + F \tag{2}$$

For this system of quarter car, we choose 4 state variables, namely x_1, x_2, x_3 and x_4 such that:
x_1 = displacement of sprung mass = z
x_2 = velocity of sprung mass = $\dot{z}$
x_3 = relative displacement of sprung mass w.r.t unsprung mass = $z - z_w$
x_4 = relative velocity of sprung mass w.r.t unsprung mass = $\dot{z} - \dot{z}_w$

Then, the time derivatives of these state variables are given by:

$$\dot{x}_1 = \dot{z} = x_2 \tag{3}$$

$$\dot{x}_2 = \ddot{z} = k/M(z_w - z) + B/M(\dot{z}_w - \dot{z}) + F/M = (-k/M)x_3 + (-B/M)x_4 + (1/M)F \tag{4}$$

$$\dot{x}_3 = \dot{z} - \dot{z}_w = x_4 \tag{5}$$

$$\begin{aligned}\dot{x}_4 &= \ddot{z} - \ddot{z}_w = k/M(z_w - z) + B/M(\dot{z}_w - \dot{z}) + F/M - k/M_w(z - z_w) - B(\dot{z} - \dot{z}_w) + F/M_w - k_w/M_w(r - z_w) \\ &= -k/Mx_3 - B/Mx_4 + F/M - k/M_wx_3 - B/M_wx_4 - k_w/M_wx_3 + k_w/M_wx_1 - k_w/M_wr + F/M_w \\ &= (k_w/M_w)x_1 + (-k/M - k/M_w - k_w/M_w)x3 + (-B/M - B/M_w)x_4 + (1/M + 1/M_w)F + (-k_w/M_w)r\end{aligned} \tag{6}$$

From above set of equations, we can write the state space equation $\dot{x} = Ax + Bu$ as:

$$\begin{bmatrix}\dot{x}_1\\ \dot{x}_2\\ \dot{x}_3\\ \dot{x}_4\end{bmatrix} = \begin{bmatrix}0 & 1 & 0 & 0\\ 0 & 0 & -k/M & -B/M\\ 0 & 0 & 0 & 1\\ k_w/M_w & 0 & -k/M - k/M_w - k_w/M_w & -B/M - B/M_w\end{bmatrix} * \begin{bmatrix}x_1\\ x_2\\ x_3\\ x_4\end{bmatrix} + \begin{bmatrix}0 & 0\\ 1/M & 0\\ 0 & 0\\ 1/M + 1/M_w & -k_w/M_w\end{bmatrix} * \begin{bmatrix}F\\ r\end{bmatrix}$$

2.2 Modelling of the Hydraulic system

A hydraulic system is a actuation system where a fluid is used to transfer the energy from e.g. an electric motor to an actuator, such as a hydraulic cylinder. Hydraulic systems employed in several industrial and mobile applications present significant advantages, such as a high power-to-weight ratio and fast dynamic response. Hydraulic system provides a smooth displacement transition throughout changes in levels of speed, which provides an opportunity to deploy the system to the automotive systems [6]. Fluid used in this system is theoretically in-compressible. In this paper, we have modelled the hydraulic system in Simscape/Matlab and combined it with the multi-body non linear system imported from ADAMS to get the required force for the Adams system. The model used is as shown in figure 3. A fixed displacement pump was used to generate the

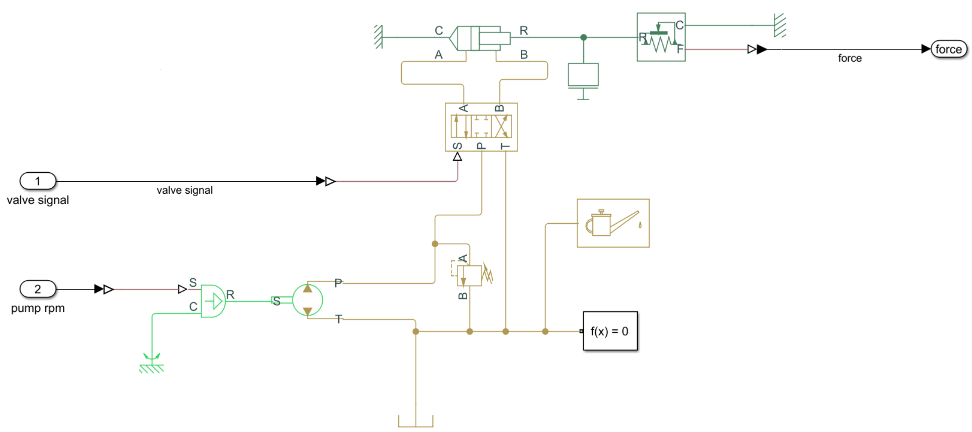


Fig. 3: Hydraulics Block

required pressure force. It is connected to a ideal velocity source which runs in an rpm controlled by the controller. Since, power and rpm are directly proportional, the controlled rpm will result in controlled power input. This results in a controlled pressure force generation for the adams subsystem. The parameters set are as tabulated below:

Table 1: Displacement Pump Parameters

Nominal Shaft Angular Velocity	500 rad/s
Nominal Pressure Gain	10^6 Pa
Nominal Kinematic Viscosity	46 cst
Nominal Fluid Density	998 kg/m^3
Volumetric Efficiency at Nominal Conditions	92%

The fluid pumped then enters 4-way directional valve, which directs the flow path of the piston-cylinder arrangement. The valve is triggered by a signal which is regulated with respect to wheel displacement. Fluid which returns from the cylinder is directed directly to the reservoir. The basic parameters of valve is as shown in table 2.

Table 2: 4 way directional valve parameters

Leakage area	$3.14mm^2$
Flow Discharge Coefficient	0.7
Laminar Flow Pressure Ratio	0.99
Maximum Opening	20 mm
Maximum Opening Area	314.15 mm^2

A pressure relief valve is placed in the flow passage of pump and valve. It regulates the pressure in the flow channel and does not let pressure exceed the nominal pressure. This prevents the flow channel from the mechanical failures. If pressure exceeds the nominal pressure, the valve opens and fluid is directed to the reservoir. Valve opening dynamics is neglected and the relation between area and pressure is assumed linear. The basic parameters of relief valve is as shown in table 3.

Table 3: Pressure relief valve parameters

Maximum Passage Area	2e-4 m^2
Valve Pressure Setting	3e7 Pa
Valve regulation range	3e6 Pa
Flow Discharge Coefficient	0.7
Leakage Area	1e-12 m^2
Laminar Flow Pressure Ratio	0.99

Fluid finally reaches piston-cylinder arrangement from where the effective work is extracted. The force is sensed with a sensor which senses force with respect to piston displacement. This force is then labelled as subsystem output and fed into force input of suspension system.

Table 4: Basic Piston Parameters

Piston Area A	1256 mm^2
Piston Area B	803.84 mm^2
Piston Stroke	100 mm
Dead Volume A	12560 mm^3
Dead Volume B	8042.24 mm^3
Specific Heat Ratio	1.4

The fluid viscosity index is highly important in mobile systems. The efficiency of the system is strongly dependent on the viscosity of the hydraulic fluid [7]. It is also stated that synthetic ester fluids gives lower friction losses than standard mineral oil [8] [7]. The shear stability of saturated esters is extremely good compared to mineral oil and unsaturated esters. The fluid used in our hydraulic system is iso vg 46 as it is a ester with efficiency of 70% even at higher loads.

2.3 Modelling of Controller

To obtain desired response from a system, an effective controller must be designed. Despite being a traditional control method, properties of PID controller like easy algorithm, high reliability, flexibility, etc. has made it dominant in many industries. The PID controller manipulates the error signal to output the control signal. The control signal is a sum of three terms: a proportional term that is proportional to the error, an integral term that is proportional to the integral of the error, and a derivative term that is proportional to the derivative of the error.

$$u(t) = k_p e(t) + k_i \int_0^t e(\tau) d\tau + k_d \frac{de(t)}{dt} \tag{7}$$

The proportional gain k_p, integral gain k_i, and derivative gain k_d are controller parameters. The proportional part acts on the present value of the error, the integral represents an average of past errors and the derivative can be interpreted as a prediction of future errors. [9]

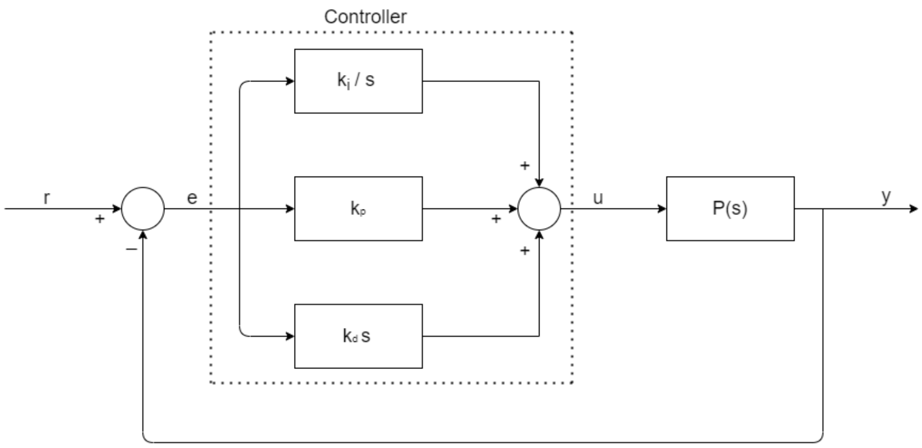


Fig. 4: PID controller

3 Modelling of the Road profile

Three kinds of road profiles are considered to illustrate the effectiveness of the proposed suspension system.

3.1 Single bump

$$r(t) = \begin{cases} \frac{h}{2}(1-\cos(\frac{2\pi V_o}{l}t)), & 0 \leq t \leq \frac{l}{V_o} \\ 0, & t > \frac{l}{V_o} \end{cases}$$

where, h = 0.05m is the height of the bump, l = 5m is the length of the bump respectively and V_o = 10m/s is the forward velocity of the vehicle.

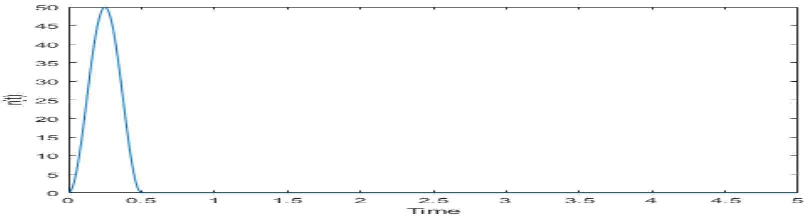


Fig. 5: Single-bump road profile

3.2 Numerically generated random road profile

The profile is generated as a sum of series of harmonics as presented in equation below:

$$r(x) = \sum_{n=1}^{N} Z_n sin(2\pi\Omega_n x + \phi_n), n = 1,2,3....N$$

where,
N is the number of harmonic samples and also the number of frequency bands.
Ω_n is the spatial frequency of nth harmonic. It is computed by linear interpolation as Ω_n = Ω_{min} + $\Delta\Omega$ (i-1), the frequency increment $\Delta\Omega$ being equal to ($\Omega_{max} - \Omega_{min}$)/N.
ϕ_n is a random phase angle uniformly disributed in the interval [0, 2π].
$Z_n = 2^a \dfrac{\Omega_0}{\Omega_n}\sqrt{2\Delta\Omega} * 10^{-3}$ is the amplitude of nth harmonic.
[10] [11]

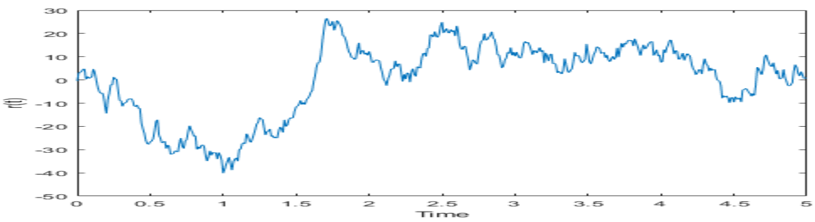


Fig. 6: Numerically generated C class road profile

3.3 Superposition of sine waves of different frequencies

This road profile is generated as a sum of three different sine waves.

$$r(t) = 0.02 \sin 2\pi t + 0.01 \sin 10\pi t + 0.001 \sin 16\pi t$$

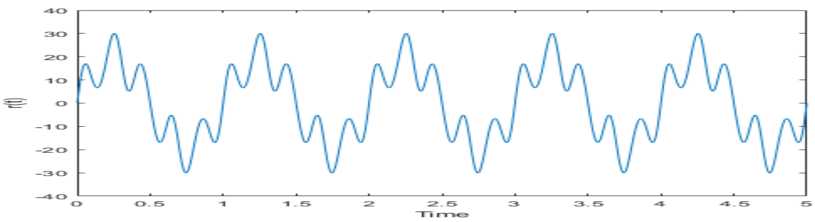


Fig. 7: Road as superposition of sine waves

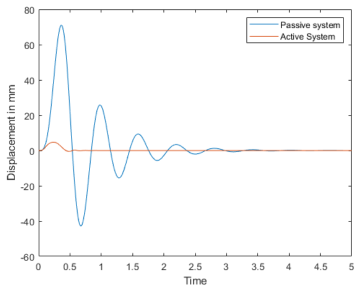
Table 5: Quarter car model parameters

M	232.7 kg
M_w	48.4 kg
k	28084 N/m
B	960.3 Ns/m
k_w	242000 N/m

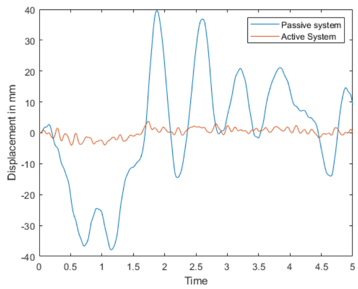
4 EXPERIMENTAL (NUMERICAL) DETAILS

S.-W. Kang et al. used a mid-size passenger vehicle, model ALL NEW K7 of KIA Motor Company as test-bed vehicles. They used the sensor data for system identification. Thus obtained parameters of quarter car model are listed in the table 5 [12].

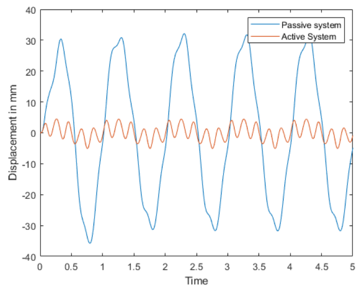
A quarter car with dynamics explained in section 2 was simulated with three different types of road profiles as described in section 3.



(a) Road type 3.1

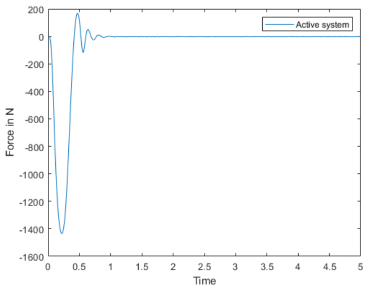


(b) Road type (b)

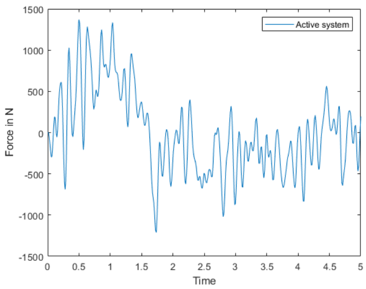


(c) Road type (c)

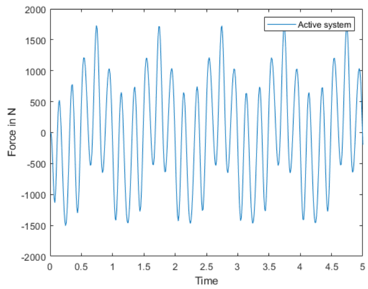
Fig. 8: Sprung mass deflection for three different road profiles (quarter car model)



(a) Road type (a)



(b) Road type (b)



(c) Road type (c)

Fig. 9: Control force for three different road profiles (quarter car model)

The non-linear model accounts friction, non-linearity of the spring, contact dynamics due to materials and other non-linear properties. The controllers designed and the technique presented in this paper is validated with non linear model of double wishbone suspension system. The non linear model was defined in Adams and co-simulated with Matlab/Simulink for

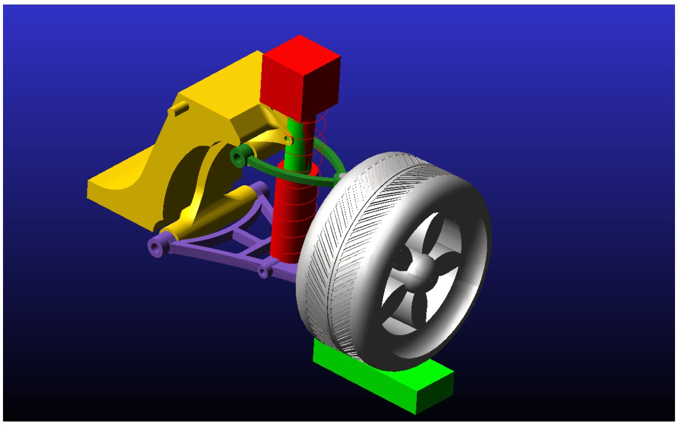
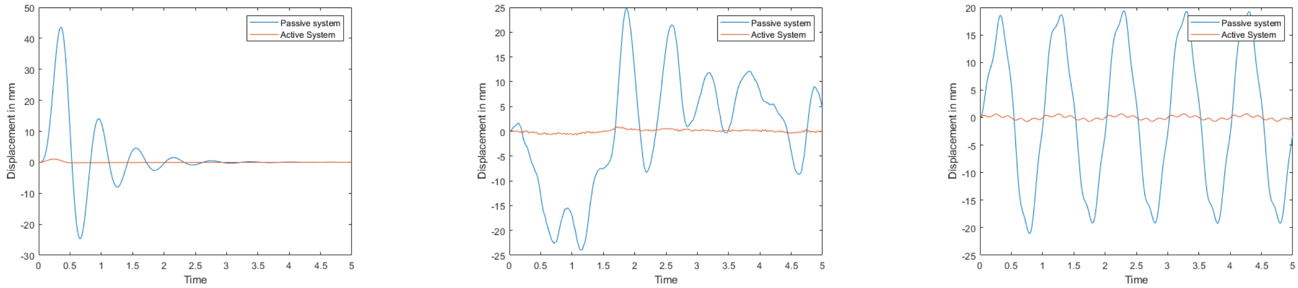


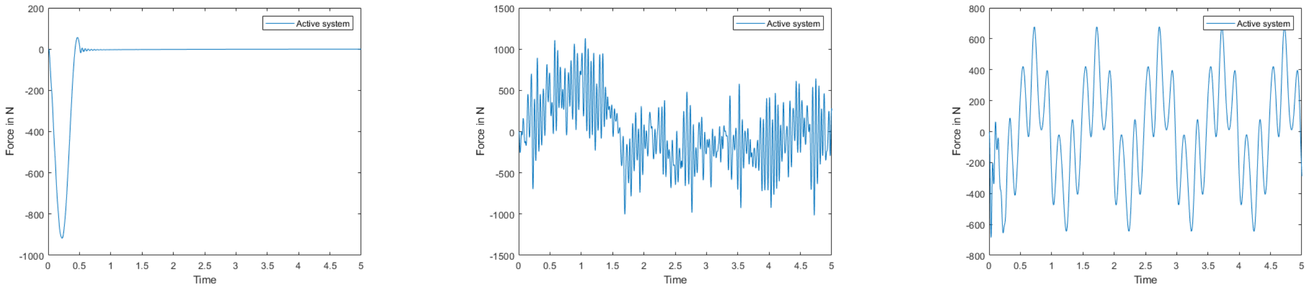
Fig. 10: Double wishbone model designed in Adams

examining the system with the proposed control method. The response of this model for the same road profile shows reduced sprung mass displacement and control force. This is due to the fact that the double-wishbone incorporates



(a) Road type (a) (b) Road type (b) (c) Road type (c)

Fig. 11: Sprung mass deflection for three different road profiles (double wishbone model)



(a) Road type (a) (b) Road type (b) (c) Road type (c)

Fig. 12: Control force for three different road profiles (double wishbone model)

The sprung mass displacement of an active system is compared to a passive system in figure 8 and 11. From the bump response of the systems, it is clear that the active system brings a significant reduction in amplitude as well as the settling time. The hydraulic system discussed in section 2 provides the necessary force for this reduction. The forces exerted by the hydraulic system for different road profiles are presented in figure 12.

It can be noted that the passive system seems to have higher response time than that of the active system. Due to this displacement curve of passive system in case of road (b) and (c) is smoother than that of active system. The jagged response curve of the active system is due to its faster response time. This sensitivity compels the system to account even trivial irregularities of the road. This can also be visualized with figures 9b, 9c, 12b and 12c where the nature of forces almost resembles the nature of road profile.

5 RESULTS AND DISCUSSION

This paper has proposed a hydraulic actuator suspension system based on a PID controller to improve vehicle ride comfort. The suspension system was first designed for a quarter-car model and then implemented on a double-wishbone model designed on Adams. The systems were simulated for different models of road profiles. Throughout the research, the simulation results of the passive and the active suspension system are juxtaposed and the comparison between them indicates a significant improvement in the system response when active suspension system is used. Although the wishbone system imitates a real suspension system and accounts non-linearity of the system to a considerable extent, there are still few insufficiencies in the methodology introduced in this paper. The hydraulic system designed in this paper is almost ideal in many aspects. The forces in the system are assumed to have only translational effects in vertical direction and the rotational effects of these forces (moments) are not considered at all. However, these assumptions do not make a substantial change in the effectiveness of the system.

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