Design and Analysis of Vehicle Suspension System

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***Abstract*— Recently the study of the vehicle dynamics is essential to design robust vehicles with better performance for customer satisfaction. Ride Comfort (RC) and Road Holding (RH) are very important concern related to vehicle suspension system. This paper presents analysis of vehicle suspension using Quarter car model developed in MATLAB. Using design of experiment (DOE) the quarter car model is tested on a test rig. Design of experiments (DOE) is used to validate simulation result which is performed in quarter car test rig. Validated model is further used for analysis. Correlation index obtained between simulation and experimentation is 0.56.**

ISO 2631:1 1997 defines the criteria to evaluate whole body vibration (WBV) experienced by passengers during the ride. It also defines vibration limits for un-comfort, safety and health hazards using vertical accelerations. Above validated model of SIMULINK analyzed further to find effect of stiffness (K), damper coefficient (C), sprung mass (M) and velocity (N) on RC and RH.

Performance of passive suspension model is compared with semi-active suspension system for different vehicle velocities and road inputs using MATLAB SIMULINK. Semi-active suspension system with Proportional-Derivative-Integrator (PID) controller is developed which gives optimal and robust system accounting different road conditions and vehicle speeds to increase RC and RH. It is observed that percentage improvement in settling time for different road profiles i.e. half sine bump, double sine bump, trapezoidal bump, step bump and square bump are 51%, 40%, 42.3%, 34.9% and 48.8% respectively.

***Keywords*— MATLAB/SIMULINK, PID controller, Passive suspension, Quarter car, semi-active suspension system suspension, Taguchi method, Ride comfort, Road holding**

I. Introduction

MODERN trend is to design vehicle for safety and comfort is very important concern. Driver seat is one of the main aspects to be considered while defining comfort in a moving vehicle. The analysis is focused on driver seat because driver comfort is of main concern since it is the most occupied seat in any vehicle and the occupancy is for longer duration. In addition to sitting, the driver's job is to manipulate different controls and concentrate parallel on many aspects. The research work aims at studying the vertical vibrations transferred to the human body via seat [1,2,3]. Vertical accelerations (RMS) directly affect ride comfort and road holding. In passive suspension system ride comfort is more for lesser root mean square (RMS) acceleration but road holding is less therefore there is need to find optimized value of K, C, M, N to get desired ride comfort and ride holding. Relative displacement between road and sprung mass measures road holding (RH). Exposure to whole body vibration associated with a prolonged seating results in un-comfort and low back pain among drivers. Both vehicle suspension system and driver seat cushion designs have attracted significant interest over the last several decades with a significant effort being directed towards their improvements. Vibration attenuation through the suspension and seat will not only provide riding comfort but also reduce the risk of low back pain due to driving [4].

There are basically three types of suspension systems passive, semi-active and active suspension system. In this paper passive and semi active suspension system are analyzed. Passive suspension system consists of spring and damper coefficient. Damper acts as an energy dissipating element while spring acts as an energy-storing element. Since these two elements do not add energy to the system this kind of suspension systems are called passive. Parameters are generally fixed in passive suspension system [5,6,7]. Passive suspension system works as open loop control system. In Semi-active suspension system spring element is retained, but the damper is replaced with a controllable damper which requires small amount of external energy [7,8]. It works as a closed loop control system.

In passive suspension system so far as several models have been developed such as quarter car, half car and full car. R. Alkhatib et al. (2004) and O¨. Gu¨ ndog˘du (2007) worked on Genetic algorithm (GA) to optimize problem of linear one degree vibration isolator and method is extended to the optimization linear quarter car suspension model by using SIMULINK model [9,10]. Galal Ali Hassaan (2014, 2015) analyzed quarter car model of SIMULINK and showed that for ride comfort the car speed has not to exceed 6.75 km/h when passing a circular hump depending on the suspension damping [11,12]. Anirban. C. Mitra et al. (2013) studied quarter model with thorax, pelvis and found effects of variations of suspension stiffness and damping coefficient on ride comfort, road holding and head displacement over wide range of road bump using SIMULINK and Bond graph and also used ISO 2631:1 1997 to study amplitude of vibrations of human body as well as vehicle [13]. M.J. Pable et al. (2007, 2008) worked on design of passive suspension system to find the best parameter of passive suspension system which provides performance as close to active suspension system [14] and also worked on design of road friendly suspension system for heavy suspension for quarter car model [15]. Anirban C. Mitra et al. (2014) and Saeed Mostaani et al. (2011) worked on DOE for optimization of vehicle suspension parameter for ride comfort [16, 17].

Spring and damper are parameters which remain constant in passive suspension system directly affect ride comfort and

road holding. If damper is heavily damped or too hard suspension it will transfer a lot of road input or throwing the car on unevenness of the road. If it is lightly damped or soft suspension reduces the stability of vehicle in turns or change lane or it will swing the car. While in case of spring ride comfort increases with soft spring and reduced with heavy spring [13].

In this paper quarter car model is developed in SIMULINK and experimental validation is performed on a quarter car test rig by using DOE and estimated the response of values of K, C, M and N on RC. Validated model is further used to analysis the effect different parameters on RC and RH. Also compare semi-active suspension system over passive suspension system in this paper.

II. Development of Quarter Car Model

Quarter-car models are extensively used in automobiles due to their simplicity and it provides qualitatively correct information, at least in the initial design stages of vehicle dynamics. Quarter-car model as shown in Figure 1 is very often used for suspension analysis as it capture important characteristics of full model. The equations of motions are found by adding vertical forces on the sprung and un-sprung masse. In quarter-car model M represents the sprung mass, while tire and axles are illustrated by the un-sprung mass Mu. The spring, shock absorber and a variable force-generating elements placed between the sprung and un-sprung masses constitute suspension and un-sprung masses connected to road by tire stiffness. Fig. 2 shows free body diagram of quarter car model.

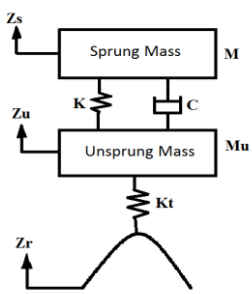

Fig. 1 Quarter car model

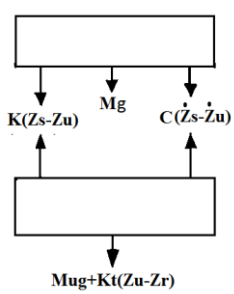

Fig. 2 Free body diagram of Quarter car

From free body diaram with the help of Newton's second law following equations written as follows:

$$M\ddot{Zs} + C \times (\dot{Zs} - \dot{Zu}) + K \times (Zs - Zu) + M \times g = 0 \quad (1.1)$$

$$Mu\ddot{Zu} - C \times (\dot{Zs} - \dot{Zu}) - K \times (Zs - Zu) + Kt \times (Zu - Zr) + Mu \times g = 0 \quad (1.2)$$

By using equations (1.1) and (1.2) mathematical model is made in MATLAB SIMULINK as shown in Fig. 3.

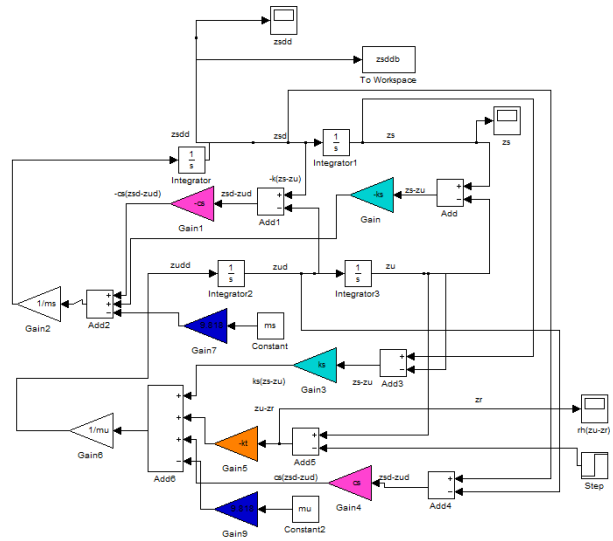

Fig. 3 SIMULINK model of quarter car test rig

III. Experimentation

A. Quarter car test rig

A quarter mass of vehicle suspension system is taken for the experimentation called as quarter car test rig as shown in Fig 5. It consists of upper mass M representing the quarter mass of vehicle and lower mass Mu representing unsprung mass of the wheel and other suspension parts. K and C represent spring stiffness and damping coefficient of suspension system respectively. The tire stiffness is represented by Kt and is assumed constant. Road profile is generted by cam which is in contact with tire. Cam is rotated by DC motor and speed is measured by Taccometer. Accelerometer measures RMS accelerations of sprung mass and unsprung mass. NI-9234 DAQ hardware and LABVIEW (ver: 2014) coding software are used for data acquation. The aim of present work is to analyze Multi-objective optimization formulations for RC by determining the effects of factors K, C, M, N and their interactions by regression analysis. The Design of Experiments (DOE) Methodology is used during experimentation. Taguchi method and minitab software is used to analyze effects of each factor on RC.

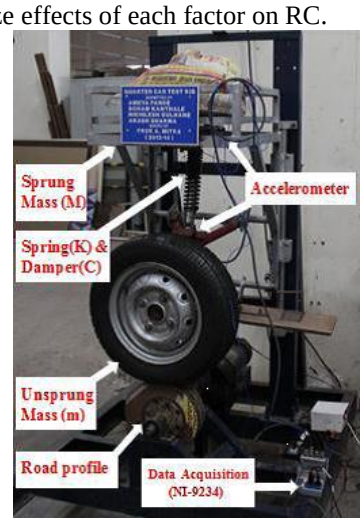

Fig. 5 Quarter car test rig

B. Design of Experiments (DOE)

DOE is statistics based approach. A methodology to achieve a predictive knowledge of a complex, multi-variable process with the fewest acceptable trials. Taguchi is one of the method in DOE which is used to determine effect of K, C, M and N on RC by experimentation.

In Taguchi method all interactions of parameters are not considered. The main concern is deviation of a characteristic from its nominal value. Uncontrollable factors (noise) are often responsible for this deviation and therefore Taguchi's approach to experimental design has as its goal the design of products/process that are robust to these noise factors. Taguchi method basically consists of three stages i.e. system design, parameter design and tolerence design. In Taguchi method two levels of parameters K, C, M, N are used for experimentation and corresponding high and low level values are as shown in Table I. Half sine bump is used as road input. To simulate SIMULINK model and do the experimentation inputs used as tire stiffness of 240000 N/m and unsprung mass 25Kg.

Fig. 4 show flow chart for Tguchi method. These values are taken from various shock absorbers data. "Orthogonal Arrays" (OA) provides a set of well balanced (minimum) experiments which serve as objective functions for optimization. OA gives significant reduction in size of experiments. OA matrix of reading is as shown in Table II. RMS acceleration of spung mass measures by accelerometer and same K, C, M, N values used in SIMULINK model obtain ride comfort (RC). Experimental RC and simulated RC are given in Table II.

TABLE I
HIGH AND LOW LEVEL INPUT FACTORS

Input factors	High level (+1)	Low level (-1)
Spring stiffness (K) (N/m)	26000	18000
Damping coefficient (C) (N-s/m)	673	418
Sprung mass(M) (kg)	81	41
Speed of cam (N) (RPM)	200	150

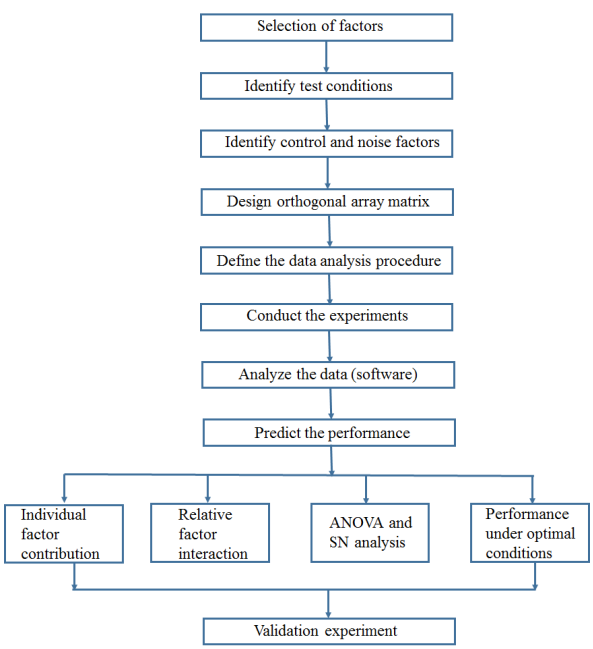

Fig. 4 Flow chart of Taguchi method

TABLE II
EXPERIMENTAL READINGS OF RC AND RH AS PER ORTHOGONAL ARRAY DOE MATRIX

Run	K	C	N	M	RC_expt	RC_Simu
1	18000	418	150	41	0.521241	0.4202
2	18000	418	150	81	0.95031	0.5164
3	18000	418	200	41	0.90962	0.5242
4	18000	418	200	81	0.579763	0.5408
5	18000	673	150	41	0.420568	0.3663
6	18000	673	150	81	0.490051	0.4504
7	18000	673	200	41	0.740373	0.4282
8	18000	673	200	81	0.594507	0.4681
9	26000	418	150	41	0.498241	0.3659
10	26000	418	150	81	0.540709	0.5841
11	26000	418	200	41	0.705375	0.5417
12	26000	418	200	81	0.760319	0.6664
13	26000	673	150	41	0.465926	0.365
14	26000	673	150	81	0.534968	0.5063
15	26000	673	200	41	0.65164	0.5063
16	26000	673	200	81	1.190254	0.5826
					Coefficient of Index (CI)=0.567415007	

Correlation of Index (CI) is defined as arregenment of RC value obtained by experimental and SIMULINK model. The CI value obtained for present analysis is 0.56 which shows good co-relation. Maximum value of CI is 1. Here reason of getting value 0.56 because in SIMULINK model there is not consideration of actual conditions which are experinced in test rig. Hence further analysis carriedout using quarter car SIMULINK model.

The signal to noise ratio (SN) is a measure of the impact of noise factors on the performance. The SN ratio depends on whether the quality characteristic is smaller-the-better, larger-the-better, or nominal-the-best[20]. Smaller to better is used in this paper which gives all negative values and most negative reprsents more sensitive factor. The larger the SN, the more robust product is against noise. Table III shows SN ratio for K,C, M, N. The null hypothesis is rejected if the P-value is less than the confidence level of the model, i.e. 0.05. Response of each parameter is as shown in Table IV. Rank of each parameter represents which is the most effective parameteer on RC. N is the most effective parameter as per the observations.Taguchi method can not judge and determine effect of individual parameters on entire process while percentage contribution of each parameter can be determined by analysis of variance (ANOVA). For the model ANOVA is shown in Table V. Fig. 6 indicates the sensitivity analysis of mean value of RC with main factors as K, C and N. From these graphs it is concluded that K and M are the most sensitive parameters and C is insensitive factor for RC.

TABLE III
MODEL COEFFICIENTS FOR SN RATIOS

Term	Coef	SE Coef	T	P
Constant	-24.5655	0.2167	-113.36	0.006
K 18000	-1.2905	0.2167	-5.955	0.106
C 418	-0.6073	0.2167	-2.803	0.218
N 150	1.5773	0.2167	7.279	0.087
K*C 18000 418	4.3755	0.2167	20.191	0.032
K*N 18000 150	6.0175	0.2167	27.768	0.023
C*N 418 150	4.7126	0.2167	21.747	0.029
S = 0.6129 R-Sq = 99.9% R-Sq(adj) = 99.6%				

Where S is estimated stadard deviation of regression which shows average deviation error in the model and value is 0.6129. The R-Sq and R-Sq(adj) values represent the proportion of variation in the response data explained by the model. The "Goodness of the Fit" of regression model can be quantified from ANOVA by the R-Sq value of RC 99.9% while the remaining 0.1% is due to noise factors. R-sq indicates how well the model fits your data. Higher value of R-Sq result due to the presence of too many insignificant factors.

TABLE IV
RESPONSE FOR SN RATION

Level	K	C	N
1	-25.86	-25.17	-22.99
2	-23.28	-23.96	-26.14
Delta	2.58	1.21	3.15
Rank	2	3	1

TABLE V
ANALYSIS OF VARIANCE FOR SN RATIOS

Source	DF	Seq SS	Adj SS	Adj MS
K	1	13.323	13.32	13.323
C	1	2.951	2.951	2.951
N	1	19.903	19.9	19.903
K*C	1	153.16	153.2	153.16
K*N	1	289.68	289.7	289.681
C*N	1	177.67	177.7	177.67
Residual Error	1	0.376	0.376	0.376
Total	7	657.06		

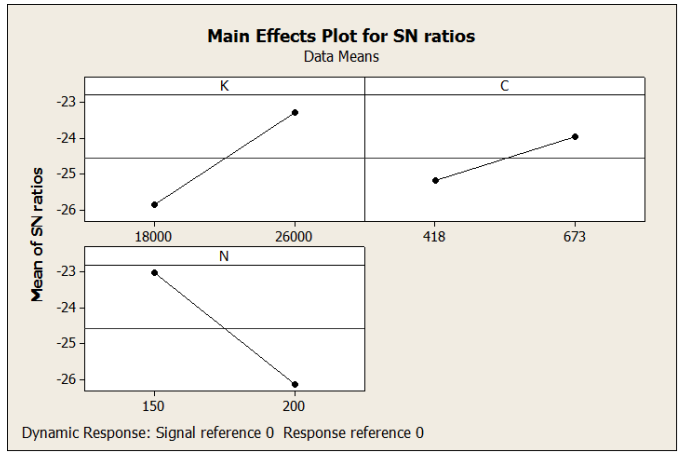

Fig. 6 Sensitivity analysis

A quarter car test rig has been used for optimizing ride comfort as per ISO 2631-1:1997 by varying three most influential parameters K, C and N within a prespecified range. Based on Taguchi method, runs are conducted and results obtained for the a model are with a high R-Sq value as high as 99.9% with R-Sq(adj) of 99.6% which indicates high level of reliability of the experimental model. This regression model can be used to determine the optimum settings and sensitivity analysis of all the parameters.

IV. ANALYSIS OF PARAMETERS ON SUSPENSION SYSTEM

In this study, Simulink model is validated with quarter car test rig with reference to correlation Index of 0.59. Validated SIMULINK model is used for further analysis with various input variables such as suspension K, C, M and N. Firstly analysis done to estimate the effect of each parameter K, C, M, N on RC and RH. Secondly the performance comparison of passive and semi-active suspension system is done for different road profiles and velocities. Following four cases of quarter car model are analyzed in MATLAB SIMULINK:

Case I] The input parameters used as sprung mass (M) 300kg, un-sprung mass (Mu) 45kg, suspension stiffness (K) 5700 N/m, tire stiffness (Kt) 10 times multiple of sprung stiffness (K), velocity (N) 40kmph [9] and damper coefficient is ranging from 500 N-s/m to 8000N-s/m[11]. Figure 7 shows the effect of variation of input parameters graphically.

Case II] The input parameters used as sprung mass (M) 300kg, un-sprung mass (Mu) 45kg, damper coefficient (C) 3500N-s/m, tire stiffness (Kt) 10 times multiple of sprung stiffness (K), velocity (N) 40kmph [11]. Suspension stiffness (K) ranging from 2.5 to 12kN/m [11]. Figure 8 shows the effect of variation of input parameters graphically

Case III] The input parameters used as un-sprung mass(Mu) 45kg, damper coefficient (C) 3500N-s/m, sprung stiffness (K) 10,000N/m, velocity (N) 40kmph [11]. Sprung mass (M) ranging from varies from 300 to 500 kg [11, 12]. Figure 9 shows the effect of variation of input parameters graphically.

Case IV] The input parameter used as sprung mass (M) 300kg, un-sprung mass (Mu) 45kg, damper coefficient (C) 3500N-s/m, suspension stiffness (K) 10,000N/m[11]. Velocity (N) ranging from 10 to 100kmph [9, 11, 12]. Figure 10 shows the effect of variation of input parameters graphically.

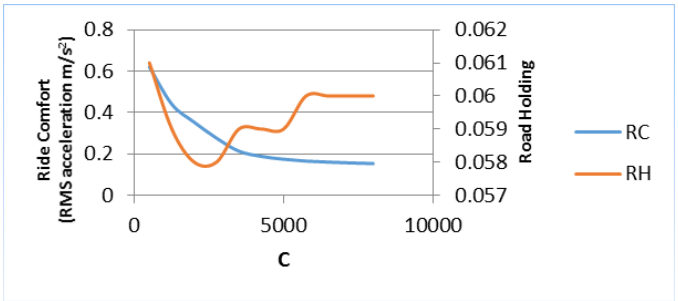

Fig. 7. Effect of C on road holding and ride comfort

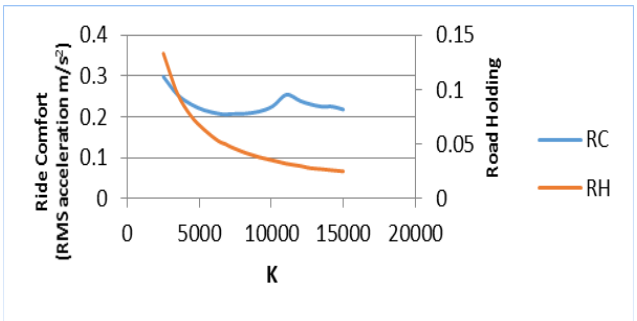

Fig. 8. Effect of K on road holding and ride comfort

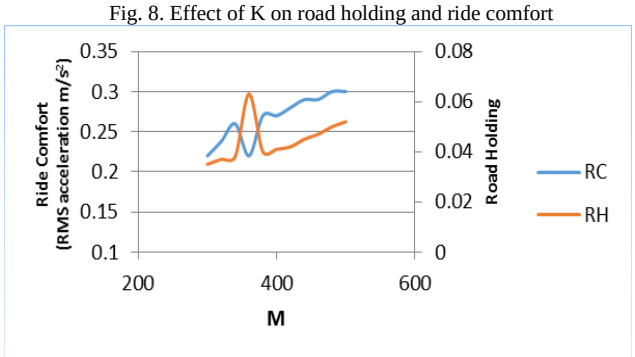

Fig. 9. Effect of M on road holding and ride comfort

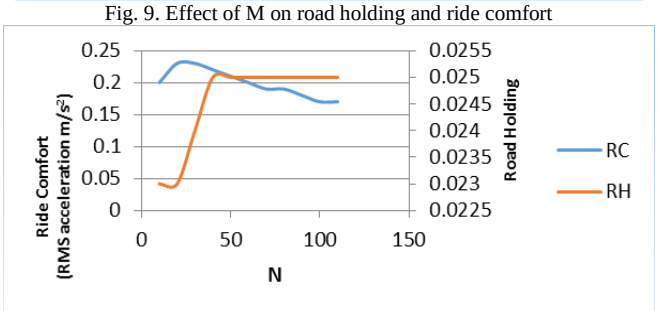

Fig. 10. Effect of N on road holding and ride comfort

A. *Quarter model of semi-active suspension system*

With the help of equation (1.1) and (1.2) self-tuned semi-active suspension model made in MATLAB SIMULINK which is shown in Fig. 11 and comparison model of passive and semi-active suspension system is shown in Fig. 12.

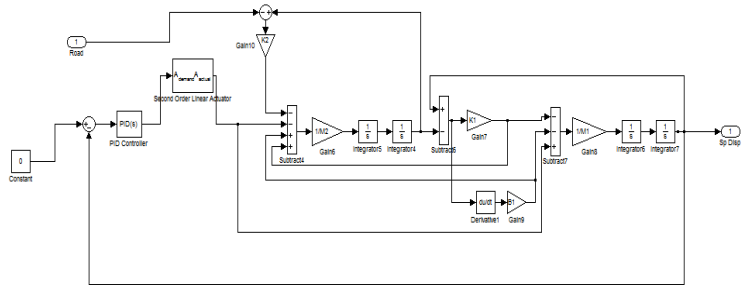

Fig. 11 Quarter car semi-active suspension model

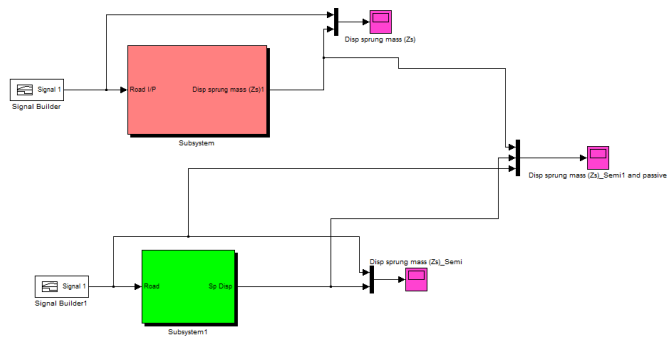

Fig. 12 Comparison Passive and Semi-active SIMULINK model

B. *PID controller*

PID basically works on principle of closed loop control system.

In which 'P' stands for proportional: In proportional control output signal will be obtained by multiplying the current error signal with gain (kp).

'I' stands for integral: the integral sign is the sum of all the instantaneous values that the signal has been from whenever you started counting until stop counting. When integral term adds to proportional term accelerates the movement of the process towards set-point and eliminates the residual steady state error that occurs with a proportional controller.

'D' stands for derivative: The derivative term slows the rate of change of the controller output and this effect is most noticeable close to the controller set-point. When all three are together it is called as PID controller. Fig. 13 shows PID controller used in semi-active suspension system.

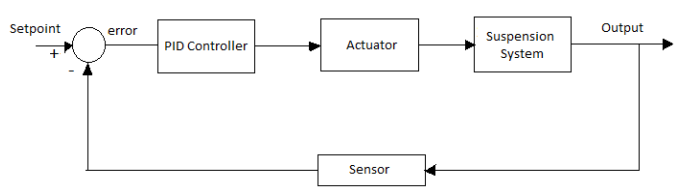

Fig. 13 Block diagram of suspension system using PID controller

As PID works on closed loop system principle process gets repeated till the output is reached equal to set-point.

Passive and semi-active suspension model are simulated by using input values shown in Table VI.

TABLE VI
INPUT VALUES FOR SIMULINK ANALYSIS

Parameters	Values
Sprung mass (M)	290 kg
Un-sprung Mass (Mu)	59 kg
Stiffness (K)	16182 N/m
Damper coefficient (C)	1000 Ns/m
Tire Stiffness (Kt)	190000 N/m
Width of Bump (A)	0.05 m
Height of bump (H)	0.1 m
Velocity (N)	20-80Kmph

Developed mathematical models for quarter car suspension are simulated in MATLAB SIMULINK software considering vehicle velocity range 20-80kmph for half sine bump with height of bump 0.1m. Fig. 14-19 show the graphs of sprung mass displacement for 20kmph, 40kmph, 60kmph and 80kmph. Further it is analyzed for different road profiles i.e. double sine bump, trapezoidal bump, step and square bump these road profiles are developed in MATLAB and used as road input.

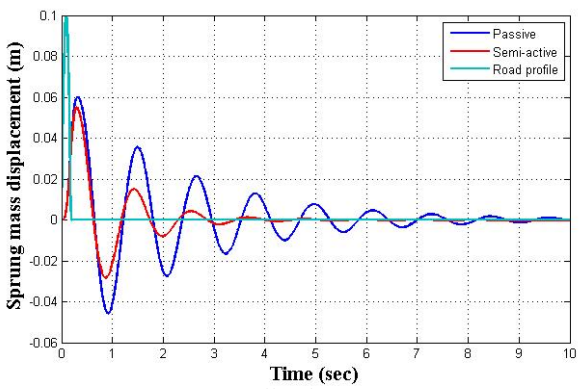

Fig. 14 Effect for 20 kmph velocity

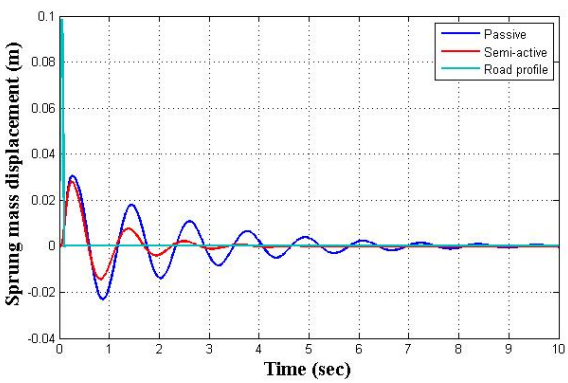

Fig. 15 Effect for 40 kmph velocity

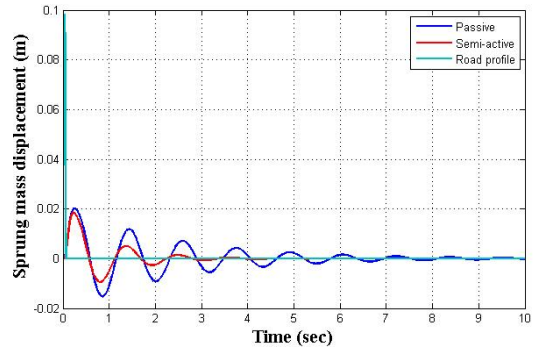

Fig. 17 Effect for 60kmph velocity

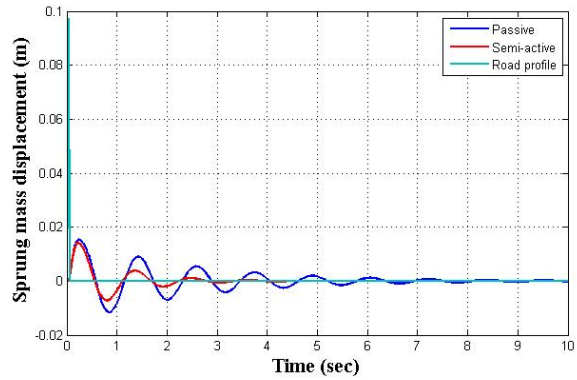

Fig. 19 Effect for 80kmph velocity

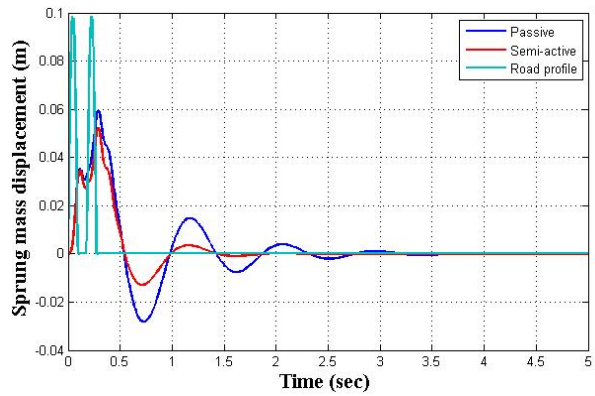

Fig. 20 For double sine wave bump displacement of sprung mass

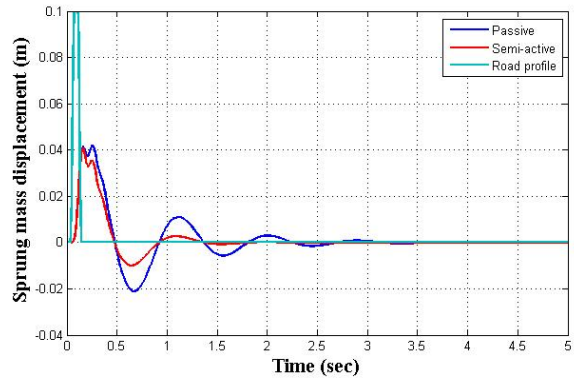

Fig.21 For trapezoidal bump displacement of sprung mass

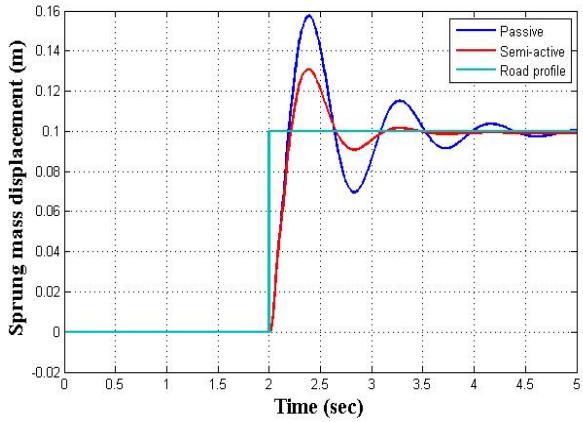

Fig. 22 For step bump displacement of sprung mass

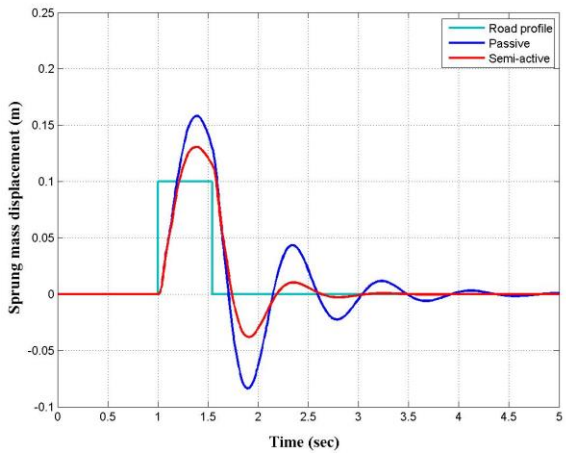

Fig. 23 For square bump displacement of sprung mass

V. RESULT AND DISCUSSION

In passive suspension system analysis of ride comfort and road holding with parameters K, C, M and N is one parameter is considered for. From graphs it is observed that as RMS acceleration value increases it gives decreasing ride comfort. Road holding is the relative displacement between road excitation and tire displacement (Zu-Zr). When (Zu-Zr) value is negatives means tire loses contact with road. Ride comfort varies inversely with road holding.

Case I] Fig. 7 shows a graph of effect of damping on ride comfort and road holding by increasing damping coefficient and keeping other parameters constant. It is observed that with increasing damping RMS acceleration decreases. Hence ride comfort goes on increasing. It reaches maximum ride comfort for damping coefficient at 6000 N-s/m. Road holding decrease initially then increases, as road holding is inverse with ride comfort maximum ride comfort is available at 6000N-s/m.

Case II] Fig. 8 shows a graph of effect of stiffness on ride comfort and road holding by increasing stiffness and keeping other parameters constant. It is observed that with increasing stiffness value ride comfort goes on increasing initially then decreasing at particular value and again it goes on increasing. Maximum ride comfort value is 7000 N-m and maximum road holding 15000N-m.

Case III] Fig. 9 shows a graph of effect of sprung mass on ride comfort and road holding by increasing sprung mass and keeping other parameters constant. It is observed that with increasing sprung mass ride comfort initially decreases then increases and decreases. Ride comfort is maximum at 370kg. Road holding is inversely with ride comfort hence maximum road holding observed at 370kg.

Case IV] Fig. 10 shows a graph of effect of damping on ride comfort and road holding by increasing velocity and keeping other parameters constant. It is observed that RMS acceleration value is below .315 m/s^2 it give fairly ride comfort within speed range 10 to 100 kmph. Initially at low speed ride comfort is less but after 45kmph it goes on increasing moderately. Road holding is very high at low speed and is maximum at 20kmph then goes on decreasing drastically but after 45 kmph increasing velocity does not have any effect for bump of height 0.1m. It is due to the fact that the suspensions are not getting sufficient time to react due to higher vehicle velocity.

From Fig. 14-19 it is concluded that change in velocity results into change in peak amplitude and settling time. Further performance analysis is carried out at 40 kmph for different road profiles i.e. double sine wave, trapezoidal, step and square bump with height of bump 0.1m by MATLAB/SIMULINK. The results in form of graph are as shown in Fig. 20-23.

It is observed that improved performance is obtained by semi-active suspension system with PID controller than passive suspension system. From Fig. 20 for double sine bump peak amplitude for passive and semi-active suspension system are 0.06, 0.054mm and settling time 3sec, 1.8 sec respectively. Similarly from Fig. 21 for trapezoidal bump peak amplitude for passive and semi-active suspension system are 0.042, 0.04mm and settling time 2.6 sec, 1.5 sec respectively. From Fig. 22 for step bump peak amplitude for passive and semi-active suspension system are 0.16, 0.11mm and settling time 4.3 sec, 2.8 sec respectively. From Fig. 23 for square bump peak amplitude for passive and semi-active suspension system are 0.155, 0.11mm and settling time 4.3 sec, 2.2 sec respectively. Peak amplitude of semi-active system is less than passive suspension system in all cases. Table VII shows percentage improvement in settling time of semi-active suspension system over the passive suspension system.

TABLE VII
SETTLING TIME IMPROVEMENT IN PERCENTAGE

Input profile	Settling time of Passive suspension	Settling time of semi-active suspension system	Percentage improvement (%)
Half sine bump	8.2	4	51
Double half sine bump	3	1.8	40

Trapezoidal bump	2.6	1.5	42.3
Step bump	4.3	2.8	34.9
Square bump	4.3	2.2	48.8

VI. CONCLUSIONS

The speed is very sensitive parameter as compared to damping coefficient and stiffness for ride comfort of passive suspension system as found from DOE analysis.

In passive suspension system both K and C values are constant hence it is difficult to achieve good RC and RH with low or high values of damping coefficient and stiffness. There is need to compromise between both K and C values to obtain good RC and RH. To overcome limitations of passive suspension performance of passive suspension system is compared with semi-active suspension system.

It is found that semi-active suspension system gives better performance due to use of PID controller. PID controller designed for good ride comfort, small amplitude value for suspension travel and reduced settling time to reach the steady state fast. Hence semi-active suspension gives better ride comfort as well as road holding under different road profiles and vehicle velocities than passive suspension. Percentage improvement in settling time for half sine bump, double sine bump, trapezoidal bump, step bump and square bump is 51%, 40%, 42.3%, 34.9% and 48.8% respectively.

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