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Parametric Optimization to Design a Passenger Car Suspension System for Better Dynamic Performance

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Abstract

Sprung mass, unsprung mass, tire material stiffness, spring and damping rate are the parameters to be optimized in order to design a light passenger car suspension system for optimum performance in the uncertain operating atmosphere. Passengers comfort, body stability, and road holding are the prime goals to be achieved as the measure of suspension system performance improvement. Full factorial design of experiments is intended at certain speed levels and at various design parameters. Vertical body acceleration root mean square values are obtained by simulation of 2-DOF nonlinear quarter car model at deliberated 64 run of experiments. For minimum RMS value of acceleration, an optimum parametric combination is identified among all the 64 parametric combinations specified by the design of experiments. The optimum set of parameters is further applied through the fabricated quarter car test setup to validate the design. In the fabrication of test setup, the nonlinear hysteretic behaviour foremost design parameters have been characterized through the theoretical and experimental analysis. Experimental results obtained from the test setup are closely related to simulation that justifies the viability of derived model for analysis. The proposed methodology ascertains the optimum design of the suspension system in a time and cost-effective manner. The resulting design exhibits the excellent vibrational characteristics andconvinces the ISO-2631-1997 suggested range of vehicle vibration.

*Keywords:*suspension system, design of experiments, full factorial design, quarter car, body vibrations

1. Introduction

Vibrations are the main cause of vehicle dynamic performance degradation in terms of driving comfort and vehicle body stability. The harmful effects ofprolonged vibrations can be observed as theserious health issues to the occupants(Kjellberg, 1990; Nagarkar et al., 2016). The International Organization for Standardization (ISO) 2631-1 has been characterized the passenger ride comfort based on the vehicle vibrations(*International Organization for Standardization*, 1997). A frequency range of 0.5 Hz to 80 Hz has been suggested to assess the potential effect of vibration on human body health(Cao et al., 2011). There is the direct association ofvehicle body acceleration to the ride quality, so that vehicle experiences the degradation in ride quality by rise in body acceleration(Sharma and Kumar, 2017b, 2017a). The vehicle also experiencesa major problem of the road holdingdue to the development of inconsistent vertical dynamic forces in the vehicle body. Therefore, the vehicle aided with the suspension system to wipe out the major source of vibrations (Yu et al., 2006).Suspension springs store the vibrational energy that further dissipates by the heating and viscous friction effect of the dampers (Chi and He, 2008; Sun et al., 2007).

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The improvement in dynamic characteristics of the suspension system is possible through the proper allocation of design parameters. As the matter of fact, a soft suspension system is very good to maintain the comfort of the passengers but at the same time, the adequate sum of normal forces fall short to provide the consistent ground-tire contacts and ultimately results in poor road holding of the vehicle (Salah et al., 2012). In the other side, stiffer or hard suspension system fails to provide good ride comfort but it exhibits good handling characteristics along with consistent vertical forces. So that the optimal trade-off amongst these conflicting requirements is the design prerequisite(Craft et al., 2003; Hyniova et al., 2009).

The comprehensive literature survey through the published articles and reports under the precincts of the proposed domain is being discussed here to find the literature gap and scope of the research and developments. The genetic algorithm approach has been pursued at a broad level in the literature to solve the vehicle suspension system control problems.Shirahatti (2008) takes GA at three objective functions- body acceleration, road holding, and shock to get the considerable improvements in vehicle performance(Shirahatti et al., 2008). In the literature, genetic algorithm approach has been applied on the quarter car, half car and the full car models with and without driver seat (Nagarkar et al., 2016; Tewari and Prasad, 1999). Parametric optimization has also been carried out for a nonlinear car model subjected to random road excitations(Verros et al., 2005). Tamboli has been applied the road estimated data in the analysis(Tamboli and Joshi, 1999) and the simulation study was made through the dynamic car model on ADAMS(Gobbi and Mastinu, 2001). Driving and ride performance has been optimized through the ADAMS model of 7 DOF full car suspension system and further most influencing parameters are carried out through the design of experiments (Christensen et al., 2000). The preceding literature reported numerous methods and significance of optimization in vehicle design. It is noted that an intelligent suspension system with an active damper and actuators are in trend over the past decade. In the active dampers, the fluid viscosity varies by the application of controlled current, while controlled actuator suspension system (active) is used to alter its behaviour by means of external control source of energy (Németh and Gáspár, 2011). It is also noted that an active or intelligent suspension system involves design complexity and high manufacturing as well as operational cost. In pure design concern, sprung mass, unsprung mass, spring-damping rate and suspension space parameters have been frequently used in the literature based in suspension system design and optimization (Baumal et al., 1998; Szőke et al., 2010).At the same time, vertical body acceleration and the tire normal force have been typically used as objective functions in the literature. Fig.1 undertakes the researchers' mind frame on choosing the effective design parameters based on reported literature.

The proposed work deals with the light passenger car suspension system design in order to improve the dynamic performance of the vehicle in uncertain operating conditions. The full factorial design approach has been applied to a2-degree of freedom quarter car model to obtain the optimum design parameters setting. The design of experiments outcome has been comprehended in the MINITAB® software.There are five substantial design parameters have been carried out to the analysis as described in fig.1.User's experience and the literature support helped to decide the search range of parametersin the analysis as mentioned in table 1 (Hemanth et al., 2017; Metered et al., 2015).

Table 1: Search range and parameters

Parametric values		*Factors with coded levels*		
Factors	***Value***	***Factors***	***Levels***	
			Min (-1)	Max (+1)
Sprung Mass (m_s) =	250 Kg	Sprung Mass –m_s (Kg)	250	300
Un-Sprung Mass (m_u) =	30 Kg	Un-Sprung Mass – m_u (Kg)	30	60
Spring Stiffness (k_s) =	14000 N/m	Spring Stiffness - k_s (N/m)	14000	18000
Damping coefficient (c_s)=	500 N/ms	Damping Coefficient – c_s (N/ms)	500	1000
Tire Stiffness (k_t) =	140000 N/m	Tire Stiffness - k_t (N/m)	140000	180000

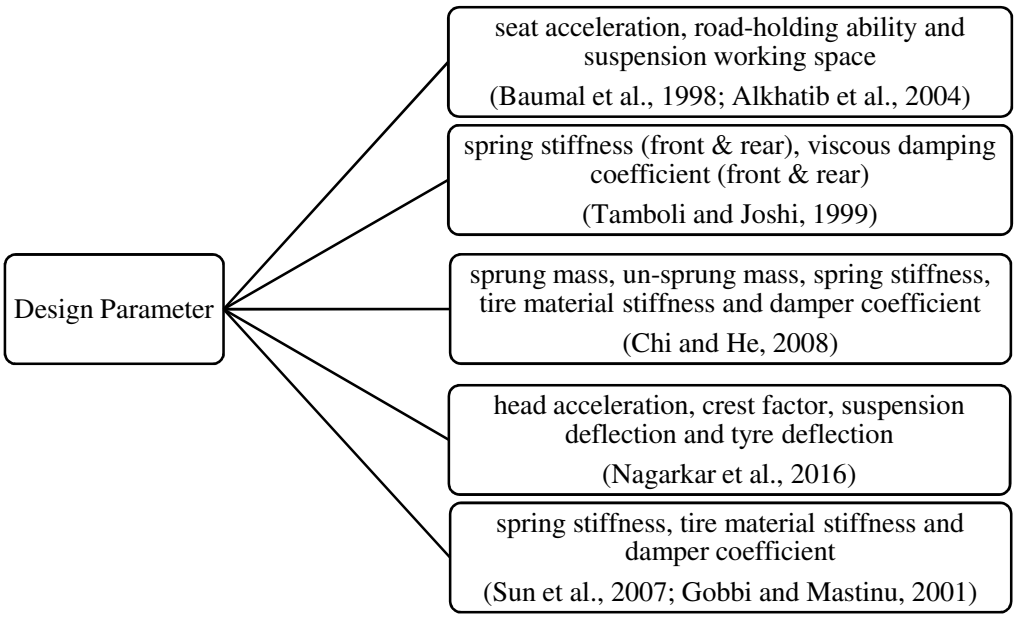

Figure 1: Parameters and objective functions identification

A nonlinear mathematical model of a quarter car has been derived with the help of basic principles and experimental study. The resulting model has been simulated on MATLAB Simulink to obtain the body acceleration, velocity and displacement data for various parametric combinations.The full modeling approach including the setupfabrication has been explained in section-2.The design of experiments methodology has been discussed in section 3 and the obtained results have been carried out in section 4 for detailed discussion. Finally, the work has been concluded in section 5after the observations and discussions.

2. Model formulation

The study carried out a two degree of freedom quarter car model subjected to a poor road terrain of class D. Fig.2(a). shows the proposed quarter car model that is comprised of sprung mass, unsprung mass, spring, damper, and tire.The dynamic behaviour of the car under the road excitations has been analysed by theoretical and experimental means. The experimentations on the fabricated car test setup have been conducted to examine the nonlinear characteristics of the system and finally fit into the mathematical model. The considered road profile characteristics are assimilated in a motor-driven wooden wheel.

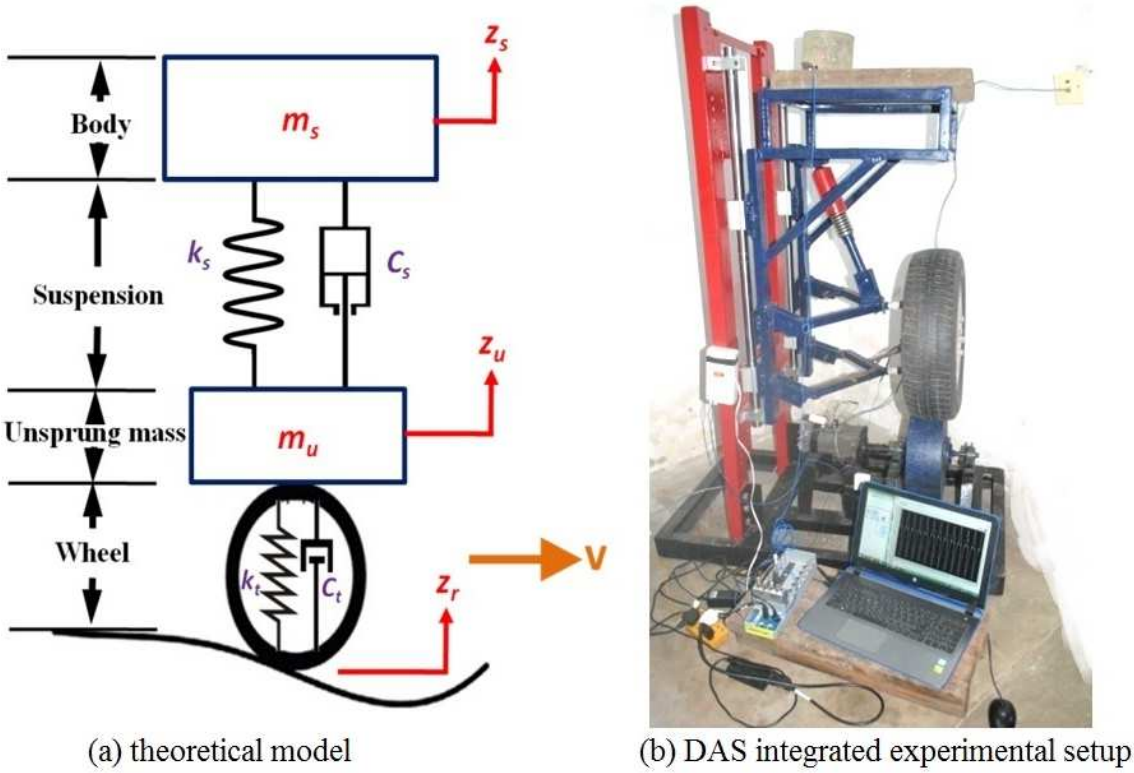

Figure 2: Quarter car testsetup

2.1 Analytical model

A single station suspension system including wheel and corresponding chassis part is carried out for modeling and simulation. The key modeling parameters are sprung mass (m_s), unsprung mass (m_u), spring stiffness (k_s), damping coefficient (c_s), tire spring constant (k_t) and tire damping coefficient (c_t). The resulting model possesses the two degree of freedom to the free vertical motion of sprung mass (z_s) and unsprung mass(z_u). A bad quality road is taken to produce the road inputs (z_r) in the model of the passive suspension system used in the simulation.

The governing equations of the proposed model are derived as 1-6 by balancing the vertical forces.Equations 3-6 described the force in the nonlinear spring, damper and the tire respectively.K_s and K_{sn} used in the equations are the linear and nonlinear terms for spring rate respectively; similarly, c_e and c_c are the damping coefficients for the extension and compression movements.

$$m_s\ddot{z}_s = F_{ss}(z_s, z_u, t) + F_{sd}(\dot{z}_s, \dot{z}_u, t) \tag{1}$$

$$m_u\ddot{z}_u = -F_{ss}(z_s, z_u, t) - F_{sd}(\dot{z}_s, \dot{z}_u, t) + F_{ts}(z_u, z_r, t) + F_{td}(\dot{z}_u, \dot{z}_r, t) \tag{2}$$

$$F_{ss}(z_s, z_u, t) = k_s(z_u - z_s) + k_{sn}(z_u - z_s)^3 \tag{3}$$

$$F_{sd}(\dot{z}_s, \dot{z}_u, t) = \begin{cases} c_e(\dot{z}_u - \dot{z}_s), \\ c_c(\dot{z}_u - \dot{z}_s), \end{cases} \tag{4}$$

$$F_{ts}(z_u, z_r, t) = k_t(z_r - z_u) \tag{5}$$

$$F_{td}(\dot{z}_u, \dot{z}_r, t) = c_t(\dot{z}_r - \dot{z}_u) \tag{6}$$

The road profile is characterized by power spectral density (PSD) function, which classifications have been presented in ISO-8608 (*International Organization for*

Standardization, 1995). Power spectral density shows the characteristic drop in magnitude with wave number, the PSD function in frequency and time domains are described in equation (7) and (8) respectively.
Road displacement PSD,

$$\Phi(\Omega) = \Phi(\Omega_0)\left(\frac{\Omega}{\Omega_0}\right)^{-w} \tag{7}$$

$$\Phi(n) = \Phi(n_0)\left(\frac{n}{n_0}\right)^{-w} \tag{8}$$

Where $\Omega\left(=\frac{2\pi}{\lambda}\right)$is the angular spatial frequency in rad/m, λ denotes the wavelength$\Phi_0\left(\triangleq \Phi(\Omega_0)\right)$ that describes the power spectral density in m^2/(rad/m) at reference wave number $\Omega_0 = 1.0\,\mathrm{rad/m}$. Here $n = \left(\frac{\Omega}{2\pi}\right)$ is the spatial frequency at a reference spatial frequency n_0=0.1 cycle/m and w is the waviness that is as a linear fitting coefficient, generally for road surface w=2. The road surface excitation is characterized through the Gaussian filter in equation 9, where w_0 denotes Gaussian white noise with a PSD of 1.

$$\dot{Z}_r = -2\pi f_0 Z_r + 2\pi\sqrt{\Phi_0 v w_0} \tag{9}$$

The applied random road profile input excitations are displayed in fig.3, the car is considered to be run at the velocity (v) of 45km/hr at the very poor road of class D with the associated degree of roughness (*International Organization for Standardization*, 1995; Tyan et al., 2009).

$$\Phi(n_0) = 1024 * 10^{-6} m^2/(\mathrm{m/cycle})$$

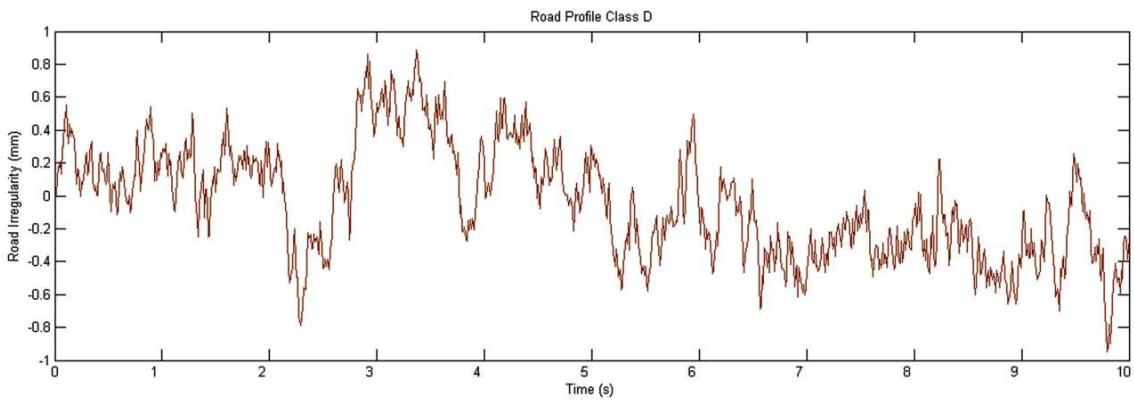

Figure 3:**I**nput excitation for rough road of Class D

2.2 Experimental setup

The fabricated test setup shown in fig.2(b) is integrated with acceleration, speed and displacement sensors supported by NI-data acquisition system. The test setup incorporated with speed regulation mechanism and it has the provision to change spring, damper and sprung mass. A force-displacement test has been carried out on the spring which characteristics are assimilated in fig. 4(a). The whole system behaviour is represented through the characteristic plots in figure 4(b). The closed correlation in the model and physical setup behaviours confirms the validity of the model.

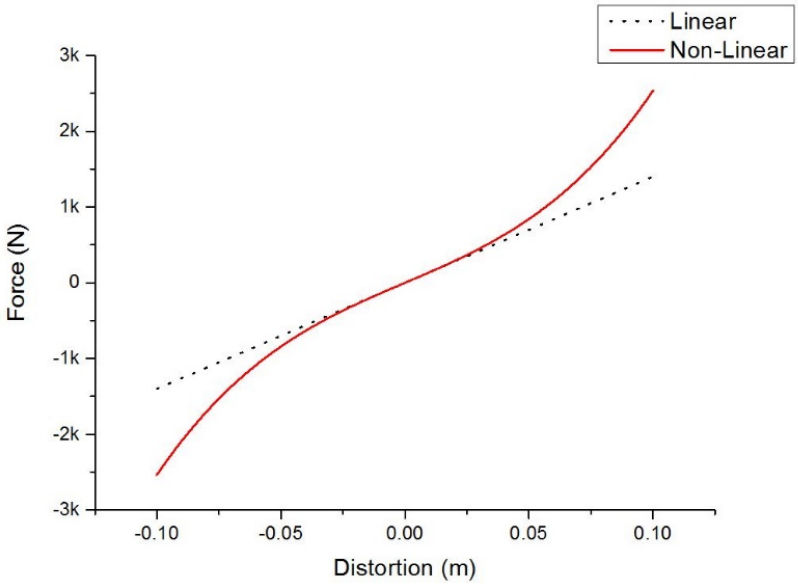

Figure 4(a): Linear and nonlinear spring's behaviour

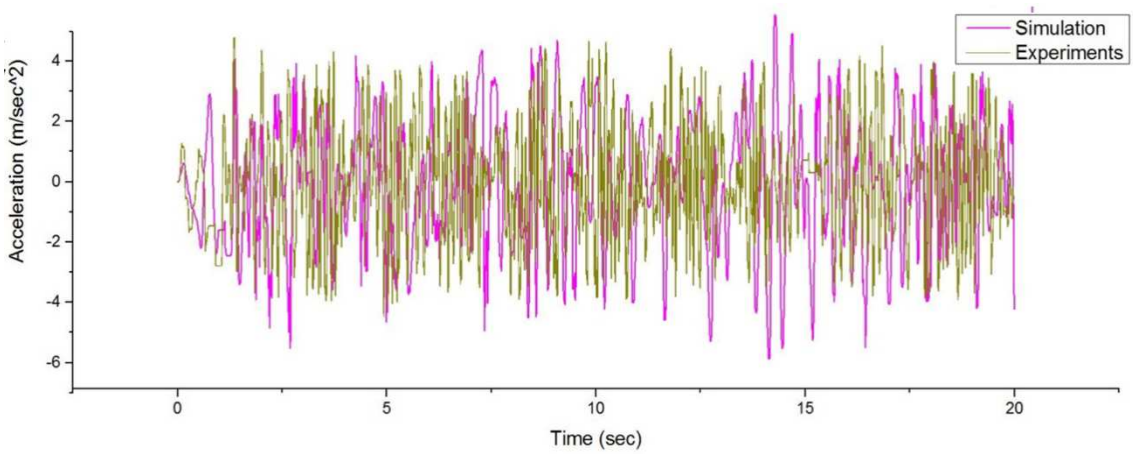

Figure 4(b): Body acceleration in time domain

3. Design of experiments

DOE carry out the multiple factors together that gives the simultaneous reduction in time, cost and amount of computations. In DOE, the significant factors and their interactions are revealed across all the possible combinations in the domain. Table 2 has the total 64 combinations of input parameters, which are obtained by 2^5 full factorial design with two replicates. Vehicle velocity is taken as the block in DOE that helps to prevent any block effect or effect due to experimental factors. Vehicle velocity and the road surface excitations are taken from section 2.1 and section 2.2. All the set of experiments are being applied on a validated nonlinear quarter car model using Matlab Simulink platform. The root mean square data of vertical body acceleration are obtained in the range of 0.1125 m/s^2 to 1.2436 m/s^2 from the experiments correspond to each possible combination of parameters. The lowest value of acceleration is itself lying over the acceptable limit suggested by ISO-2631-1997 (*International Organization for Standardization*, 1997). Therefore the significant role of the desired parameters and their interactions is explored in regression analysis to get the best-fitted design through graphical as well as analytical methods.

Table 2:Experimental Runs

Run Order	M_s (Kg)	M_u (Kg)	K_s (N/m)	K_t (N/m)	C_s(N/s-m)	RMS acceleration (m/s^2) Experimental	RMS acceleration (m/s^2) Predicted
1	300	30	18000	180000	500	0.3083	0.3193
2	250	30	18000	140000	1000	0.5514	0.5732
3	250	30	14000	140000	1000	0.2434	0.2609
4	250	60	14000	180000	1000	0.4194	0.3975
5	300	30	18000	140000	500	0.2835	0.3015
6	300	30	14000	180000	1000	0.5308	0.5324
7	250	30	18000	140000	500	0.6258	0.6368
8	250	60	18000	140000	500	0.7639	0.7193
9	250	30	18000	180000	1000	0.8352	0.8239
10	300	60	14000	140000	1000	0.4997	0.4960
11	300	60	14000	140000	500	0.1125	0.1046
12	250	60	14000	140000	1000	0.9888	0.1015
13	250	30	14000	140000	500	0.6906	0.6863
14	250	60	14000	180000	500	0.3838	0.3760
15	250	60	14000	140000	500	0.5904	0.5902
16	300	60	18000	140000	1000	0.2200	0.2180
17	250	60	18000	140000	1000	0.1351	0.1349
18	300	30	14000	180000	500	0.2345	0.2243
19	300	30	18000	140000	1000	0.5514	0.5598
20	300	60	18000	180000	500	0.6675	0.6675
21	300	60	14000	180000	500	0.8800	0.8759
22	300	30	14000	140000	500	1.1252	1.1244
23	300	60	18000	140000	500	0.5314	0.5244
24	300	30	18000	180000	1000	0.4742	0.4518
25	250	30	14000	180000	1000	0.1756	0.1783
26	300	60	18000	180000	1000	0.8547	0.8561
27	300	60	14000	180000	1000	0.4281	0.4091
28	300	30	14000	140000	1000	0.3094	0.3015
29	250	30	18000	180000	500	0.4638	0.4550
30	250	30	14000	180000	500	0.7565	0.7719
31	250	60	18000	180000	500	0.5816	0.5877
32	250	60	18000	180000	1000	0.3611	0.3704
33	300	60	14000	140000	1000	0.4491	0.4451
34	250	60	18000	140000	500	0.7314	0.7323
35	300	60	14000	180000	500	1.2436	1.2572
36	250	30	18000	180000	500	0.5927	0.5961
37	250	60	14000	180000	1000	0.4985	0.4990
38	300	30	14000	140000	500	0.9659	0.9864
39	300	60	14000	140000	500	0.2023	0.2124
40	300	30	14000	140000	1000	0.4082	0.4168
41	250	30	18000	140000	500	0.6668	0.6764
42	250	60	18000	180000	1000	0.4714	0.4831
43	300	30	18000	180000	1000	0.5873	0.5972
44	300	30	18000	180000	500	0.3925	0.3972
45	300	60	18000	180000	1000	0.6152	0.6258
46	250	30	18000	180000	1000	0.7886	0.7985
47	300	60	18000	180000	500	0.8282	0.8329
48	300	60	14000	180000	1000	0.2606	0.2538
49	300	30	14000	180000	500	0.4442	0.4170
50	300	30	14000	180000	1000	0.5722	0.5838
51	250	30	18000	140000	1000	0.4552	0.4544
52	250	60	14000	140000	1000	0.8947	0.8959
53	250	60	14000	140000	500	0.5271	0.5196
54	300	30	18000	140000	1000	0.7251	0.7074

55	250	60	18000	180000	500	0.6482	0.6918
56	300	60	18000	140000	500	0.3233	0.3237
57	250	30	14000	180000	500	0.8352	0.8478
58	250	30	14000	180000	1000	0.3917	0.3748
59	250	30	14000	140000	500	0.6763	0.6946
60	250	60	14000	180000	500	0.3037	0.3094
61	300	60	18000	140000	1000	0.2193	0.2195
62	250	30	14000	140000	1000	0.2995	0.3263
63	300	30	18000	140000	500	0.3237	0.3444
64	250	60	18000	140000	1000	0.2331	0.2124

The quantitative analysis is made through the average effect observations. The obtained P-value in the table 3 described the statistically significant relationship of factors and their interactions.

Table 3: Average effect

Term	Effect	Coef	SE Coef	T-Value	P-Value	Significance
Constant		0.5391	0.119	45.28	0.000	Yes
Ms	-0.0318	-0.0158	0.019	-1.35	0.189	No
Mu	-0.0131	-0.0067	0.019	-0.55	0.583	No
Ks	-0.0167	-0.0084	0.019	-0.70	0.487	No
Kt	0.0475	0.0238	0.019	2.00	0.054	No
Cs	-0.1028	-0.0514	0.019	-4.32	0.000	Yes
Ms*Mu	0.0194	0.0097	0.019	0.82	0.420	No
Ms*Ks	-0.0313	-0.0156	0.019	-1.31	0.197	No
Ms*Kt	0.0833	0.0416	0.019	3.50	0.001	Yes
Ms*Cs	0.0294	0.0147	0.019	1.24	0.224	No
Mu*Ks	-0.0146	-0.0073	0.019	-0.60	0.544	No
Mu*Kt	0.0802	0.0401	0.019	3.37	0.002	Yes
Mu*Cs	-0.0091	-0.0045	0.019	-0.38	0.708	No
Ks*Kt	0.0870	0.0435	0.019	3.65	0.001	Yes
Ks*Cs	0.0615	0.0306	0.019	2.58	0.014	Yes
Kt*Cs	0.0207	0.0104	0.019	0.87	0.390	No
Ms*Mu*Ks	0.0858	0.0429	0.019	3.61	0.001	Yes
Ms*Mu*Kt	0.1955	0.0978	0.019	8.22	0.000	Yes
Ms*Mu*Cs	-0.0746	-0.0373	0.019	-3.13	0.004	Yes
Ms*Ks*Kt	-0.0221	-0.0110	0.019	-0.92	0.362	No
Ms*Ks*Cs	0.0862	0.0431	0.019	3.62	0.001	Yes
Ms*Kt*Cs	-0.0327	-0.0164	0.019	-1.38	0.178	No
Mu*Ks*Kt	0.0215	0.0107	0.019	.90	0.374	No
Mu*Ks*Cs	-0.1979	-0.0991	0.019	-8.32	0.000	Yes
Mu*Kt*Cs	-0.1145	-0.0573	0.019	-4.81	0.001	Yes
Ks*Kt*Cs	0.0843	0.0421	0.019	3.54	0.001	Yes
Ms* Mu* Ks*Kt	-0.0709	-0.0354	0.019	-2.98	0.005	Yes
Ms* Mu* Ks* Cs	0.0957	0.0478	0.019	4.02	0.000	Yes
Ms* Mu* Kt* Cs	-0.0855	-0.0428	0.019	-3.59	0.001	Yes
Ms* Ks* Kt* Cs	-0.0621	-0.0310	0.019	-2.61	0.014	Yes
Mu* Ks* Kt* Cs	0.1508	0.0755	0.019	6.33	0.000	Yes
Ms* Mu* Ks* Kt*Cs	0.3174	0.0687	0.019	5.77	0.000	Yes

In qualitative analysis the extent of parameters influence over response is perceived through the normal probability plots in fig.5. The identification of the parameters is carried out through the color distinctions. Small red squares show the significant factors while blue dots describe the non-significant factors in the domain.

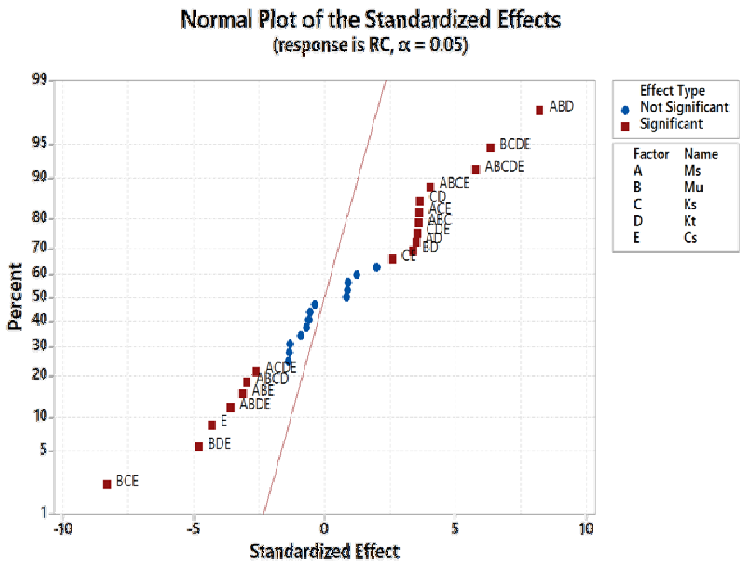

Figure 5: Normalplot

In table 4 high coefficient of determination (R-Sq) signifies the 92.65% of the response can be rendered by the input factors with 8.35% noise error and adjusted R-Sq value approves the legitimacy of the model. A higher value of R-Sq(pred) is used to predict the model for future observations and prevent over fitting.

Table 4: Regression Statistics

S	Standard Error of Regression	0.0951978
R-Sq	Coefficient of Determination	92.65%
R-Sq(adj)	Adjusted Coefficient of Determination	90.51%
R-Sq(pred)	Predicted Coefficient of Determination	89.55%

The residual plots in fig.6 graphically elucidate the goodness of fit through the deviation in obtained and predicted data. The symmetrical scattering of residuals above and below the mean residual zero lines in versus fits verify the homoscedasticity property. The histogram follows a bell-shaped pattern of the normally distributed area with a little skewness due to the presence of outliers in the extreme left and right. Randomization of experimental data verified by versus order.

ANOVA table 5 demonstrate the sources of influence. There are two main properties one is the sequential sum of squares (SEQ SS) and the other is adjusted sum of squares (AJD SS) that shows the variance of previous to all factors presented in the model. The similarity in SEQ SS and ADJ SS proves the orthogonality of design.

Table 5: ANOVA

Source	DF	SEQ SS	ADJ SS	ADJ MS	F-Value	P-Value
Model	31	3.66236	3.66236	0.118140	13.04	0.000
Main Effects	5	0.22859	0.22859	0.045718	5.04	0.002
2-Way Interactions	10	0.44256	0.44256	0.044256	4.88	0.000
3-Way Interactions	10	1.91991	1.91991	0.191991	21.18	0.000
4-Way Interactions	5	0.76916	0.76916	0.153832	16.97	0.000
5-Way Interactions	1	0.30213	0.30213	0.302129	33.34	0.000
Residual Error	32	0.29000	0.29000	0.009063		
Total	63	3.95236				

The obtained regression fit model is given below-
RC=-551.6 + 2.186Ms + 10.80Mu + 3.19E-2Ks + 3.616E-3Kt + 0.603Cs - 4.358E-2Ms*Mu - 1.26E -4Ms*Ks - 1.4E-5Ms*Kt - 2.423E-4Ms*Cs - 6.05E-4Mu*Ks - 7.3E-5Mu*Kt - 1.105E-2Mu*Cs - 3E-6Ks*Kt - 3.6E-5Ks*Cs - 4E-6Kt*Cs + 2E-6Ms*Mu*Ks + 2.3E-5Ms*Mu*Kt + 4.6E-5Ms*Mu*Cs + 7.8E - 5Ms*Ks*Kt + 9E-7Ms*Ks*Cs + 3.4E-5Ms*Kt*Cs + 8.9E-5Mu*Ks*Kt + 1.1E-5Mu*Ks*Cs + 1.2E - 5Mu*Kt*Cs + 9E-5Ks*Kt*Cs - 5E-6Ms*Mu*Ks*Kt - 7E-6Ms*Mu*Ks*Cs - 2.2E - 5Ms*Mu*Kt*Cs - 4.3E - 5Ms*Ks*Kt*Cs - 6.7E-5Mu*Ks*Kt*Cs + 4.1E-5Ms*Mu*Ks*Kt*Cs

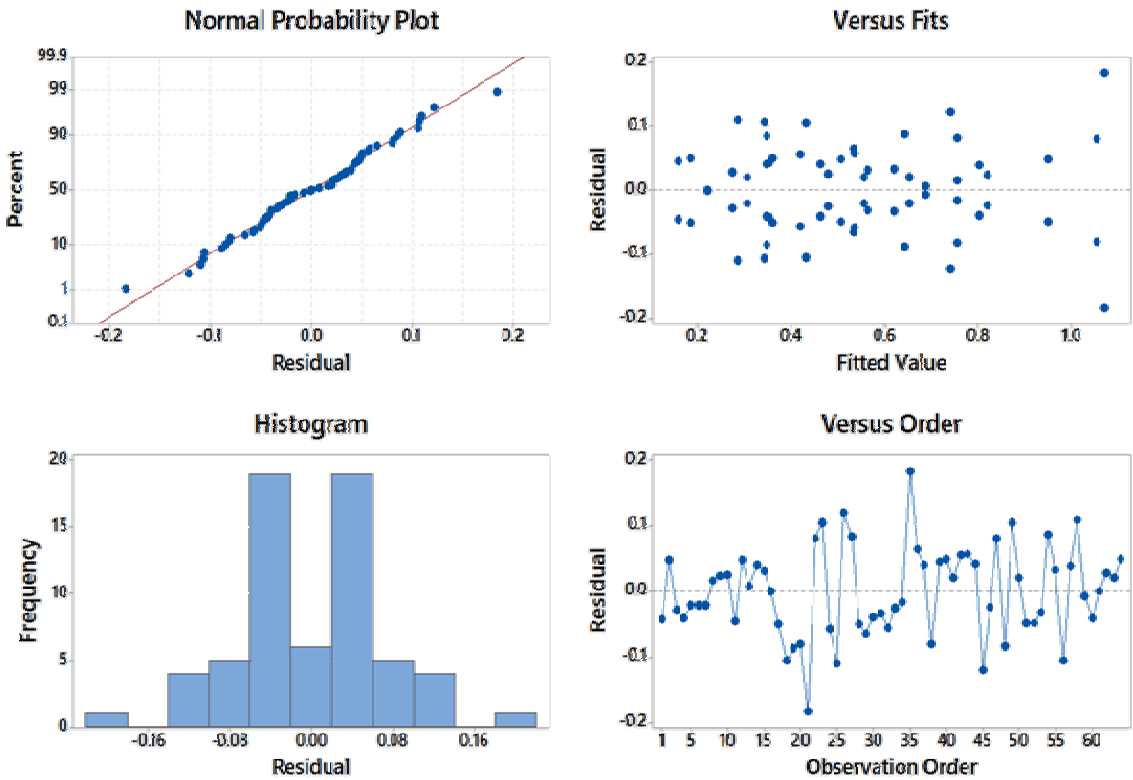

Figure 6: Residual plots

4. Result and discussion

The parameters response sensitivity examination is carried out through the individual factors and their interactions. The parameters k_t and c_s have been identified the most sensitive parameters in the domain, fig.7 revealed degree of sensitivity of parameters through its slops. The parameter interaction terms m_u*k_s, m_s*k_s and k_s*k_t are found highly sensitive. The two dimensional plots in fig.8 shows the parameters sensitivity by color darkness; more elevation of c_s, m_s, k_t represent the high sensitivity towards the response.

The analysis has been carried out with the objective of minimization of RMS acceleration. The user selected upper extreme response value and target values are 0.5m/s^2 and 0.315m/s^2 respectively. The optimum parametric setting is revealed in fig.9, minimum RC on the optimum settings of parameters can be 0.1589 m/s^2. The RMS acceleration obtained from the test rig experiments for the optimized design is 0.1125 m/s^2. The obtained design has also been fitted in the Simulink model and tested at the

optimum values of parameters, then the RC obtained was 0.1967 m/s^2. The level of accuracy obtained for RMS acceleration on physical test is 58.76% and the accuracy in simulation is observed as 76.22%.

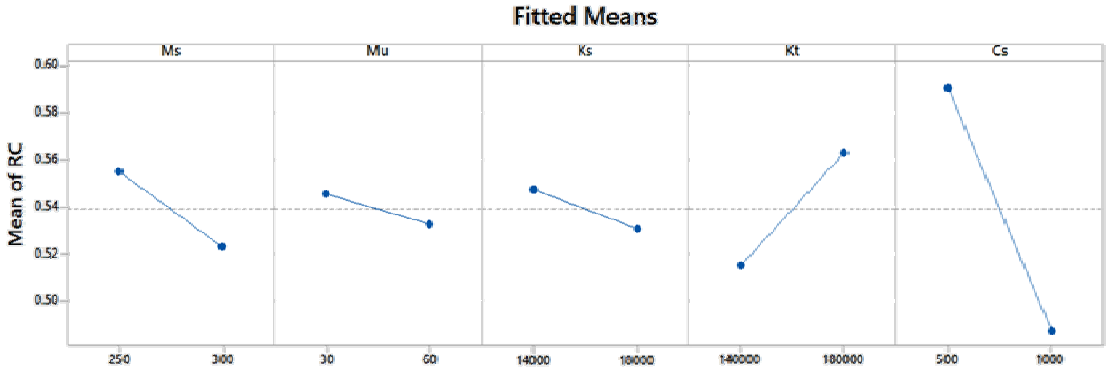

Figure 7: Main effects plot

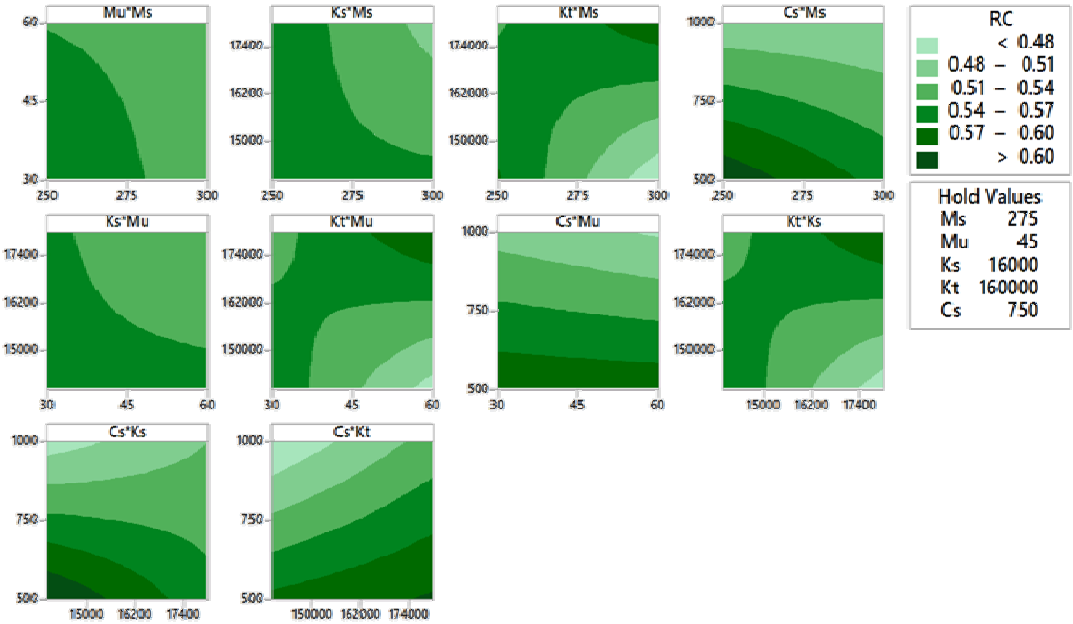

Figure 8: Ride comfort contour plots

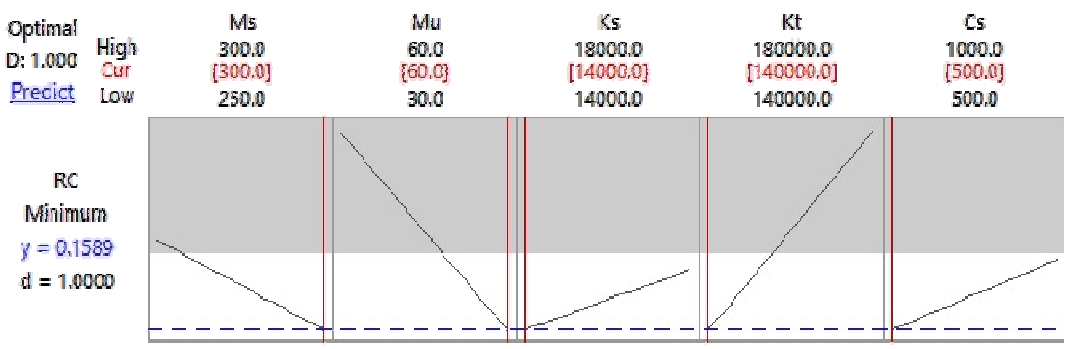

Figure 9: Plot of optimization

5. Conclusion

A quarter car suspension system model subjected to random road inputs has been studied through dynamic behaviour analysis and optimization.The vehicle structure has been optimized for five design parameters to achieve the desired range of acceleration. Linear and nonlinear characteristics of the components have been examined through the experimental data obtained from the fabricated quarter car test rig. Full factorial design of experiments methodology explored through 64 set of experiments and all the experiments have been simulated in MATLAB which save the huge experimental cost and time. With respect to the desirable value of RMS acceleration 0.1589 m/s^2, the accuracy in ride comfort measurement at the physical setup obtained 58.46%on the optimum design values that is increased to 76.22% in simulations. The experimental model exhibited a consistent result of R-Sq-92.65%, R-Sq(adj)-90.51%, and R-Sq(pred)-89.55% which explains the variability, reliability, and predictability of the model. The quantified results in terms of R-Sq, R-Sq(adj), and R-Sq(pred) can be improved by considering various other parameters of suspension geometry namely toe, camber, caster, tire pressure, etc. To make the experimental process more economical number of runs can be reduced by considering the fraction factorial method of DOE where main effects are only confounded with 4-way interactions and higher.

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