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Optimization of Passive Suspension System for Enhancement of Ride Comfort

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Abstract : *Vehicle passive suspension systems require an investigation to determine the spring, mass and damper settings that assure optimal Ride Comfort (RC). In this paper, Ride comfort has been maximized by optimizing various suspension parameters using a quarter-car test rig and implementing full factorial methodology of Design of Experiments (DOE). As per ISO 2631-1:1997 standard, the ride comfort has been measured in terms of the R.M.S acceleration of the sprung mass which must be below 0.5 m/s2 for a comfortable ride. Here, a quarter-car test rig, equipped with NI-LabVIEW data acquisition system, has been developed and incorporated with provisions to vary the factors within their predetermined range. Then the DOE methodology was implemented to formulate a single-objective model of Ride Comfort using MINITAB. The generated model possesses R-Sq value 87.42% which signify that regression model are fitted with the data. The response optimizer methodology has been implemented to arrive at the optimized set of variables, which results in RC value up to 0.4 m/s2. Concurrently, a quarter car simulation model is also developed in MATLAB-SIMULINK and the experimental model is verified. Both the models are found to exhibit a very high correlation index (CI).*

Keywords - *Design of Experiments, Quarter Car Test Rig, Full Factorial Method, Suspension Optimization, ANOVA*

I. INTRODUCTION

Suspension system refers to the interface between the vehicle chassis and the wheel assembly, which provides the scope for compliance to the vehicular body while traversing over variegated profiles. Due to simplicity and low cost, most of the passenger vehicles use passive suspension [1]. Passive suspension is linear in nature and generally involves a parallel mounting of a spring and damper. The spring is chosen based solely on the weight of the vehicle, while the damper defines the suspension's placement on the compromise curve. Depending on the type of vehicle, a damper is chosen to render maximum vibration absorption as per the transit environment [2]. The forces acting on the car body result from relative motion and relative velocity of the suspension [3]. The limitation in using standard elements is that, the suspension can only be designed for narrow range of performance. For example, suspension systems of sports cars, which are tuned for increased handling are very stiff and do not provide a plush ride quality, while a luxury passenger car designed for minimal force transmission sacrifices high-speed stability to attain this effect. Mohamed Bouazara et al. [4], in one of his works have provided comparison graphs for active, semi active and passive suspension in terms of ride comfort.

Automobiles travel at a high speed over varying road profiles, bearing different static loads and as a consequence experience a broad spectrum of vibrations. As per SAE J670 terminology, the term RIDE is defined as the vibrations of low frequency (up to 5Hz) as experienced by the sprung mass as a rigid body [5]. The lower frequency ride vibrations are manifestations of Dynamic behavior.

ISO 2631-1:1997 standards imply that humans are most sensitive to vertical vibrations in the range of 4 to 8 Hz. Katu U.S et al. [6] have stated that exposure for a period of more than 30 to 35 minutes on a rough road will make the driver feel uncomfortable. Mitsuhiko et al. [7] conducted a survey of 284 male taxi drivers and found that 45.8% drivers suffered from Low Back Pain (LBP). Whole body vibrations (WBV) on driver seats of 12 taxis under actual dynamic conditions were evaluated and it was found that the majority of the weighted R.M.S. accelerations of the taxis were unacceptable as per ISO 2631-1:1997 [8]. Sawant et al. [9] have mentioned that vehicle drivers are exposed to ride vibrations for 8 to 10 hrs per day. A.M. Darby et al. [10] have postulated that time equivalent to 4.9 million working days was lost in 2003 /2004 due to musculoskeletal disorders.

A study of the subject of ride comfort indicates that it is highly susceptible to subjective perceptions. However, several approaches have been proposed for the objective treatment of ride comfort. Janeway's [11] recommendations for automobile and railroad practice segregate the frequency sensation thresholds into 3 domains: 'strongly noticeable', 'uncomfortable' and 'very uncomfortable'. The Goldman [12] presentation handles the subjective responses of the human body to vibratory motion. It evaluates frequencies wherein subjects 1) perceive vibrations 2) find it unpleasant or 3) refuse to tolerate it. The data is relevant to a body on a vibrating support exposed to vertical oscillation for a short duration. For evaluating and standardizing ride comfort, the ISO 2631 standards 1997) [8] is used in Europe, the BS 6841 (1987) [13] in UK and VDI 2057 [14] in Germany. The determination of ride comfort, i.e. the vibrations transferred to the human occupant at the vehicle seat person interface is measured and a comfort rating is determined based on the above standards.

S. Kilian et al. [15] had worked on the optimization of torsion, bending and swaying of suspension designs by using finite element methods like topology optimization and topography optimization in Altair Opti-Struct software to maximize ride comfort during the design stage. In a similar work, a two- dimensional 8 Degree of Freedom (DOF) model was developed to simulate and animate the response of a vehicle to different road, traction, braking and wind conditions in a 3D VRML environment [16]. A model validation was conducted by comparing time taken to attain 100 kmph from standstill against a Honda Accord car equipped with an accelerometer and an engine rpm recorder.

Joao P. C et al. [17] have proposed a methodology for optimization of ride and stability of a vehicle based on the use of flexible multi-body model. The ride optimization is achieved by finding the optimum of ride index by measuring acceleration in several key points of the vehicle. Time histories of accelerations, velocities and displacement at the center of gravity, have been considered by A.F. Naude and J.A. Snyman [18,19] along with the time histories of forces, deflections and deflection rate of wheels and suspension components.

A 7 DOF full car model has been developed and optimum ride comfort has been achieved by trying out different spring damper setting using DOE by Mostani Saeed et al. [20]. Road surface has been simulated using power spectral density (PSD) it was found that car spring stiffness is most sensitive. RMS acceleration and pitch angle for optimum setting at different speeds was generated and only ride comfort has been optimized. Javad Marzbanrad et al. [21] performed optimization of passive suspension system on a 7 DOF model in MATLAB using DOE for speeds ranging from 60kmph to 90kmph.

In this paper, a quarter car test rig has been developed and equipped with NI-LabVIEW data acquisition system. A half sine wave bump-profile has been actuated via a cam to study and evaluate the suspension behavior. Provisions to vary sprung mass (M), spring stiffness (K) and damping factor (C) have been incorporated in the test rig. As per ISO 2631-1:1997 standards, the RC were measured in terms of the R.M.S acceleration of the sprung mass. The DOE methodology was implemented to formulate a single-objective model with RC as the response and to optimize it for maximum comfort.

II. QUARTER CAR TEST RIG

To evaluate the influences of various factors over the ride comfort, a quarter car test rig as shown in Fig. 1 was designed and fabricated with provisions to vary sprung mass (M), spring stiffness (K) and damping factor (C) and Speed (N) discretely and was integrated with NI- LabVIEW for data assimilation.

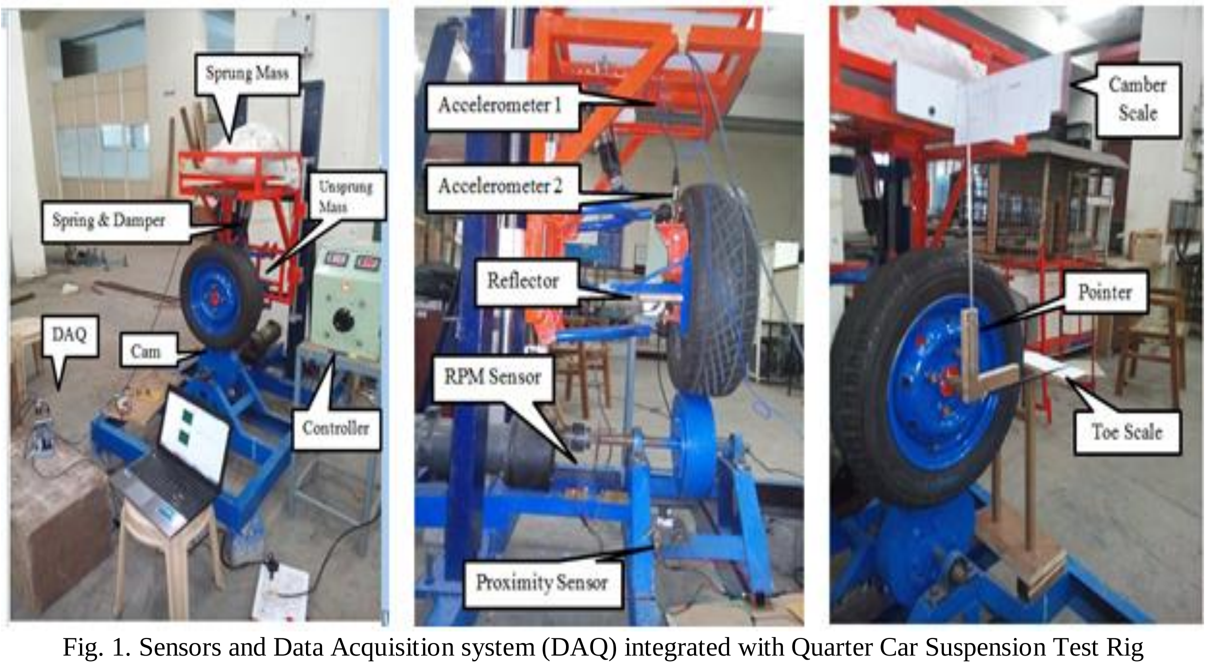

Fig. 1. Sensors and Data Acquisition system (DAQ) integrated with Quarter Car Suspension Test Rig

III. FULL FACTORIAL DESIGN

Since the requirement is to minimize R.M.S. acceleration for maximum RC, it is necessary to determine set of parameter that fulfills our requirement along with the sampling range, within which experimental runs will be conducted, as shown in Table 1 and Table 2.

TABLE 1. Full Factorial Design

Factors	3
Base design	3,8
Runs	16
Replicates	2
Blocks	2

TABLE 2. 2-Level Determination of Factors

Factors	Low(-1)	High(+1)
Spring Stiffness (K), N/mm	18000	26000
Damping Coefficient (C), N-s/m	418	673
Sprung Mass (M), Kg	41	81
Speed (N), rpm	155	200

IV. DATA ASSIMILATION

The experiment was conducted according to the 2^3 full factorial orthogonal design matrix generated in MINITAB software with three factors K (N/mm), C (N-s/m) and M (Kg) and Speed N (rpm) as a block, with two replicates i.e. total 16 number of observations are taken and corresponding value of RC in terms of RMS acceleration as a response are tabulated in Table 3.

TABLE 3. Full Factorial Design Matrix

Run Order	Blocks	M(Kg)	K (N/mm)	C (N-s/m)	RC (m/s^2)
1	1	41	26000	418	0.35
2	1	81	26000	418	0.28
3	1	81	26000	673	0.87
4	1	81	18000	418	0.34
5	1	41	18000	673	0.53
6	1	81	18000	673	1.38
7	1	41	26000	673	0.71
8	1	41	18000	418	0.41
9	2	81	26000	418	0.56
10	2	41	26000	673	0.99
11	2	81	18000	673	1.10
12	2	41	18000	418	0.66
13	2	41	26000	418	0.47
14	2	81	18000	418	0.66
15	2	81	26000	673	0.74
16	2	41	18000	673	0.72

V. REGRESSION ANALYSIS

The observations were subjected to linear regression analysis to estimate and quantify the relationships between different variables. Regression analysis helps to understand how the response changes when any one of the independent variables is varied, while the other variables are fixed.

The Pareto chart in Fig. 2, shows the effect estimates sorted by their absolute influence over the response. A vertical line shows that minimum magnitude of statistically significant effects.

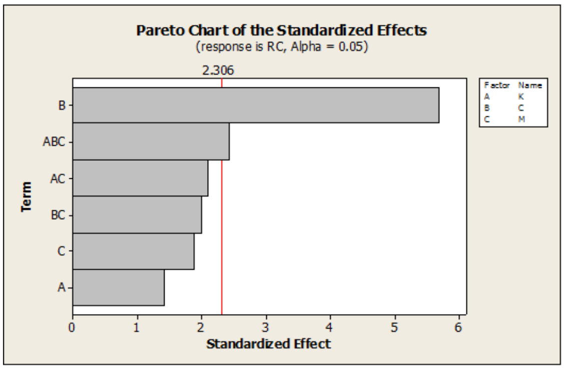

Fig. 2. Pareto Chart For Ride Comfort

The Pareto chart is very useful for reviewing a large number of factors for presenting the results of an experiment. A term is considered insignificant if 't' value is less than a critical value of 2.306 marked as the red line for 95% confidence interval. As per Fig 2. interaction of sprung mass and speed can be neglected due to its insignificance. The damping coefficient cannot be neglected as its interactions are significant.

To validate the assumptions of normality, orthogonality and homoscedasticity, the effects coefficients Table 4 is generated.

TABLE 4. Effect Coefficients Table

Term	Effect	Coef	SE Coef	T	P
Constant		0.67438	0.03624	18.61	0.000
Block		-0.06437	0.03624	-1.78	0.114
K	-0.10375	-0.05188	0.03624	-1.43	0.190
C	0.41250	0.20625	0.03624	5.69	0.000
M	0.13750	0.06875	0.03624	1.90	0.094
K*M	-0.15250	-0.07625	0.03624	-2.10	0.069
C*M	0.14625	0.07313	0.03624	2.02	0.078
K*C*M	-0.17625	-0.08813	0.03624	-2.43	0.041

VI. QUANTITATIVE ANALYSIS

The value of S, 0.144976 in Table 5 is the estimated standard deviation of regression which shows average deviation error in the model. A 89.13% R-Sq indicates that 89.13% of the variation in RC is explained by input variables and 10.87% is due to error or some unexplained factors. The closeness of the values of the R-Sq and R-Sq adjusted depict the reliability of the model. Predicted R-sq 49.68% reflects the influence of some noise factors.

TABLE 5. R Square Statistics

S	Standard Deviation	0.144976
PRESS	Prediction Sum of Squares	0.672575
R-Sq	Coeff. of Multiple Determination	87.42%
R-Sq (pred)	Predicted Coeff. of Determination	49.68%
R-Sq (adj)	Adjusted Coeff. of Determination	76.41%

VII. RESIDUAL PLOTS

Normal probability plot indicates whether the data is normally distributed, other variables are influencing the response, or outliers exist in the data.

TABLE 6. ANOVA Table

Source	DF	Seq SS	Adj SS	Adj MS	F	P
Blocks	1	0.06631	0.06631	0.06631	3.15	0.114
Main Effects	3	0.79931	0.79931	0.26644	12.68	0.002
K	1	0.04306	0.04306	0.04306	2.05	0.190
C	1	0.68063	0.68063	0.68063	32.38	0.000
M	1	0.07562	0.07562	0.07562	3.60	0.094
2-Way Interactions	2	0.17858	0.17858	0.08929	4.25	0.055
K*M	1	0.09302	0.09302	0.09302	4.43	0.069
C*M	1	0.08556	0.08556	0.08556	4.07	0.078
3-Way Interactions	1	0.12426	0.12426	0.12426	5.91	0.041
K*C*M	1	0.12426	0.12426	0.12426	5.91	0.041
Residual Error	8	0.16814	0.16814	0.02102		
Total	15	1.33659				

It is evident that residuals are normally distributed over the range. Histogram in Fig. 3 shows that due to some outlier points on extreme left and right it is not exactly symmetrical over mean value. Residual over the fitted value must be randomly scattered above and below of the mean residual zero line with constant variance as proven by residuals versus fitted values plot in Fig. 3.

Analysis of variance (ANOVA) as depicted in Table 6 is a collection of statistical models used in order to analyse the differences between group means and their associated procedures. From the Table 6 in the ANOVA main effect is observed which is significant having p-value less than 0.05. The interaction effect of all factors is less than 0.05 except that of damping coefficient and speed.

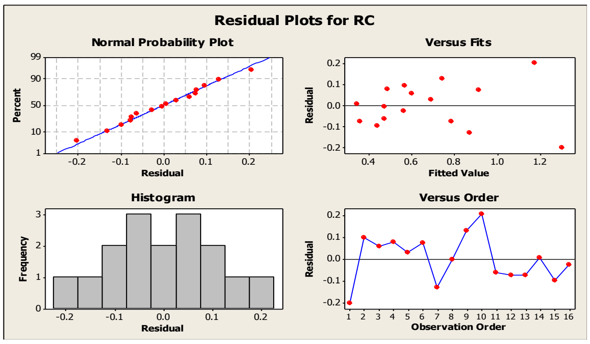

Fig. 3. Normal Probability Plots

The interpretation of linear effects is less than p-value except the linear effect of damping coefficient is more than p-value which is insignificant. There are many factors which affect the ride comfort, after including these factors the model can be fitted again by excluding insignificant interactions and covariant effect.

VIII. REGRESSION MODEL OF RC

A Regression model in equation (1) of RC in terms of significant input variables and their interaction effect is obtained from the regression analysis.

$$\begin{aligned} RC &= (-2.40982 \times 10^{-4})K - (0.0116722)C - (0.0949219)M \\ &+ (3.75983 \times 10^{-6})(K \times M) + (0.00021875)(C \times M) \\ &- (8.63971 \times 10^{-9})(K \times C \times M) - (0.0643750)Block + 5.83808 \end{aligned} \tag{1}$$

IX. SIMULATION MODELS

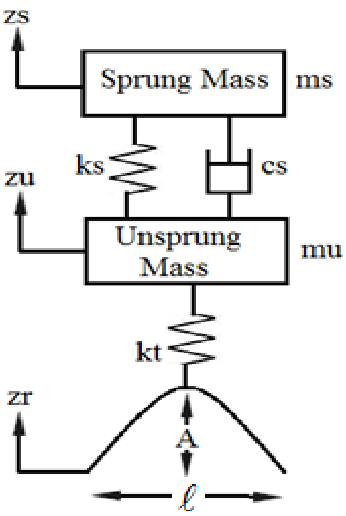

Fig. 4. Ouarter Car Model

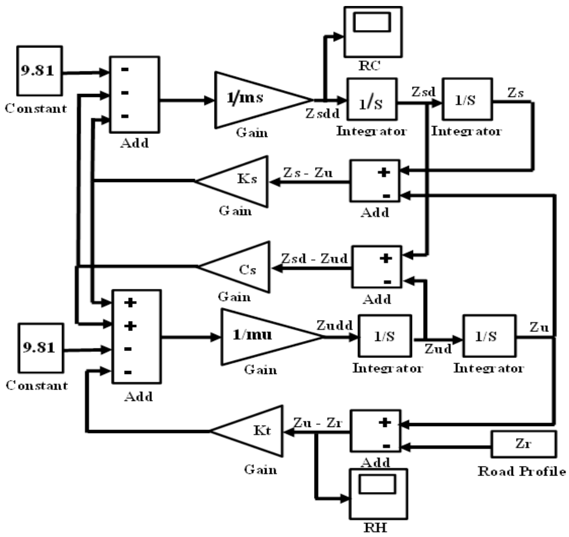

Fig. 5. Simulink Model Of Quarter Car

To analyze effects of change in various parameters on the performance characteristics of a suspension system, it was necessary to have a simulation model of the system. With this as an objective, the models of a quarter car were created in MATLAB-SIMULINK workspace as shown in Fig, 4 and Fig. 5.

A quarter car model is as shown in Fig. 4. In this model, the upper mass (ms) is the sprung mass representing the body of vehicle and the lower mass (mu) is the un-sprung mass of the wheel and other suspension parts. A road bump with half sinusoidal profile has been considered to simulate road excitation.

The effect of the bump has been modelled as zr in equations (2), (3), (4).

$$zr = 0 , \quad when \ (t < \frac{d}{v}) \tag{2}$$

$$zr = h \times \sin \left[\frac{\Pi \times v}{\ell} \times \left(t - \frac{d}{v} \right) \right] \quad for \ (\frac{d}{v} > t < \frac{d + \ell}{v}) \tag{3}$$

$$zr = 0 , when \ (t > \frac{d + \ell}{v}) \tag{4}$$

For

In the equations (2), (3) and (4) v is the velocity in m/s, d is the distance between front and rear axle in meter and t is the time lag in sec. between the crossing of front and rear wheels across the bump. This profile is created in MATLAB.

$$ms \times \ddot{z}s + cs \times (\dot{z}s - \dot{z}u) + ks \times (zs - zu) + mu \times g - ms \times g = 0 \tag{5}$$

$$mu \times \ddot{z}u - cs \times (\dot{z}s - \dot{z}u) - ks \times (zs - zu) + kt(zu - zr) + mu \times g = 0 \tag{6}$$

Fig. 5 shows the SIMULINK model of quarter car designed in MATLAB-SIMULINK workspace. This model is designed based on the equations (5) and (6), and using the readily available blocks in SIMULINK library. The parameters namely sprung mass ms, un-sprung mass mu, suspension spring stiffness ks, damping coefficient cs, tyre stiffness kt, velocity v and effect of bump zr are provided as input to this model from MATLAB workspace so that sprung mass acceleration, i.e. RC is obtained as output.

X. COMPARISON AND VALIDATION OF EXPERIMENTAL AND SIMULATION MODELS

So as to compare and determine the precision with which the models have been formulated, the CI was computed, taking into consideration the experimental and simulation models as shown in Fig. 6 and Fig. 7.

A CI value 0.929 was found to exist, which shows a high degree of conformation of the simulation model with that of the experimental model.

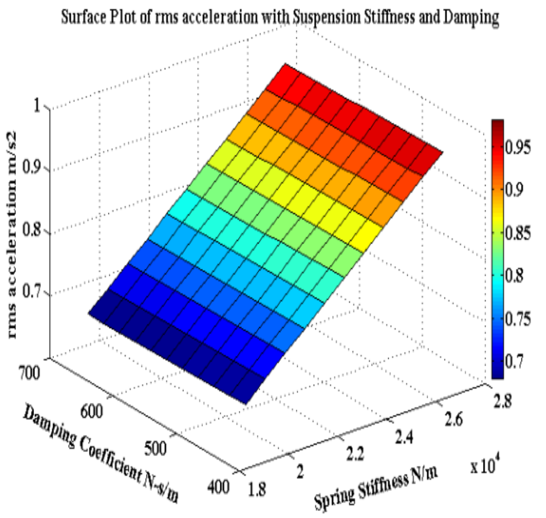

Fig. 6. Surface Plot From Experimental Model

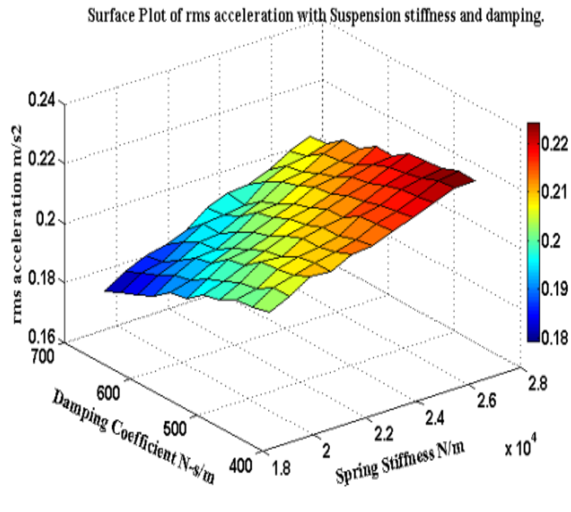

Fig. 7. Surface Plots From Simulation Model

XI. RESPONSE OPTIMIZATION

The goal of the experiment model is to find the best setting at minimum value of R.M.S. acceleration i.e. Maximum RC. As per ISO-2631-1:1997 standards ride comfort is measured in terms of the RMS acceleration of the sprung mass i.e. the vehicle body, which should be preferably below 0.5 m/s^2 for comfortable ride. The goal for RC is to obtain a value at or near the target value of 0.3 m/s^2.

The Fig. 8 shows how the factors affect the predicted responses and allows for the modification of factor settings interactively. The vertical red lines on the graph represent the current factor settings. The horizontal blue lines represent the current response values. The grey regions indicate factor settings where the corresponding response has zero desirability.

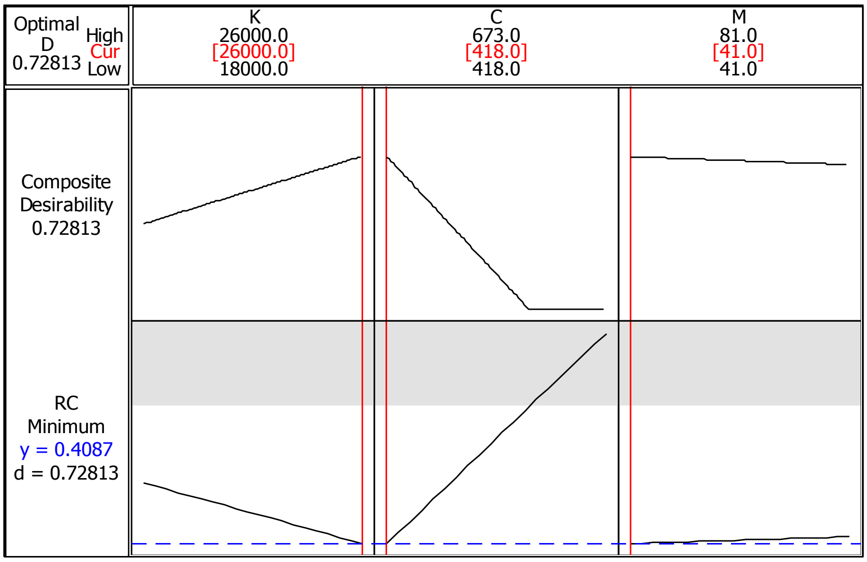

Fig. 8. Response Optimization Plot

The composite desirability is 1 as shown in Fig. 8. Desired value of RC was obtained at spring stiffness (K) 26800 N/mm, damping coefficient (C) 418 N-s/m, mass (M) 81 Kg.

XII. CONCLUSION

Ride comfort was maximized by optimizing the various suspension parameters like sprung mass, damping coefficient, spring stiffness using the DOE methodology. The generated model possesses an R-sq value 87.42%, R-sq adjusted value 76.41% and an R-sq, predicted value 49.68%. The relatively low values of R-sq predicted denote the consistent influence of significant noise factors. The values of R-sq and R-sq adjusted signify that, apart from the variables considered under this study, the ride comfort is likely to be affected by other variables also. The quarter car model developed in SIMULINK was compared with the experimental model and was found to have a high CI value 0.929. The response optimizer has been implemented to arrive at the optimized set of variables, namely a spring stiffness of 26800 N/mm, damping coefficient of 418 N-s/m, mass of 81 Kg and speed 250 rpm. The speed of 250 rpm of the cam suggests that, minimum discomfort will be experienced by the occupant of the vehicle when a bump is negotiated at a speed of 11 km/h.

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