Sweet Area Determination by Performance Sensitivity for an Automotive Vehicle Suspension*

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Using quarter car model, an analytic method for performance estimation of a vehicle suspension system is presented and as the variables of performance estimation, frequency response, RMS response and performance index are considered. From frequency response function, the characteristics of response to the whole frequency range is verified and from RMS response and performance index, sensitivity of ride and handling characteristics are examined. Using full car model, sweet area (stable ride area) are determined and performance sensitivity is estimated according to the change of suspension parameters. In order to estimate the output sensitivity, response curve is displayed using 3-dimensional plot. Design data are suggested for optimal design parameter estimation, which maximize the performance of the given suspension system.

Key Words: Quarter Car Model, Full Car Model, Suspension, Frequency Response, RMS Response, Performance Index, Sweet Area, Parameter Estimate

1. Introduction

Decreasing the spring constant and damping coefficient of a car's suspension provides good, soft riding comfort on well-paved roads⁽¹⁾. However, it results in large variation in vehicle's attitude during cornering, acceleration, and deceleration. This causes poor accident avoidance performance and inferior stability at high speeds. Besides, small values of these suspension parameters lead to a much shortened suspension stroke-leading to suspension full bumping on rough road⁽²⁾.

Solving these problems requires increasing spring constants and damping coefficients during high-speed driving, cornering, acceleration and deceleration-while decreasing them under other conditions. For

this purpose, optimum values for these two parameters must be studied and chosen for different road and driving conditions. From these values, characteristics are to be determined for both the soft (riding comfort oriented) and hard (handling oriented) conditions.

Characteristics defined by the above investigation were combined with electronics technology to develop a system which simultaneously can supply both riding comfort and handling stability. This system is called Electronic Controlled Suspension (ECS), and is capable of both automatic soft/hard switching and automatic suspension stroke adjustment by sensing driving condition⁽³⁾⁻⁽⁵⁾.

In this study, suspension spring and damping coefficient are selected as the variable parameters, and performance sensitivity analysis is carried out as the function of them. The trade-offs between vibration isolation and road holding are examined under the suspension stroke constraints. Using quarter car model, an analytic method for performance estimation of a vehicle suspension system is presented and frequency response, RMS response and performance index are considered. From frequency response function, compromise of response performance to the whole frequency range is verified and from RMS

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response and performance index, sensitivity of ride and handling characteristics are examined. Also using full car model, sweet area (stable ride area) are determined and performance sensitivity is estimated according to the change of suspension parameters. In order to estimate the output sensitivity, response curve is displayed using 3-dimensional plot. Design data are suggested for optimal design parameter estimation, which maximize the performance of the given suspension system⁽⁶⁾⁻⁽⁸⁾.

2. Performance Sensitivity Evaluation Using Quarter Car Model

2.1 Modeling and formulation

The passive suspension shown in Fig. 1 is modeled as a linear spring of stiffness k_s and a linear damping of damping rate C_s . The tire is simply modeled by a spring of stiffness k_t , i.e., tire damping is assumed to be zero. The linearized equations of motion can be easily derived as follows:

$$\begin{aligned} m_s \ddot{Z}_s + c_s(\dot{Z}_s - \dot{Z}_u) + k_s(Z_s - Z_u) &= 0 \\ m_u \ddot{Z}_u - c_s(\dot{Z}_s - \dot{Z}_u) - k_s(Z_s - Z_u) + k_t(Z_u - Z_r) &= 0 \end{aligned} \quad (1)$$

The state variables are chosen to be

$$X = \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} = \begin{bmatrix} Z_s - Z_u \\ \dot{Z}_s \\ Z_u - Z_r \\ \dot{Z}_u \end{bmatrix}, \quad A = \begin{bmatrix} 0 & 1 & 0 & -1 \\ -k_s/m_s & -c_s/m_s & 0 & c_s/m_s \\ 0 & 0 & 0 & 1 \\ k_s/m_u & c_s/m_u & -k_t/m_u & -c_s/m_u \end{bmatrix}$$

$$L = \begin{bmatrix} 0 \\ 0 \\ -1 \\ 0 \end{bmatrix}$$

To express the ride quality by numerical value, the PI (performance index) defined by Eq. (4) is introduced. The performance index to evaluate this system is given by,

$$J = \lim_{T \rightarrow \infty} \frac{1}{T} E \left[\int_0^T \{ \dot{Z}_s^2 + \rho_1 (Z_s - Z_u)^2 + \rho_2 \dot{Z}_s^2 + \rho_3 (Z_u - Z_r)^2 + \rho_4 \dot{Z}_u^2 \} dt \right] \quad (4)$$

$\rho_1, \rho_2, \rho_3, \rho_4$: Weighting factors

Equation (4) can be rewritten as:

$$\lim_{T \rightarrow \infty} \frac{1}{T} E \left[\int_0^T (X^T Q X) dt \right] \quad (5)$$

where, $Q = \begin{bmatrix} k_s^2/m_s^2 + \rho_1 & c_s k_s/m_s^2 & 0 & -c_s k_s/m_s^2 \\ c_s k_s/m_s^2 & c_s^2/m_s^2 + \rho_2 & 0 & c_s^2/m_s^2 \\ 0 & 0 & \rho_3 & 0 \\ c_s k_s/m_s^2 & c_s^2/m_s^2 & 0 & -c_s/m_s + \rho_4 \end{bmatrix}$

Since the input is white noise and the system is linear, the ergodic hypothesis is satisfied, which says that a time average is equal to an ensemble average.

$$\begin{aligned} J &= \lim_{T \rightarrow \infty} \frac{1}{T} E \left[\int_0^T (X^T Q T) dt \right] \\ &= \lim_{T \rightarrow \infty} \frac{1}{T} \left[\int_0^T (E[X^T Q X]) dt \right] \\ &= \text{Trace} [Q E[X^T X]] \\ &= \text{Trace} [Q \Sigma] \end{aligned} \quad (6)$$

where, Σ is a covariance matrix as follows:

Fig. 1 Quarter model of a vehicle suspension

$$\begin{aligned} x_1 &= Z_s - Z_u : \text{suspension deflection} \\ x_2 &= \dot{Z}_s : \text{absolute velocity of sprung mass} \\ x_3 &= Z_u - Z_r : \text{tire deflection} \\ x_4 &= \dot{Z}_u : \text{absolute velocity of unsprung mass} \end{aligned} \quad (2)$$

Therefore, Eq. (1) can be rewritten in state space representation using definitions Eq. (2).

$$\dot{X} = AX + L\dot{Z}_r \quad (3)$$

where,

$$\Sigma = E[X^T X] = \begin{bmatrix} \sigma_{11} & \sigma_{12} & \sigma_{13} & \sigma_{14} \\ \sigma_{12} & \sigma_{22} & \sigma_{23} & \sigma_{24} \\ \sigma_{13} & \sigma_{23} & \sigma_{33} & \sigma_{34} \\ \sigma_{14} & \sigma_{24} & \sigma_{34} & \sigma_{44} \end{bmatrix} \quad (7)$$

and the covariance propagation equation of Eq. (3) can be obtained as follows:

$$A\Sigma + \Sigma A^T + L\Theta L^T = 0 \quad (8)$$

where, Θ is the intensity of white noise determined by road profile and vehicle speed

The covariance propagation Eq. (8) gives us the root mean square values of acceleration of unsprung mass, suspension deflection, and tire deflection, i.e.

$$\begin{aligned} \bar{x}_2 &= \sqrt{\text{Var}[\dot{x}_2]} \\ &= \sqrt{\frac{1}{m_s} E[k_s^2 x_1^2 + c_s^2 x_2^2 + c_s^2 x_4^2 + 2c_s k_s x_1 x_2 - 2c_s x_1 x_4 - 2c_s^2 x_2 x_4]} \\ &= \sqrt{\frac{1}{m_s^2} [k_s^2 \sigma_{11} + c_s^2 \sigma_{22} + c_s^2 \sigma_{44} + 2c_s k_s \sigma_{12} - 2c_s k_s \sigma_{14} - 2c_s^2 \sigma_{24}]} \\ \bar{x}_1 &= \sqrt{\text{Var}[x_1]} = \sqrt{\sigma_{11}} \\ \bar{x}_3 &= \sqrt{\text{Var}[x_3]} = \sqrt{\sigma_{33}} \end{aligned} \quad (9)$$

where,

$$\begin{aligned} \sigma_{11} &= \frac{m_s + m_u}{2c_s} \\ \sigma_{12} &= \frac{m_s + m_u}{2m_s} \\ \sigma_{13} &= \frac{k_s(m_s + m_u)^2 - m_s m_u k_t}{2m_s k_t c_s} \\ \sigma_{14} &= \frac{m_s + m_u}{2m_s} \\ \sigma_{22} &= \frac{k_s^2(m_s + m_u)^2 - k_t\{(m_s + m_u)c_s^2 - m_s^2 k_t\}}{2m_s^2 k_t c_s} \\ \sigma_{23} &= -\frac{m_u}{2m_s} \\ \sigma_{24} &= \frac{k_s^2(m_s + m_u)^3 + k_t\{(m_s + m_u)c_s^2 - m_s m_u k_s\}}{2m_s^2 k_t c_s} \\ \sigma_{33} &= \frac{k_s^2(m_s + m_u)^3 + k_t c_s^2(m_s + m_u)^2 + m_s m_u k_t\{m_s k_t - 2(m_s + m_u)k_s\}}{2m_s^2 k_t c_s} \\ \sigma_{34} &= -\frac{1}{2} \\ \sigma_{44} &= \frac{k_s^2(m_s + m_u)^2 + k_t\{(m_s + m_u)c_s^2 + m_s^2(k_t - k_s) - 2m_s m_u k_s\}}{2m_s^2 k_t^2 c_s} \end{aligned}$$

Then, the performance index J is calculated by using covariance matrix as follows:

$$J = \frac{1}{m_s^2} (k_s^2 \sigma_{11} + c_s^2 \sigma_{22} + c_s^2 \sigma_{44} + 2c_s k_s \sigma_{12} - 2c_s k_s \sigma_{14} - 2c_s^2 \sigma_{24}) + \rho_1 \sigma_{11} + \rho_2 \sigma_{22} + \rho_3 \sigma_{33} + \rho_4 \sigma_{44} \quad (10)$$

To evaluate the frequency response function of suspension system, three kinds of transfer function are used.

$$\begin{aligned} H_{AC}(s) &= \frac{\ddot{Z}_s(s)}{\dot{Z}_r(s)} : \text{sprung mass acceleration} \\ &\quad \text{transfer function} \\ H_{SD}(s) &= \frac{Z_s(s) - Z_u(s)}{\dot{Z}_r(s)} : \text{suspension deflection} \\ &\quad \text{transfer function} \\ H_{TD}(s) &= \frac{Z_u(s) - Z_r(s)}{\dot{Z}_r(s)} : \text{tire deflection} \\ &\quad \text{function} \end{aligned} \quad (11)$$

2.2 Performance sensitivity of suspension parameters

2.2.1 Frequency response evaluation The first performance evaluation considers the effects of

parameter variation on the model's frequency response to input velocity. The variation of the spring and damping coefficient are examined for five cases of 25, 50, 100, 200, 400 percent of the nominal parameter value. Figures 2 - 4 show the three main measures of suspension performance, HA, HSD, and HTD to parameter variation. Figure 2 shows the sprung mass acceleration response due to changing c_s and k_s and the arrow represents the increasing direction of parameter value. Increasing damping reduces the response level in the first mode range, but increases at higher frequencies as shown Fig. 2(a) and (b) shows spring stiffness variation effects, not only do the natural frequencies change, but also the maximum amplitude grows as the first mode approaches the second mode.

(a) Suspension damping variation

(b) Suspension stiffness variation

Fig. 2 Frequency response of sprung mass acceleration to varying suspension parameters

The behavior of the suspension deflection is considered in Fig. 3. Figure 3(a) shows that increased damping only reduces the level. On the other hands, changing k_s as in Fig. 3(b) shows a trade-off of attenuation and amplification over the frequency range. Increased k_s amplifies the level below the first mode and 4 to 10 Hz region, but attenuates in 2–3 Hz. Spring stiffness variation also shift the system natural frequencies, especially with the first mode.

The tire deflection variation response is evaluated in Fig. 4. The damping variation of Fig. 4 shows that decreased c_s accentuates the response level at both modes, and reversed effects is shown between the two modes. With increased damping, the model approaches a single mass system with one mode at approximately 4 Hz. The variation of k_s also demonstrates trade-offs in performance over frequency. Although increasing spring stiffness shows an immediate trade-off at various frequencies, decreasing k_s attenuates lower frequencies of 1–3 Hz with little change at higher frequencies.

2.2.2 RMS response evaluation From the

(a) Suspension damping variation

(b) Suspension stiffness variation

Fig. 3 Frequency response of suspension deflection to varying suspension parameters

preceding results, it is shown that in many cases trade-offs are noticed in performance improvement at different frequencies, and this is important facts in evaluation the performance. For this reason, the RMS responses are calculated for each of the performance measures. These responses are shown in Fig. 5 as a function of suspension parameters. All RMS responses are normalized by the response obtained with the nominal suspension.

The RMS responses of the three performance criteria to variations in damping coefficient rate are displayed in Fig. 5(a) with the nominal point N. The RMS response to the variation of spring and damping coefficient are examined in the range from 10% to 400% of the nominal values. The results show that there is good compromise between isolation and road holding. It is impossible to improve performance by reducing sprung mass acceleration, suspension deflection, or tire deflection without sacrificing one or both of the other two.

The influence on performance of a variation in spring constant is shown in Fig. 5(b), the RMS suspension deflection is not a function of k_s . Examine

(a) Suspension damping variation

(b) Suspension stiffness variation

Fig. 4 Frequency response of tire deflection to varying suspension parameters

again Fig. 3(b), the suspension deflection frequency response for varying spring stiffness. It appears that the trade-offs in attenuation-amplification over all frequencies average to zero and thus the mean square remains constant. Figure 5(b) also shows that an increase in isolation (by reducing RMS sprung mass acceleration) can be achieved by reducing k_s without markedly affecting tire deflection. However, other effects of a soft spring need to be considered, such as response to an external body force or load leveling.

The next step in this analysis is to simultaneously vary two parameters and to examine the surfaces created in the response direction that is orthogonal to the varied parameter directions. These plots are an attempt to facilitate the interpretation of suspension performance and are useful in tuning adaptive suspension. Figures 6–8 show the 3-dimensional plots for RMS response surfaces to changes in spring and damping coefficients at the same time. The axes of the surfaces range in parameter values are constrained in 20 steps from 10% to 200% of the nominal values. In the figures, the numbers along the axes

(a) Suspension damping variation

(b) Suspension stiffness variation

Fig. 5 RMS response of sprung mass acceleration, suspension deflection and tire deflection to varying suspension parameters

Fig. 6 RMS response surface of sprung mass acceleration to varying suspension parameters

Fig. 7 RMS response surface of suspension deflection to varying suspension parameters

Fig. 8 RMS response surface of tire deflection to varying suspension parameters

represent each step. Above the ranges of parameter variation were studied and found to not indicate any different trends.

In Fig. 6, minimum sprung mass acceleration is obtained with a reduction of both spring stiffness and damping coefficient. A zero spring stiffness and damping coefficient produce perfect isolation because the ground input disturbance is not transferred to the sprung mass. This is of academic interest only; a suspension is still necessary to hold up the vehicle. Also, the trough in this surface shows that for a given k_s , there is a c_s that maximizes isolation.

(a) Suspension damping variation

(b) Suspension stiffness variation

Fig. 9 Performance index variation curve to varying suspension parameters

Figure 7 shows suspension deflection RMS response as a function of c_s and k_s . As shown in Fig. 5 (b), the spring stiffness has no effect on the suspension deflection, and is influenced by c_s alone. It is certainly intuitive that very small values of damping would limit suspension stroke.

In Fig. 8, consider the variation of damping and spring stiffness on the variation in tire deflection. Low values of c_s increase the RMS value for tire deflection quickly, whereas values of c_s greater than the nominal have smaller effects. Also, the tire deflection is insensitive to the variation of spring stiffness in the ranges considered, when k_s becomes larger, the sensitivity increases. Notice that there is a value of spring stiffness that minimizes tire deflection for any values of c_s along with Fig. 5(b).

2.2.3 Performance index evaluation For performance index evaluation, the PI (Performance Index) defined by Eq.(4) is calculated due to the variation of spring and damping coefficient, and the results are shown in Fig. 9.

Figure 9(a) shows the variation curve to damping coefficient, from the type of convex parabola, for a given k_s , there is a optimal value of damping that

Fig. 10 Performance index variation surface to varying suspension parameters

minimizes PI, such that, maximizes the performance. In Fig. 9(b), it is shown that the PI curve increase monotonously with increasing k_s , the smaller value of stiffness, the better the performance.

Next, for effective evaluation, 3-dimensional plot of PI curve to the simultaneous variation of spring and damping coefficient are shown in Fig. 10. As the preceding case, the axes of surface range are con-

strained in 20 steps (from 10 to 200 percent of nominal), the point N represents the nominal point. From the figures, we come to know that the design parameter more effective to system performance is spring stiffness. Compared with Fig. 9, the performance is mainly dominated by spring constant. However, it is shown that for a fixed spring there is a optimal value of damping that maximizes the system performance, this results are to be useful for optimal design of suspension system.

3. Simulation of Sweet Area Using Full Car Model

3.1 Formulation and determination of sweet area

Full-car model is 7 d.o.f (degree of freedom) modeling which consists of 3 d.o.f for sprung mass in center of gravity and 4 d.o.f for unsprung mass in front and rear axle as shown in Fig. 11. The parameter values of the given suspension system are shown in Table 1.

This full car model is described in divided degree of freedom of sprung and unsprung mass as follows :

$$Z_c = \begin{Bmatrix} Z \\ \theta \\ \phi \end{Bmatrix} \quad Z_u = \begin{Bmatrix} Z_{u1} \\ Z_{u2} \\ Z_{u3} \\ Z_{u4} \end{Bmatrix} \quad (12)$$

Using matrix and vector representation, it can be shown as follows ;

$$\begin{bmatrix} M_s & 0 \\ 0 & M_u \end{bmatrix} \begin{Bmatrix} \ddot{Z}_c \\ \ddot{Z}_u \end{Bmatrix} + \begin{bmatrix} RC_s R^T & -RC_s \\ -C_s R^T & C_s \end{bmatrix} \begin{Bmatrix} \dot{Z}_c \\ \dot{Z}_u \end{Bmatrix} + \begin{bmatrix} RK_s R^T & -RK_s \\ -K_s R^T & K_s + K_t \end{bmatrix} \begin{Bmatrix} Z_c \\ Z_u \end{Bmatrix} = \begin{Bmatrix} 0 \\ K_t Z_r \end{Bmatrix} \quad (13)$$

where,

$$M_s = \begin{bmatrix} M_z & 0 & 0 \\ 0 & I_x & 0 \\ 0 & 0 & I_y \end{bmatrix} \quad R = \begin{bmatrix} 1 & 1 & 1 & 1 \\ -t_f & t_f & -t_r & t_r \\ -a & -a & b & b \end{bmatrix}$$

$$K_s = \begin{bmatrix} KS_f + AR_f/2 & -AR_f/2 & 0 & 0 \\ -AR_f/2 & kR_f + AR_f/2 & 0 & 0 \\ 0 & 0 & KS_r + AR_r/2 & -AR_r/2 \\ 0 & 0 & -AR_r/2 & KS_r + AR_r/2 \end{bmatrix}$$

$$C_s = \begin{bmatrix} bS_f & 0 & 0 & 0 \\ 0 & bS_f & 0 & 0 \\ 0 & 0 & bS_r & 0 \\ 0 & 0 & 0 & bS_r \end{bmatrix} \quad M_u = \begin{bmatrix} m_f & 0 & 0 & 0 \\ 0 & m_f & 0 & 0 \\ 0 & 0 & m_r & 0 \\ 0 & 0 & 0 & m_r \end{bmatrix}$$

$$K_t = \begin{bmatrix} KT_f & 0 & 0 & 0 \\ 0 & KT_f & 0 & 0 \\ 0 & 0 & kT_r & 0 \\ 0 & 0 & 0 & kT_r \end{bmatrix}$$

By modal analysis technique to the equation, transient response is calculated and white noise model is used for road profile. The ride location is arranged as 105 points (15×7) consist of 15 divisions on vehicle length and 7 divisions on vehicle width. The RMS value of vertical acceleration by heaving, rolling, and pitching motion is shown in Fig. 12. The RMS distribution of each ride location is represented like a convex bowl type. And, the more the location goes backwards from the center of gravity and goes to sides from the center line, the more the RMS acceleration level increases. Therefore, the minimum location of RMS value must exist, and it can be

Table 1. Design parameter of full car model

Model name	Symbol	Unit	Value
sprung mass	M_s	kg	1460
roll moment of inertia	I_x	$\text{kg} \cdot \text{m}^2$	460
pitch moment of inertia	I_y	$\text{kg} \cdot \text{m}^2$	2460
unsprung mass	m_f (front)	kg	40
	m_r (rear)	kg	35.5
damping coefficient	bS_f (front)	$\text{N} \cdot \text{sec}/\text{m}$	1290
	bS_r (rear)	$\text{N} \cdot \text{sec}/\text{m}$	1620
spring coefficient	kS_f (front)	N/m	19960
	kS_r (rear)	N/m	17500
anti-roll bar stiffness	AR_f (front)	$\text{N}/\text{m}/\text{rad}$	19200
	AR_r (rear)	$\text{N}/\text{m}/\text{rad}$	9600
tire stiffness	kT_f (front)	N/m	175500
	kT_r (rear)	N/m	175500
distance from C.G to axle	L_f (front)	m	1.011
	L_r (rear)	m	1.803
half-track	t_f (front)	m	0.761
	t_r (rear)	m	0.755

Fig. 11 Full car model of passive suspension

guessed that a comparatively low level region surrounding the point is formed.

The stable ride area, so called sweet area, is defined as follows: The sweet area for the original model calculated by A1, A2, and B, which are determined from the RMS distribution level, is shown in Fig. 13. That is, we search the point S which has the minimum acceleration level on the center line, and then determine A1 as much as it forms 50% of the area calculated by integrating the distribution curve in front of the point S. Next, A2 is determined so as to the acceleration level of A1 is equal to that of A2, and B as much as 20% of the total integrated area on side distribution curve at the point S, which is cared for making equal acceleration level of A1 or A2. And also, sweet area of full car model is defined by the area which consists of the left and right half ellipse constituted by A1, A2, and B. The calculation is normalized by the nominal value so that the data can be used for comparison reference.

3.2 Sweet area evaluation

In this study, simulation is carried out to survey

RMS value of vertical acceleration (m/s^2)

Fig. 12 Distribution of RMS vertical acceleration for full car model

Fig. 13 Determination of the sweet area for full car model

the changing tendency of the sweet area defined previously to the variation of suspension parameters. Also spring constant and damping coefficient are selected for the suspension parameters, 20 step variation (from 10% to 200% of the nominal) is considered for the calculation.

Figure 14 shows the sweet area variation to the change of damping and spring constant, the increasing trend of sweet area is found for the smaller value of the suspension parameters. In the figure, it is mainly dependent upon damping variation rather than spring variation. This trend is in accord with the previous results of Fig. 5, it also suggests that there is a damping value which maximize the sweet area in the region of small damping value. As the parameters for deter-

Fig. 14 The change of sweet area due to damping and spring rate variation

Fig. 15 The change of distance A1 due to damping and spring rate variation

Fig. 16 The change of distance A2 due to damping and spring rate variation

mining the sweet area, the variations of A1, A2, and B are represented in Figs. 15–17. From the figures, it can be shown that the three parameters have the decreasing trend for the larger value of spring and damping coefficient, and responses more sensitively for damping variation similarly to the previous results. The levels of A1 and A2 are almost equal, but B has the comparatively low level and small variation, and they are all more dependent on increasing damp-

Fig. 17 The change of distance B due to damping and spring rate variation

(a) Before design modification

(b) After design modification

Fig. 18 The sweet area of full car model before and after design modification

ing rather than decreasing damping.

Simulation is carried out for the wide range from 10 to 200 percent variation, but the constraint exists in reality, the results of this study may be useful to the design modification in the constraint range. For example, assumed that the constraint of only 50 percent variation is considered, the maximum sweet area can be obtained when spring and damping coefficient

Fig. 19 Comparison of the acceleration at the optimal ride location before and after design modification

are taken as the minimum value in that constraint range, that is, by decreasing 50 percent of the nominal value. Figure 18 shows the sweet area before and after the design modification, it is increased by 90 percent for the original system. Also, the comparison for the acceleration record at driver's location before and after the design modification is shown in Fig. 19, it can be found that the level after the design modification is remarkably reduced, and guarantees the preceding results.

4. Conclusions

Suspension spring and damping coefficient are selected as variable parameters, and the results of performance sensitivity analysis carried out as the function of them are as follows:

(1) Frequency and RMS response are evaluated by the function of suspension parameters, and the trade-offs are identified between vibration isolation (ride quality) and road holding (handling).

(2) Suspension performance variation is visualized by 3-dimensional plot, and the data for estimating optimal parameter value and effective tuning in adaptive suspension are suggested.

(3) From the results of performance evaluation examined by the sweet area, its sensitivity is more dependent upon damping rather than spring, and also upon decreasing than increasing in damping variation.

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