

Ride Comfort Evaluation and Suspension Design Using Axiomatic Design

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(Manuscript Received March 8, 2007; Revised March 21, 2007; Accepted April 10, 2007)

Abstract

This study presents a theoretical formulation based on the axiomatic design (AD) approach to suspension systems for improving both ride comfort and static design factors (SDFs) of passenger vehicles. This approach was adapted to the kinematic design of suspension systems to create a decoupled or less coupled relationship between the functional requirements (FRs) and design parameters (DPs). SDFs related to wheel alignment and ride comfort are selected for FRs and suspension hardpoint positions are chosen for common DPs. A flexible commercial vehicle body model is used to mathematically express SDFs by defining the performance index and analyzing the dynamic characteristics for ride comfort evaluation. The sensitivity matrices are defined between the FRs and DPs. The SDF design sequences are proposed by using these matrices with the vehicle model. This study improves both ride comfort and SDFs by properly designing the kinematic DPs.

Keywords: Ride comfort; Suspension system; Kinematic design; Independence axiom; Flexible body; Static design factor; The performance index

1. Introduction

Suspension systems can be considered to be an interconnection of rigid bodies through kinematic joints or compliance elements such as rubber bushings or elastic springs. The suspension system plays an important role with regard to the performance of a vehicle in terms of its handling ability, stability, and ride comfort. Even if the power train is excellent, good driving is impossible without a good suspension system. The kinematic design of a suspension system is used to determine the positions of hardpoints or kinematic design points. The suspension is known to be a very complex mechanical system. Thus far, people have relied on their intuition and experience

for formulating the kinematic design; therefore, the objective functions of the suspension system have not yet been clarified.

Several studies based on axiomatic design (AD) theory have been conducted to overcome the disadvantages of an inefficient kinematic design process for a mechanical system. AD theory was introduced as a systematic approach to the design process (Suh, 1990 and Suh, 1995a), which is achieved using technical know-how and tentative decisions. The design axioms are applied to the creative process in the design stage, and the basic principle of analysis and decision at the conceptual design stage was derived from the independent axiom and the information axiom, respectively (Suh, 1990). Although the AD theory is very simple and distinct, it is not employed in mechanical systems. However, it has been applied to product design (Suh, 1990 and Moon

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et al., 1999) and manufacturing process design (Almström, 1998). In particular, AD has hardly been applied to the suspension system; the currently used design processes for improving the static design factor (SDF) of the suspension system include the McPherson, double wishbone, and multi-link systems. A coupled design matrix of the existing McPherson suspension system was obtained using the AD theory. However, it is difficult to predict which parts require modification since hardpoints and kinematic design points are closely coupled and the suspension system is highly nonlinear (Choi, 1999). The SDF value of the modified McPherson suspension system is optimal since it was obtained by decoupling a design matrix by using a modified design element of the coupled part (Bae, 2003). Although a suspension system can be decoupled by applying the independence axiom using the AD approach, it may be difficult to determine its effect on the ride comfort.

In this paper, the functional requirements (FRs) are defined in terms of the SDFs and the ride comfort related wheel alignment of a conventional vehicle by adapting the AD theory, and the design parameters (DPs) are hardpoints of the suspension. A target vehicle with a McPherson independent suspension at the front and a multi-link independent suspension at the rear is used. The performance index was defined in terms of an SDF in order to express the numerical values, and a dynamic analysis was performed using a flexible body for evaluating the ride comfort. We defined the sensitivity matrix between the FRs and DPs, and then formulated its design process in terms of an SDF. The design factor was independent of the SDF, and we selected a sensitive design factor for the ride comfort by applying the ride comfort and an SDF. Therefore, this study proposes a proper design method that can improve both the SDF and the ride comfort of the passenger vehicle by using the AD approach.

2. Kinematics of suspension systems

A suspension system comprises the constraint joints and multi-bodies, which can be assumed to be rigid or flexible bodies. They are constrained to move relative to one another by means of kinematic connections. The kinematic motion of the suspension system was determined using rigid bodies and joints (Nikravesh, 1988 and Bae, 2003). The suspension system generally uses a composite joint that

comprises links, arms, and struts. The kinematics of the suspension system is defined by the position of the wheel assembly, which is connected to the chassis using a kinematic joint, in the case of vertical motion of the wheel. The wheel assembly configuration comprises three translations x , y , and z and three rotations ϕ , θ , and ψ about the x , y , and z axes. The generalized coordinates of the wheel assembly may be written as $q = [x, y, z, \phi, \theta, \psi]^T$. The suspension system of a non-steerable system can be partitioned into one independent coordinate $v \equiv [z]$ and five dependent coordinates $u \equiv [x, y, \phi, \theta, \psi]^T$. The conventional method for finding the static equilibrium of a multi-body system involves solving the nonlinear constraint given by Eq. (1) (Nikravesh, 1988 and Kang, 2003).

$$\Phi(q) = 0 \quad (1)$$

$$\begin{aligned} \Delta u^j &= -\Phi_u^{-1}(u^j)\Phi(u^j) \\ u^{j+1} &= u^j + \Delta u^j \end{aligned} \quad (2)$$

In order to solve five dependent coordinates for a given independent coordinate z , the Newton-Raphson algorithm shown in Eqs. (2) is used. The dependent coordinates are computed by the numerical iteration algorithm with the constraint Jacobian Φ_u and iteration number j . If all the Newton differences are lesser than the specified tolerance, the algorithm terminates the process by returning to the routines.

In order to connect two bodies, every joint shown in Fig. 1 is required. The constraint Jacobian for these joints in the multi-body system can be expressed by the variation method shown in Table 1. The total number of constraint equations and coordinates that describe a mechanical system can be reduced if some of the joints and bodies are combined into composite joints. This technique simplifies the analytical formulation without changing the physical kinematic characteristics of the system. The constraint equations for composite joints are listed in Table 2. The link has two spherical joints at both rigid bodies. This type of joint is called a spherical-spherical (SS) joint and it requires one constraint equation. The arm has a spherical and a revolute joint. This type of joint is called a spherical-revolute (SR) joint and it requires two constraint equations. The strut has a spherical and a cylindrical joint.

This type of joint is called a spherical-cylindrical (SC) and it requires two constraint equations. Further,

the strut has a cylindrical and a universal joint, and it is called a cylindrical-universal (CU) joint, which requires two constraint equations. The suspension types of the target vehicle are simply expressed using

Table 1. Constraint equations and Jacobian matrices.

Constraint	$\Phi(q)$	$[\Phi_{q1} \ \dots \ \Phi_{qn}]$
Distance	$d^T d - l^2$	$[-2d^T \ 2d^T \tilde{s}_i \ \dots \ 2d^T \ -2d^T \tilde{s}_j]$
Spherical	d	$[-I \ \tilde{s}_i \ \dots \ I \ -\tilde{s}_j]$
Cylindrical	$\tilde{h}_i \ h_j \ \tilde{h}_i \ d$	$\begin{bmatrix} 0 & \tilde{h}_i \tilde{h}_j & \dots & 0 & -\tilde{h}_i \tilde{h}_j \\ -\tilde{h}_i & \tilde{h}_i \tilde{s}_i + d\tilde{h}_i & \dots & \tilde{h}_i & -\tilde{h}_i \tilde{s}_j \end{bmatrix}$
Revolute	$\tilde{h}_i \ h_j \ d$	$\begin{bmatrix} -I & \tilde{s}_i & \dots & I & -\tilde{s}_j \\ 0 & \tilde{h}_i \tilde{h}_i & \dots & 0 & -\tilde{h}_i \tilde{h}_j \end{bmatrix}$

Table 2. Constraint equations for composite joints.

Element	Constraint type	DOC	$[\Phi_{q1} \ \dots \ \Phi_{qn}]$
Link	SS	1	$\Phi = d^T d - l^2 = 0$
Arm	SR	2	$\Phi = \begin{bmatrix} d^T d - l^2 \\ h_i^T d \end{bmatrix} = 0$
Strut	SC, CU	2	$\Phi = \tilde{h}_i^T d = 0$

Table 3. Suspension type synthesis.

No. of link	Link combination	Scheme	Existence strut	Example
3	CU, SR, SS		○	McPherson with Tie-Rod
5	SS, SS, SS, SS, SS		×	Multi-link

Note: * is a spherical joint.
 * is a universal joint.
 * is a revolute joint.
 * is a cylindrical joint.

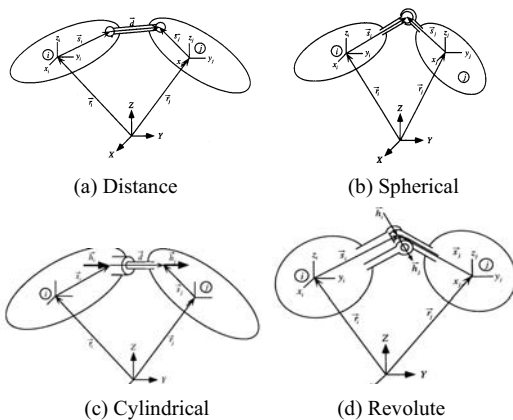


Fig. 1. Constraint joints.

the composite joints listed in Table 3 (Choi, 1999).

3. Axiomatic design approach to suspension system

3.1 Theory of the axiomatic design

The design is defined as the development and selection of the means (defined by DPs) required to satisfy the given objectives (defined by FRs), subject to the constraints specified in the concepts of the AD theory. The axiom has two design principles that determine whether a design is good or not. Axiom 1 is called the independent axiom (maintain the independence of the FRs), which is explained between the functional domain and the physical domain. Axiom 2 is called the information axiom (minimize the information content of the design), which explains the design complexity. It can be written as follows:

$$\{FR\} = [A]\{DP\} \tag{3}$$

where $\{FR\}$ is the functional requirement vector that has the m-th term and $\{DP\}$ is the design parameter vector that has the n-th term. The design matrix $[A]$ can be written as follows:

$$[A] = \begin{bmatrix} A_{11} & A_{12} & \dots & A_{1n} \\ A_{21} & A_{22} & \dots & A_{2n} \\ \vdots & \vdots & \ddots & \vdots \\ A_{m1} & A_{m2} & \dots & A_{mn} \end{bmatrix} \tag{4}$$

The design equation may have different estimations according to the design matrix and it may be classified in three forms, as listed in Table 4. The concepts of a good design can be illustrated that the type of DPs is made up of a square matrix, a diagonal matrix and a triangular matrix. In particular, a diagonal matrix is coincident with the independent axiom because the entry of a DP individually affects the entry of an FR.

Table 4. Classification of the design matrix.

No.	Condition	Example
1	$Rank(FR) = Rank(DP)$	All of elements are non zero : coupled each other Diagonal matrix : independent each other Triangular matrix : successively solvable
2	$Rank(FR) < Rank(DP)$	Redundant design
3	$Rank(FR) > Rank(DP)$	Need to increase design parameters

3.2 Development of the design matrix

The FRs and DPs of the suspension should be defined in order to design the suspension systems using the AD approach. DPs are defined as the set of elements of the design object that have been selected to satisfy the FRs. Suspension hardpoints may be considered as DPs for the kinematic analysis of the suspension system, as shown in Fig. 2. The McPherson-type front suspension system comprises an arm (LA), a strut (ST), and a link (TR). Twenty-one coordinates act as DPs according to the axiomatic design view. The global coordinates of the hardpoints are listed in Table 5. This type of suspension has seven hardpoints that have three coordinates in the global coordinate system.

The multi-link-type rear suspension system comprises five links, upper lateral (UL), lower lateral (LL), upper trailing (UT), lower trailing (LT), and tie rod (TR), as shown in Fig. 3. Twenty-seven coordinates act as DPs according to the axiomatic design view. The global coordinates of the hardpoints are listed in Table 6. This type of suspension has ten hardpoints that have three coordinates in the global coordinate system.

The kinematic characteristic of a suspension system is called the suspension geometry, which is related to the vertical motion of the wheel assembly. Five SDFs defined as the FRs of the front suspension system are described in Table 7. In the case of the front suspension, values such as the toe angle, camber angle, wheelbase change, kingpin offset, and caster angle should maintain the desired performance of the FRs. Three SDFs, with the exception of FR4 and FR5, are defined as the FRs of the rear suspension system. Values such as the toe angle, camber angle, and wheelbase change should maintain the desired performance for the FRs.

The design matrix of a suspension system is mathematically derived from the kinematic equation. The suspension geometries are functions of the wheel stroke and can be calculated from the kinematic equation. It is difficult to determine what type of effect the DPs have on the FRs. Therefore, in order to study the relationship between the previously selected FRs and DPs, we define a performance index to mathematically express the FRs after applying the AD theory. The performance index defined on the basis of Fig. 4 is as follows:

Table 5. Global coordinates of the McPherson type for front suspension.

Link	Attach	X	Y	Z
TR	Wheel	1855	703	265
TR	body	1915	330	264
LA	wheel	1716	738	135
LA	body front	1763	375	143
LA	body rear	2079	358	152
ST	wheel	1728	622	325
ST	body	1756	582	787
Wheel	cg	1724	772	256
Tire CP	cg	1724	772	-67.4

Table 6. Global coordinate of the Multi-link type for rear suspension.

Link	Attach	X	Y	Z
TR	wheel	4505	724	143
TR	body	4505	323	162
LL	wheel	4346	679	190
LL	body	1316	379	198
LT	wheel	4387	710	114
LT	body	1155	495	186
UL(UF)	wheel	4434	765	644
UL(UF)	body	4624	542	635
UT(UR)	wheel	1131	765	611
UT(UR)	body	4350	561	620
Wheel	cg	1121	768	210
Tire CP	cg	4424	768	-83.4

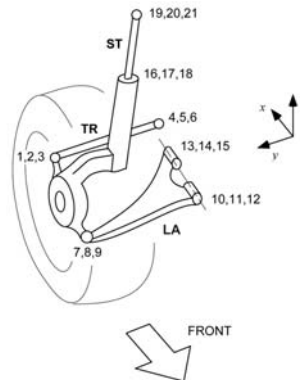


Fig. 2. McPherson type front suspension system.

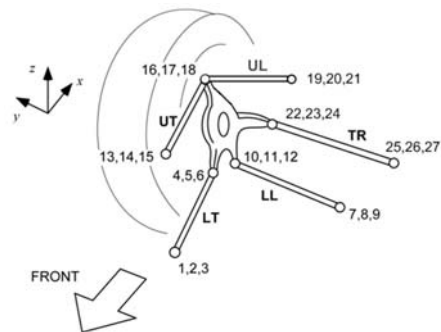


Fig. 3. Multi-link type rear suspension system.

Table 7. Definition of the FRs for static design factors.

FRs	Description
FR1	Maintain desired toe angle under wheel stroke
FR2	Maintain desired camber angle under wheel stroke
FR3	Maintain desired wheelbase change under wheel stroke
FR4	Maintain desired kingpin offset under wheel stroke
FR5	Maintain desired caster angle under wheel stroke

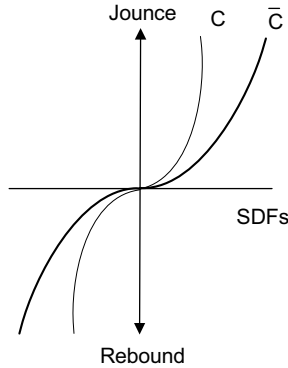


Fig. 4. Definition of the performance index.

$$I = \frac{\int (C - \bar{C})^2 dz}{\int \bar{C}^2 dz} \tag{5}$$

where \bar{C} is the desired curve of each kinematic objective and C is the vertical displacement of the wheel stroke.

The values of C are obtained from the results of the previous research (Bae, 2003). If the performance index I becomes zero, it is considered to be a desired value.

Since the kinematic equation of a suspension system is highly nonlinear, the design matrix may be written in a differential form by numerical perturbation in the neighborhood of the design position.

$$\{\Delta FR\} = \begin{bmatrix} A_{11} & A_{12} & \dots & A_{1n} \\ A_{21} & A_{22} & \dots & A_{2n} \\ \vdots & \vdots & \ddots & \vdots \\ A_{m1} & A_{m2} & \dots & A_{mn} \end{bmatrix} \{\Delta DP\} \tag{6}$$

where the elements of the design matrix are defined as $A_{ij} = \Delta I / \Delta DP = \Delta FR_i / \Delta DP_j$. For the McPherson-type suspension system, the number of FRs (m) is 5 and the number of DPs (n) is 21. The design

Table 8. Design matrix of the McPherson suspension (\otimes = strong, \emptyset = normal, 0 = weak).

DP	FR1 (Toe)	FR2 (Camber)	FR3 (W/B)	FR4 (KPO)	FR5 (Caster)
1	- 0	0	∅	- 0	∅
2	- 0	0	∅	- 0	∅
3	- \otimes	0	\otimes	0	- ∅
4	0	- 0	- ∅	- 0	∅
5	0	- 0	- ∅	0	- ∅
6	\otimes	- 0	- \otimes	- 0	∅
7	0	0	- ∅	- 0	\otimes
8	0	0	∅	\otimes	∅
9	\otimes	\otimes	- \otimes	\otimes	- ∅
10	- 0	- 0	∅	0	∅
11	- 0	- 0	∅	- 0	- ∅
12	- \otimes	- \otimes	\otimes	0	∅
13	- 0	0	- ∅	- 0	- ∅
14	- 0	0	- ∅	- 0	∅
15	- 0	0	- \otimes	- 0	- ∅
16	- 0	0	- ∅	0	∅
17	\otimes	- \otimes	- ∅	- 0	∅
18	- 0	0	∅	0	- ∅
19	0	- 0	∅	0	- \otimes
20	- \otimes	\otimes	∅	- \otimes	- ∅
21	- 0	- 0	- ∅	- 0	∅

Table 9. Design matrix of the Multi-link suspension (\otimes = strong, \emptyset = normal, 0 = weak).

DP	FR1 (Toe)	FR2 (Camber)	FR3 (W/B)
1	- 0	- 0	0
2	- 0	- 0	0
3	- 0	- 0	\otimes
4	0	0	- 0
5	0	- 0	- 0
6	0	0	- \otimes
7	0	0	- 0
8	0	- 0	- 0
9	\otimes	\otimes	- 0
10	- 0	- 0	0
11	- 0	0	0
12	- \otimes	- \otimes	0
13	0	- 0	0
14	0	0	0
15	0	- 0	\otimes
16	- 0	0	- 0
17	0	- 0	0
18	- 0	\otimes	- \otimes
19	0	0	- 0
20	- 0	0	- 0
21	0	- \otimes	- 0
22	- 0	- 0	- 0
23	0	0	- 0
24	- \otimes	- 0	- \otimes
25	0	0	0
26	- 0	- 0	0
27	\otimes	0	\otimes

matrix that is calculated using the sensitivity of the design state is shown in Tables 8 and 9. The number of DPs in the suspension system is more the number of FRs. The design matrix can be written in the form

Table 10. New DPs of McPherson suspension.

New DP	Old DP	Definition
DP_1'	DP_2, DP_1	Z coordinates of the Tie-rod
DP_2'	LP_3	Y coordinate of the Strut
DP_3'	DP_4, DP_4	Z coordinates of the A and C bush (LA)
DP_4'	DP_5, DP_5	Y, Z coordinates of the Ball joint (LA)
DP_5'	DP_7, DP_3	X coordinates of the Ball joint and the Strut Top each

Table 11. New DPs of multi-link suspension.

New DP	Old DP	Definition
DP_1'	DP_{14}, DP_{25}	Z coordinates of the Tie-rod
DP_2'	DP_{22}, DP_{22}	Z coordinates of the Upper lateral
LP_3'	DP_3, DP_{12}	Z coordinates of the Upper and Lower trailing, each

of $m < n$. For simplicity, only the effective DPs are shown and each element is represented as either \odot (strong), \emptyset (normal), or 0 (weak). Negative signs in Tables 8 and 9 represent the sign of each sensitivity value. The shaded ones are selected as the dominant DPs for each FR. To decouple the design matrix, the effective DPs are separated from the negligible DPs. Since the number of selected hardpoints (shaded ones) is still greater than the number of FRs, new DPs of the McPherson suspension are shown in Table 10.

After rearranging, the design matrix for the new set of DPs may be written as follows. Since the design matrix for the suspension system is almost a triangular matrix, it can be calculated successively. However, it cannot satisfy two elements of the design matrix because FR_1 and FR_3 are coupled with DP_1' and DP_3' , respectively. The design process and SDF values may be approximated using the design matrix (Moon et al., 1999).

$$\begin{Bmatrix} FR_5 \\ FR_4 \\ FR_3 \\ FR_2 \\ FR_1 \end{Bmatrix} = \begin{bmatrix} \times & 0 & 0 & 0 & 0 \\ 0 & \times & 0 & 0 & 0 \\ 0 & 0 & \times & 0 & \otimes \\ 0 & \times & \times & \times & 0 \\ 0 & \times & \times & 0 & \times \end{bmatrix} \begin{Bmatrix} DP_5' \\ DP_4' \\ DP_3' \\ DP_2' \\ DP_1' \end{Bmatrix} \quad (7)$$

New DPs of the multi-link suspension are shown in Table 11.

After rearranging, the design matrix for the new set of DPs may be written as

$$\begin{Bmatrix} FR_1 \\ FR_2 \\ FR_3 \end{Bmatrix} = \begin{bmatrix} \times & \otimes & 0 \\ 0 & \times & 0 \\ \times & \times & \times \end{bmatrix} \begin{Bmatrix} DP_1' \\ DP_2' \\ DP_3' \end{Bmatrix} \quad (8)$$

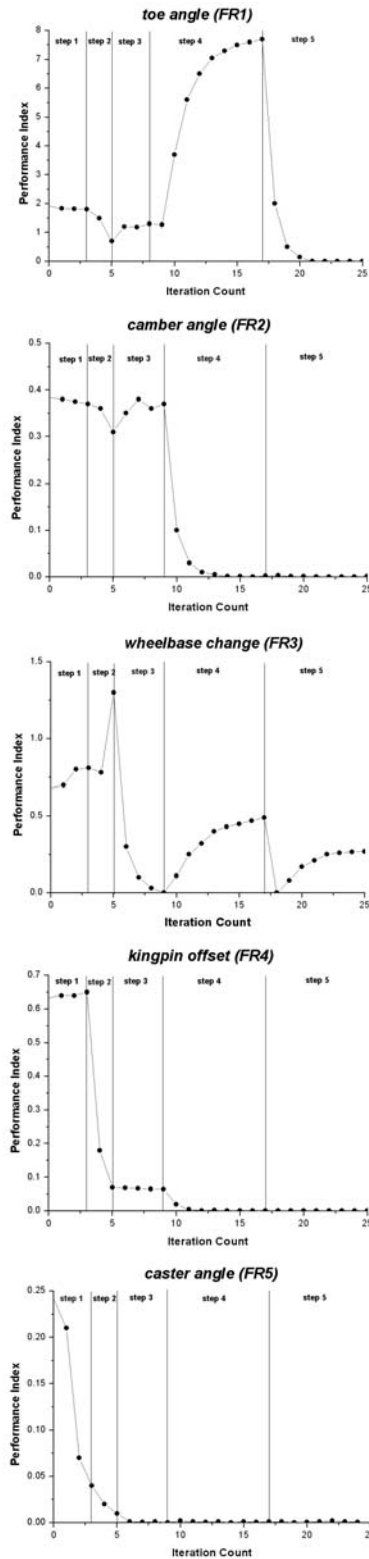


Fig. 5. Sequential design of the McPherson suspension.

Table 12. Design order of the McPherson and the multi-link suspension.

Design Order	McPherson type	Multi-link type
1	FR_1 (caster angle)	FR_2 (camber angle)
2	FR_4 (kingpin offset)	FR_3 (wheelbase change)
3	FR_5 (wheelbase change)	FR_1 (toe angle)
4	FR_2 (camber angle)	
5	FR_4 (toe angle)	

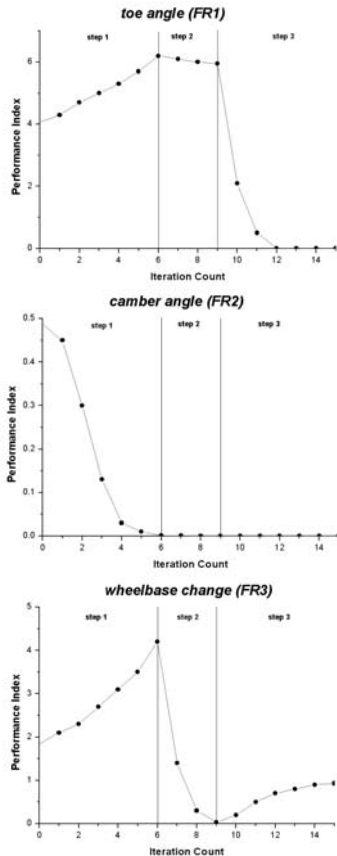


Fig. 6. Sequential design of the multi-link suspension.

To mathematically express a sequential design procedure, the increment of the hardpoints given by Eq. (6), which is selected as the DP in each step, may be calculated as

$$\{\Delta DP\} = -[A]^{-1} \{FR\} \tag{9}$$

Figure 5 shows the traces of FRs as the results of the sequential designs for the McPherson suspension.

The desired FRs can be achieved by sequential design. FR_1 and FR_3 are coupled with FR_2 ; however, FR_3 cannot be minimized because of the design sequence.

Figure 6 shows the traces of FRs as the results of the sequential designs for the multi-link suspension. The design order of the McPherson and the multi-link suspension is shown in Table 12.

4. Applications in ride comfort

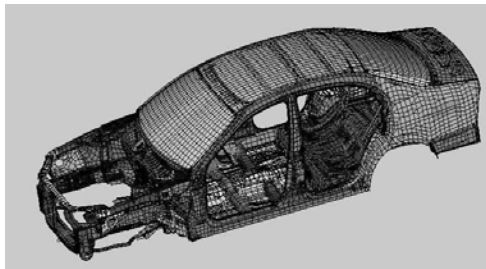
4.1 FRs and DPs for ride comfort

Researches related to human vibrations are concerned with the response of the human body to the forces exerted, frequencies, force directions, and the time of exposure. Typical examples of the standardization processes that have been developed for the research of these factors include ISO2691 (ISO2631, 1985) and BS6841 (BS6841, 1987). The human body recognizes vibrations, although the sensitivity for each vibration is different. Many researchers investigated the weightings for these vibrations and proposed axis-multiplying factors. Further, the frequency-based human sensitivities are defined as frequency weighting functions. Depending on the location and direction, different frequency weighting functions are applied with a multiplying factor. However, in order to apply these methods, results of the vehicle road test and the human body modeling for vehicle simulations are required. Therefore, to verify the responses to acceleration by using a deterministic input, Impulsive Root Mean Squares (RMSI) were used in this study in order to evaluate the ride comfort using time history data. We consider that

$$RMSI(t) = \sqrt{\frac{\int_{t-\Delta T/2}^{t+\Delta T/2} |x(t)|^2}{\Delta T}} \tag{10}$$

where $x(t)$ is the time history of the transient signal and the time interval ΔT is 0.04.

FRs and DPs should first be defined for improving the ride comfort using the AD approach. Table 13 shows the FRs defined for ride comfort. In the case of the ride comfort, positions such as the driver, floor, steering, VIP, and passenger should minimize the mean RMSI value of the FRs. In order to evaluate the ride comfort on the basis of the driver seat, driver's



(a) Flexible body



(b) Numerical analysis model

Fig. 7. Vehicle model for ride comfort simulation.

foot, steering wheel, VIP (rear right), and assistant driver, the FRs are defined such that the mean acceleration value of RMSI is minimized. The DPs that were previously selected in Figs. 2 and 3 were used equally for the connection of SDFs.

4.2 Numerical analysis model for ride comfort

A flexible body was simulated using MSC/ NASTRAN in order to analyze the ride comfort. MSC/ADAMS, which is a multi-body dynamic (MBD) analysis program, is employed for dynamic simulations of the flexible body model. Finite element analysis was performed by MSC/NASTRAN. To adjust the mass of a full vehicle using the trimmed vehicle body, Engine/TM modeling was performed. The finite element models [see Fig. 7(a)] comprise mixed vehicle bodies that use rigid body element 2 (RBE2) and spring element (CELAS2). The numerical analysis model for ride comfort is as shown in Fig. 7(b).

The modal neutral file (MNF) result of MSC/ ADAMS using MSC/NASTRAN can be verified by the modal test result. The modal test results were obtained at the front left and rear right end of the vehicle body using an electromagnetic shaker. The natural frequencies and natural modes were investigated using LMS modal software. The 5th mode shape is found to be different between the test (torsion) and MSC/ADAMS results (bending), as shown in Table 14. These results are in good agreement because the

Table 13. Definition of the FRs for ride comfort.

FRs	Description
FR1	Minimize the mean value of RMSI at the driver position
FR2	Minimize the mean value of RMSI at the floor position
FR3	Minimize the mean value of RMSI at the steering position
FR4	Minimize the mean value of RMSI at the VIP position
FR5	Minimize the mean value of RMSI at the passenger position

Table 14. Natural frequency of flexible body.

Mode	Mode shape	TEST (Hz)	ADAMS (mnf) (Hz)	Error
1	1 st Torsion	22.10	22.60	10.50
2	1 st Bending	24.04	24.13	+0.09
3	Torsion	27.88	27.95	0.83
4	Torsion	35.42	34.41	-1.01
5	Torsion (Bending)	42.30	43.38	+1.58
6	Rear Torsion	47.90	51.39	+3.49
7	Rear Torsion	56.72	63.66	+6.94

Table 15. Design matrix of the McPherson suspension by considering ride comfort (⊗ = strong, 0 = weak).

DP	FR1 (Driver)	FR2 (Floor)	FR3 (Steering)	FR4 (Passenger)	FR5 (VIP)
1	0	- 0	- 0	- 0	0
2	- ⊗	⊗	- 0	- 0	- 0
3	- 0	⊗	0	0	- 0
4	0	0	0	0	0
5	0	0	0	0	⊗
6	- 0	- 0	- 0	- 0	- 0
7	- 0	0	0	0	0
8	- 0	- 0	- 0	- 0	- 0
9	- ⊗	- ⊗	- ⊗	- ⊗	- 0
10	- 0	- 0	- 0	- 0	- 0
11	- 0	- 0	- 0	- 0	0
12	⊗	⊗	⊗	⊗	- 0
13	0	0	0	0	0
14	0	- 0	- 0	- 0	- 0
15	- 0	- 0	- 0	- 0	0
16	⊗	⊗	⊗	⊗	⊗
17	⊗	⊗	⊗	⊗	⊗
18	0	0	0	- 0	0
19	⊗	⊗	⊗	⊗	⊗
20	⊗	⊗	⊗	⊗	- 0
21	0	- 0	0	0	- 0

error tolerance is very small.

The dynamic simulation uses the 1st~7th modes for evaluating the ride comfort. In this research, the result is computed using MSC/ADAMS and Ftire, which simulates a durable tire for a deterministic input road (bar type road, 1 cm × 1 cm) with a speed of 40 km/h.

4.3 Formulation of the design matrix

To mathematically express the matrix for ride comfort, the elements of the design matrix may be written as:

Table 16. Design matrix of the Multi-link suspension by considering ride comfort (☼ = strong, 0 = weak).

DP	FR1 (Driver)	FR2 (Floor)	FR3 (Steering)	FR4 (Passenger)	FR5 (VIP)
1	- 0	- 0	- 0	0	- 0
2	0	0	0	0	0
3	- 0	- 0	- 0	- ☼	- 0
4	☼	☼	☼	☼	☼
5	- ☼	- 0	- ☼	- ☼	- 0
6	- 0	0	0	- 0	☼
7	0	☼	☼	- 0	☼
8	0	0	0	0	0
9	☼	0	0	0	0
10	- 0	- 0	- ☼	- 0	- ☼
11	- 0	- 0	0	- 0	- 0
12	- 0	0	- 0	- 0	- 0
13	0	0	0	0	0
14	0	0	0	0	0
15	- 0	- 0	- 0	- 0	- 0
16	- ☼	- 0	- 0	- 0	- 0
17	0	0	0	0	0
18	☼	0	0	☼	0
19	0	0	0	0	0
20	- 0	- 0	- 0	- 0	- 0
21	- ☼	- 0	- 0	- 0	- 0
22	0	0	0	0	0
23	0	0	0	- 0	0
24	0	- 0	0	0	0
25	0	- 0	- 0	- 0	- ☼
26	- 0	- 0	- 0	- 0	- 0
27	- 0	0	0	- 0	0

$$A_{ij} = \frac{\Delta RMSI}{\Delta DP} = \frac{\Delta FR_j}{\Delta DP_j} \quad (11)$$

The design matrices for the ride comfort of the McPherson- and multi-link-type suspension systems are calculated as shown in Tables 15 and 16, respectively. For simplicity, only the effective DPs are shown and each element is represented as either ☼ (strong) or 0 (weak). Negative signs in Tables 15 and 16 represent the sign of each sensitivity value. The shaded ones are selected as the dominant DPs for each FR.

5. Design improvement

5.1 Sensitivity analysis results of ride comfort

The design matrices in Tables 17 and 18 were acquired through the design factors of SDFs and ride comfort by combining the results of the first (see Tables 8 and 9) and second (see Tables 15 and 16) design matrices. A is a DP that is sensitive to ride comfort, B is sensitive to SDFs, C is sensitive to ride comfort and SDFs, and the shaded elements are decoupled DPs that are satisfied for the target SDF values.

Tables 17. Sensitivity matrix of the McPherson suspension (A=ride comfort, B=SDF, C=ride comfort & SDF, Shaded one= decoupled).

DP	Toe (Driver)	Camber (Floor)	W/B (Steering)	KPC (Passenger)	Chassis (VIP)
1					
2	A	A			
3	B	A			
4					
5					A
6	B				
7					B
8				B	
9	C	C	A	A	
10					
11					
12	C	C	C	A	
13					
14					
15			B		
16	A	A	A	A	A
17	A	C	A	A	A
18					
19	A	A	A	A	C
20	A	C	A	A	
21					

Tables 18. Sensitivity matrix of the Multi-link suspension (A=ride comfort, B=SDF, C=ride comfort & SDF, Shaded one= decoupled).

DP	Toe (Driver)	Camber (Floor)	W/B (Steering)	(Passenger)	(VIP)
1					
2					
3			B	A	
4	A	A	A	A	A
5	A		A	A	
6					A
7		A	A		A
8					
9	A	B			
10			A		A
11					
12		B			
13					
14					
15			B		
16	A				
17					
18	A	B		A	
19					
20					
21	A	B			
22					
23					
24	B				
25					A
26					
27	B				

The ride comfort can be improved through decoupled DPs that are independent of the design matrices of the SDFs given in Tables 17 and 18. It

was compared with SDFs along the z direction between the McPherson- [see Fig. 8 (a)] and multi-link- [see Fig. 8 (b)] type suspensions. The design is the initial values of the target vehicle, desired curves are target values for SDFs, axiom 1 curves are values that converge on the desired values using SDFs by applying the axiomatic design, and axiom 2 curves are calculated using the decoupled design matrix (shaded ones) that was developed for ride comfort after improving the SDFs. Based on the results, the

FRs of axiom 1 can be minimized to a greater extent than those of axiom 2. However, the FRs of axiom 2 are closer than the design values, which are the initial values for the desired curves.

5.2 Evaluation of ride comfort

Numerical results were acquired with MSC/ADAMS using Ftire with a vehicle speed of 40 km/h and a deterministic input road. The parameter study

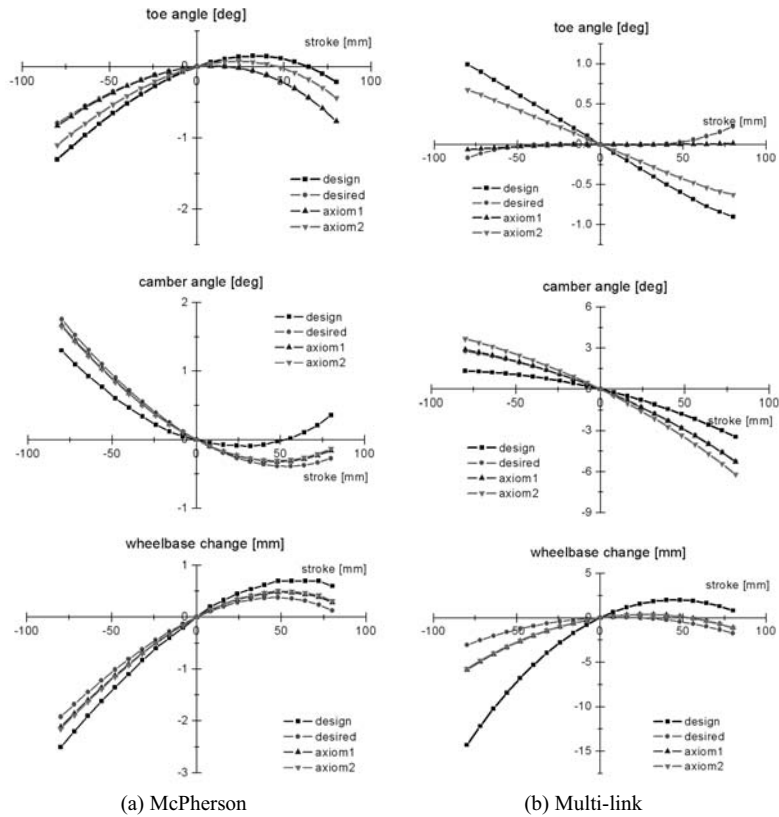


Fig. 8. SDFs of the suspension in proportion to z direction.

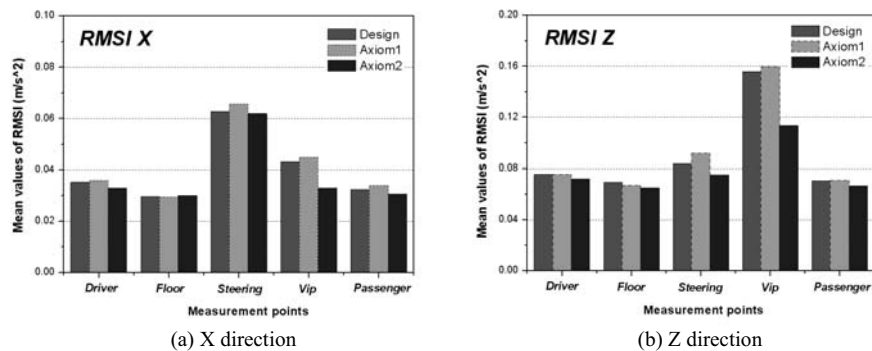


Fig. 9. Mean values of RMSI.

was performed using a decoupled design matrix developed for the ride comfort after improving the SDFs.

The design parameters that are the global coordinates of the hardpoints were changed along the x and z directions using the decoupled design matrix. They were compared with the ride comfort through the numerical analysis using the parameters of the decoupled design matrix.

Figure 9 shows the RMSI values along the x and z directions. Axiom 1 was expected to exhibit poorer ride comfort than the original design parameters. Axiom 2, which was developed for ride comfort after improving the SDFs, was expected to exhibit a good ride comfort value. In particular, the VIP position exhibited a remarkable ride comfort value.

6. Conclusions

The AD approach is adapted to a passenger vehicle that is mounted on a McPherson- and multi-link-type suspension. Kinematic analysis was conducted to define the FRs of SDFs with the vertical motion of the suspension system, and the defined FRs were numerically expressed. The design matrix is derived by using the sensitivity of the requirement index with regard to the hardpoints of the joint. The decoupling of the design matrix is carried out by using the independence axiom because it is a coupled system. Design sequences were classified as the McPherson or the multi-link suspension. The desired FRs could be achieved from the results of the sequential design. In order to evaluate the ride comfort, FRs are defined to minimize the mean acceleration value of RMSI. The natural frequency and mode shape of the flexible body as obtained by a finite element analysis code (MSC/NASTRAN) was verified by the test results. The design matrix was acquired using the design factors of the SDFs and the ride comfort by combining the results of axioms 1 and 2. Therefore, the design matrix for the ride comfort can be decoupled. We suggested that the ride comfort value obtained after improving the SDFs be used. This idea can be applied to the reduction of response peaks

based on the ride comfort by the AD approach, and the design change scheme for the improvement of the ride comfort value may be used in the suspension design.

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