International Journal of Automotive Technology, Vol. 6, No. 2, pp. 141−148 (2005)

1229−9138/2005/020−08

IMPROVEMENT OF RIDE AND HANDLING CHARACTERISTCS USING MULTI-OBJECTIVE OPTIMIZATION TECHNIQUES

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(Received 11 June 2003; Revised 15 September 2004)

ABSTRACT−In order to reduce the time and costs of improving the performance of vehicle suspensions, the techniques for optimizing damping and air spring characteristic were proposed. A full vehicle model for a bus is constructed with a car body, front and rear suspension linkages, air springs, dampers, tires, and a steering system. An air spring and a damper are modeled with nonlinear characteristics using experimental data and a curve fitting technique. The objective function for ride quality is WRMS (Weighted RMS) of the power spectral density of the vertical acceleration at the driver's seat, middle seat and rear seat. The objective function for handling performance is the RMS (Root Mean Squares) of the roll angle, roll rate, yaw rate, and lateral acceleration at the center of gravity of a body during a lane change. The design variables are determined by damping coefficients, damping exponents and curve fitting parameters of air spring characteristic curves. The Taguchi method is used in order to investigate sensitivity of design variables. Since ride and handling performances are mutually conflicting characteristics, the validity of the developed optimum design procedure is demonstrated by comparing the trends of ride and handling performance indices with respect to the ratio of weighting factors. The global criterion method is proposed to obtain the solution of multi-objective optimization problem.

KEY WORDS : Ride quality, Handling performance, Damping curve, Spring curve, Multi-objective optimization

1. INTRODUCTION

In order to improve ride and handling performance, springs and dampers are tuned through repetitive subjective feeling tests. However, these subjective tests depend on the sensibility and experience of a test driver, and require considerable time and cost. Thus, an alternative to time and cost consuming subjective tests, a ride and handling evaluation method based on computer simulation, and subsequent optimization of spring and damper characterisictics, are proposed in this research.

Driving test or simulation with 80 km/h velocity on the random profile road is performed to acquire ride quality index. A ride quality index is well known and commonly used as described in ISO standards (ISO2631, 1985(a); ISO2631, 1985(b); ISO2631, 1989) and other research (Boileau *et al.*, 1989; Griffin and Whitham, 1976; Griffin, 1986; Dahlberg, 1979; Dahlberg, 1980). Since ride quality is strongly related to human perception of vibration, a weighting function in the frequency domain was introduced to account this subjective evaluation. In order to evaluate handling performance, circular driving test (ISO4138), lane change maneuver (ISO/TR3888; Naude and Steyn, 1993), j-turn driving test and many other test techniques are used (ISO1975; ISO9816; Choi, 1991). The test methods and available index should be determined according to the design goal. For example, a lane change maneuver has been used to evaluate handling performance index.

Optimization algorithms and 'experiment design' techniques have been used to solve various design problems efficiently with computers. In order to improve ride and handling performance of commercial truck suspensions, regression analysis and optimum algorithm are used (Kang *et al.*, 1996). Truck cabin suspension design parameters are designed using experiment design method and optimization in order to achieve optimum ride quality (Bae *et al.*, 1996). The Taguchi method is widely used to control product quality and improve performance (Peace, 1993).

Since ride and handling characteristics are subjective, it is important to determine an objective function to account for those subjective measures. In this study, a ride quality index is determined by WRMS (weighted root mean squares) of PSD (power spectral density) of vertical accelerations at the 3 positions on the floor, and handling performance index is determined by RMS (root mean squares) of roll angle, roll rate, yaw rate and lateral acceleration.

Generally, vehicle suspension design problems have

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multi-objective formulation with ride and handling characteristics. It is well known that ride quality and handling performance are mutually conflicting. The 'utility function method', 'global criterion method' and 'bounded objective function method' can be used to solve multi-objective optimization problems (Rao, 1996).

The main goal of this study is the finding optimum characteristic curves of a suspension system with 2 conflicting vehicle dynamic behaviors, ride quality and handling performance, using optimization techniques. Sensitivity analysis results will be presented using the Taguchi method with respect to the objective function. A utility function that consists of 2 objective functions will be investigated with the change of weighting factors and optimum design results will be discussed. Optimum design results obtained by the global criterion method will also be presented and verified.

2. MODELING OF FULL VEHICLE

The full vehicle model for a bus is composed of a body, front axle, rear axle, link mechanism, hydraulic dampers and air springs. It is modeled using the DADS multi-body analysis software. The front and rear suspension models and rear suspensions are rigid axle types as shown in Figure 1.

The front suspension consists of an axle, tires, air springs, dampers, bushings, a stabilizer bar and a steering mechanism. Revolute joints between the axle and knuckles represent king pins. The rear suspension model consists of two pairs of dampers and air springs. The tire force is modeled with a tire carpet plot using 2D curve fitting technique. The non-linearity of the hydraulic damper and air spring is also modeled using static load test curves.

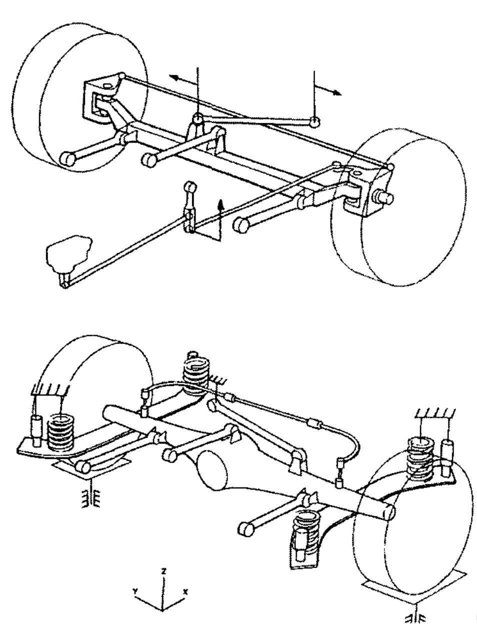
(front)
(rear)

Figure 1. Front and rear suspension model of a large bus.

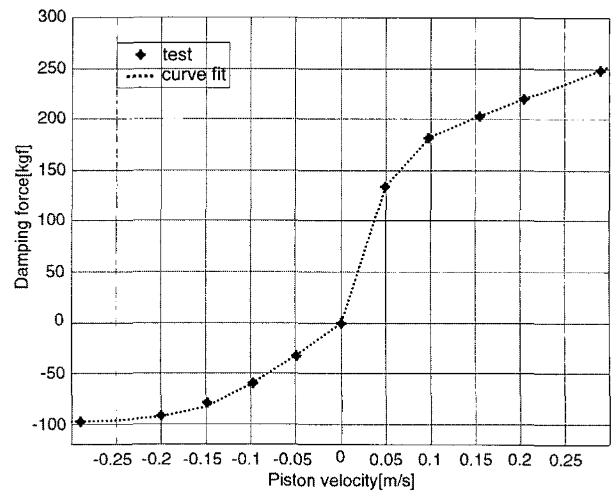

Figure 2. Comparison of test data and curve fit data of damping force curve.

2.1 Damping Curve Modeling

The damping force of a hydraulic shock absorber is obtained from pressure generated by a moving piston in the cylinder. A damping curve can be changed by adjusting the valve and orifice which control oil flow between the piston and the cylinder. In order to get a specific damping force curve, various types of valves and orifices are implemented. Generally, damping force at low speed operation depends on valve characteristics. At high speed operation, damping force depends on the orifice size of a damper. Damping force has linear characteristics during low speed operation, but non-linear (e.g. exponential curve or high order polynomial) during high speed operation. Figure 2 shows an example of damping force curve of the shock absorber which is used in the commercial large bus. The test data and the curve fitting data are in good agreement.

Since an air spring and a hydraulic damper have non-linearity, a curve fitting technique is used to model the nonlinear characteristics of the suspension. In this study, it is assumed that the damping characteristics curve has linearity in the relative velocity range $|v| \leq 0.05$(m/sec), and has non-linearity in the relative velocity range $|v| >$ 0.05(m/sec). Thus, the damping curve can be formulated using a damping coefficient and a damping exponent as in equation (1).

$$F_{d(f)}(v)=\begin{cases} -C_1|v|^{d_1} & v<-0.05 \\ C_2 v & -0.05 \leq v < 0 \\ C_3 v & 0 \leq v \leq 0.05 \\ C_4 v^{d_2} & 0.05 < v \end{cases} \tag{1a}$$

$$F_{d(r)}(v)=\begin{cases} -C_5|v|^{d_1} & v<-0.05 \\ C_6 v & -0.05 \le v < 0 \\ C_7 v & 0 \le v \le 0.05 \\ C_8 v^{d_2} & 0.05 < v \end{cases} \tag{1b}$$

where $F_{d(f)}$ and $F_{d(r)}$ are front and rear damping force respectively, and C_2, C_3, C_6, and C_7 are damping coefficients for compression and bounce in the linear range $|v| \le 0.05$(m/sec). C_1, C_4, C_5, and C_8 are coefficients of nonlinear part, and d_1, d_2, d_3, and d_4 are non-linear damping exponents. If we consider continuity condition of damping curves at $v = \pm 0.05$(m/sec), the number of parameters can be reduced. C_1, C_4, C_5, and C_8 can be replaced with the term of C_2, C_3, C_6, C_7, d_1, d_2, d_3, and d_4. Thus, 8 variables will be used as design parameters.

2.2. Spring Curve Modeling

The characteristics of a rolling robe type air spring depend on air bellows volume, effective section area of bellows, piston shape, and the installation height of the leveling valve lever. In this study, the pressure of air bellows, and the installation height of leveling valve lever are used as design variables. The characteristic curves of air springs are obtained by a 'bi-cubic spline' curve fitting technique using the data of static load test as in equation (2).

$$\begin{aligned} F_{s(f)} &= F_{s(f)}(z_f, P_f, H_f) \\ F_{s(r)} &= F_{s(r)}(z_r, P_r, H_r) \end{aligned} \tag{2}$$

where z, P, and H are the deflection of the axle, pressure of each air spring, and installation height of the leveling valve, respectively, and f and r are the front and rear axles. Figure 3 illustrates an example of an air spring characteristic curve which represents the spring force at various air bellows pressure as a function of leveling valve height.

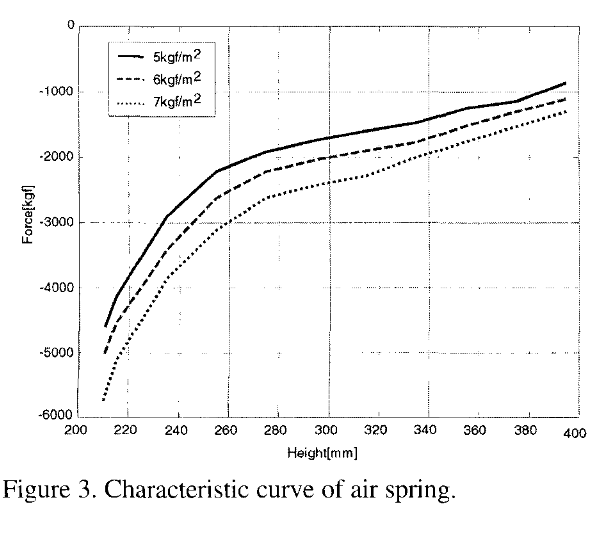

Figure 3. Characteristic curve of air spring.

3. EVALUATION OF RIDE QUALITY AND HANDLING PERFORMANCE

3.1. Ride Quality

The RMS of accelerations has been commonly used for the objective evaluation of ride quality. Since human body response to the vibration depends on the exciting frequency range and direction (Boileau *et al.*, 1989; Griffin and Whitham, 1976; Griffin, 1986), a frequency domain weighting function represented in ISO 2631 is used. According to the limit of exposure for vertical direction vibration, the weighting function $W(f)$ can be derived as in equation (3).

$$W(f)=\begin{cases} 1/0.255, & 0.1 \le Hz < 1 \\ f/0.255, & 1 \le Hz < 4 \\ 1/0.0636 & 4 \le Hz \le 8 \\ 1/(0.00094f^2), & 8 < f\ Hz \end{cases} \tag{3}$$

Using the 'utility function method' (Rao, 1996) to evaluate ride quality performance, WRMS of vertical acceleration at the 3 positions on the floor can be formulated as in equation (4).

$$J_R = \sum_{i=1}^{3} w_{R,i} \cdot J_{R,i} \tag{4}$$

where $J_{R,i}$ is obtained from the weighted RMS at position i as defined in equation (5).

$$J_{R,i} = WRMS_i = [\int W^2(f) P_i^2(f) df]^{1/2} \tag{5}$$

where $P_i(f)$ is the PSD of vertical acceleration at each position i (i = 1: front floor, i = 2: middle floor, i = 3: rear floor).

3.2. Handling Performance

In order to evaluate handling performance, a lane change maneuver is commonly used, even though it subjective since it depends on the test driver's skill and experience. A possible way to exclude the effect of the driver model for full vehicle simulation, PID controller with fixed gains is used.

A lane change maneuver is formulated to evaluate the objective handling performance in the same was as ride quality performance is evaluated. The RMS of the roll angle, roll rate, yaw rate and lateral acceleration is used for evaluating handling performance indices as in equation (6).

$$J_H = \sum_{i=1}^{4} w_{H,i} \cdot J_{H,i} \tag{6}$$

where $J_{H,1} = \sqrt{\frac{1}{T}\int_0^T \phi^2(t)dt}$, $J_{H,2} = \sqrt{\frac{1}{T}\int_0^T \dot{\phi}^2(t)dt}$,

$J_{H,3} = \sqrt{\frac{1}{T}\int_0^T \dot{\psi}^2(t)dt}$, $J_{H,4} = \sqrt{\frac{1}{T}\int_0^T \ddot{y}^2(t)dt}$, and

$\phi(t)$, $\dot{\phi}(t)$, $\dot{\psi}(t)$ and $\ddot{y}(t)$ are the response of roll angle, roll rate, yaw rate and lateral acceleration, respectively.

4. SENSITIVITY ANALYSIS

In order to perform sensitivity analysis and determine weighting factors of the terms of the cost function, the Taguchi technique is used. The sensitivity of a design variable is obtained by a main effect for S/N ratios.

The first step of Taguchi analysis is to select, considerable design variables control factors with a desired number of levels for each variable as shown in Table 1. Our design problem has 12 design variables (sections 2.1 and 2.2). A Taguchi design problem can be composed of a $L_{27}(3^{13})$ orthogonal array table which has 27 experiments and 13 control factors with 3 levels each (Peace, 1993). The main effects for the S/N ratio is obtained using performed 27 experiments with $L_{27}(3^{13})$ orthogonal array table.

In order to obtain weighting factors used in equations (4) and (6), slopes of main effects for S/N ratios are used to calculate correlation coefficients between components

Table 1. Control factors.

Design variable	Unit	Level 1	Level 2	Level 3
P_f	kg_f/m^2	5.0	6.0	7.0
P_r	kg_f/m^2	5.0	5.0	7.0
H_f	mm	240	280	320
H_r	mm	280	320	360
C_2	$kg_f/(m/s)$	300	400	500
C_3	$kg_f/(m/s)$	500	1000	1500
C_6	$kg_f/(m/s)$	200	300	400
C_7	$kg_f/(m/s)$	500	1000	1500
d_1		0.4	0.6	0.8
d_2		0.4	0.6	0.8
d_3		0.4	0.6	0.8
d_4		0.4	0.6	0.8

Table 2. Weighting factors.

$w_{R,1}$	$w_{R,2}$	$w_{R,3}$	
1.415×10^6	1.701×10^6	8.044×10^5	
$w_{H,1}$	$w_{H,2}$	$w_{H,3}$	$w_{H,4}$
1.585×10^{-1}	6.087×10^{-2}	8.738×10^{-2}	1.698×10^{-1}

of cost functions. Since the slopes give information about which control factors are sensitive to the performance, weighting factors should be determined to retain a strong correlation between design variables and components of the cost function. The weighting factors are obtained as in Table 2. Since the correlation coefficients range from 0.88–0.96, we can expect optimum design process will give us reasonable results. (Table A-1 in APPENDIX)

In this study, three cases are investigated. In the first case, the cost function consists of only the ride quality index as written in equation (4). In the second case, the cost function consists of only the handling performance index as written in equation (6). In the third case, the cost function consists of the sum of the ride quality and handling performance indices. The main effects for S/N ratios plots for the three cases are shown in Figures 4, 5 and 6.

In case 1, the control factors d_1 and d_2 are sensitive, and the damping exponents of front suspension and damping coefficient of the front suspension are effective design variables. In case 1, the damping characteristic is the

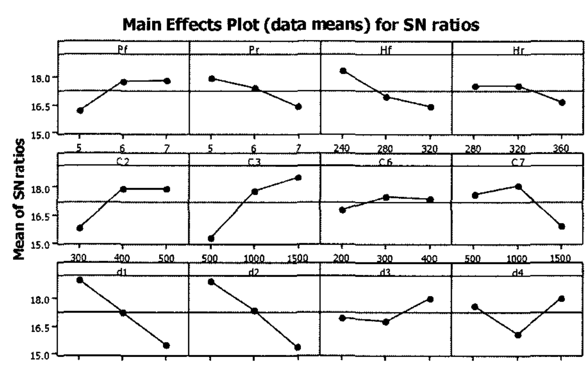

Figure 4. Main effects for S/N ratios for the case 1 (ride quality only).

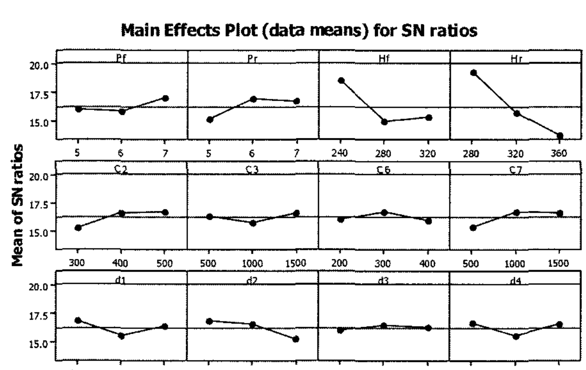

Figure. 5 Main effects for S/N ratios for the case 2 (handling performance only).

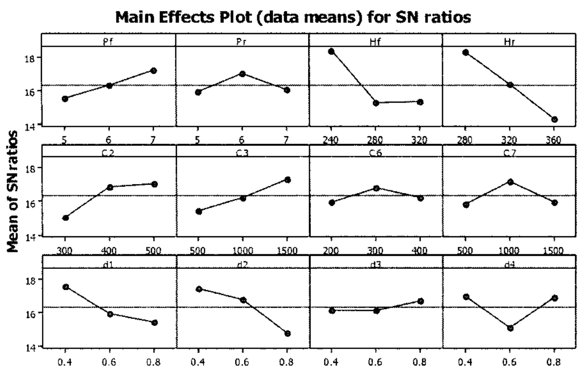

Figure 6. Main effects for S/N ratios for the case 3 (ride quality + handling performance).

dominant factor since only ride quality is considered. In case 2, the control factors H_f and H_r are sensitive, and the leveling valve height and effective pressure of the air springs are effective design variables. H_f and P_f are more sensitive than H_r and P_r . This indicates that the front suspension is more effective in controlling handling performance. In case 3, H_f, H_r, d_1, and d_2 are more sensitive than the other design variables, since the cost function is obtained from summation of ride quality index and handling performance index.

In Figures 4, 5 and 6, the optimum level of each control factor can be found and compared for the three cases. The optimum level of a design variable is indicated by the control factor level which has the biggest S/N ratio, when the Taguchi design is formulated as 'the-smaller-the-better' characteristic problem.

5. OPTIMIZATION

5.1. Utility Function Method

The formulation of an optimum design problem should be precede optimization. The ride quality index and handling performance index are used to evaluate objective functions. The design variable is defined by the damping force characteristic curve and spring force curve. If the objective function is derived using terms of ride quality index and handling performance, the objective function can be determined by a multi-objective optimization problem, and can also be formulated in vector form.

In general, no optimal solution exists that minimizes all the objective functions simultaneously. Several methods have been developed for solving a multi-objective optimization problem. The most simple technique is the 'utility function method' (Rao, 1996) using weighting factors for each objective functions as in the following equation, known as the weighting function method.

$$U=W_R J_R+W_H J_H \tag{7}$$

where J_R and J_H are cost functions for ride quality and handling performance, respectively, and, and W_R are W_H weighting factors.

5.2. Global Criterion Method

Since the optimum solution depends on the selected weighting factors, the global criterion technique is also implemented. Additionally, a benefit of the global criterion is that it does not need to be used for the selection of the weighting factors. In the global criterion method (Rao, 1996), optimum solution $\mathbf{X}^*$ is found by minimizing a pre-selected global criterion, $G(\mathbf{X})$, such as the summation of the squares of the relative deviations of the individual objective functions from the feasible ideal solutions $\mathbf{X}_k^*$ of objective function $J_k(\mathbf{X})$. Thus, $\mathbf{X}^*$ is found by a minimizing equation (8).

$$G(\mathbf{X})=\sum_{k=1}^{N}\left\{\frac{J_k(\mathbf{X}_k^*)-J_k(\mathbf{X})}{J_k(\mathbf{X}_k^*)}\right\}^2 \tag{8}$$

In order to minimize ride quality index and handling performance index simultaneously, the global criterion can be derived as in equation (9).

$$G(\mathbf{X})=\left\{\frac{J_k(\mathbf{X}_R^*)-J_k(\mathbf{X})}{J_R(\mathbf{X}_R^*)}\right\}^2+\left\{\frac{J_H(\mathbf{X}_H^*)-J_H(\mathbf{X})}{J_H(\mathbf{X}_H^*)}\right\}^2 \tag{9}$$

The modified feasible directions algorithm is used as an optimizer, and the golden section method is used as a one dimensional search algorithm. DOT (design optimization tools) (VR&D, 1995) is used to solve the optimum design problem in this study.

6. RESULTS AND DISCUSSION

6.1. Optimum Design using Utility Function Method

In order to determine appropriate weighting factors of the utility function, four cases are tried with respect to the ratio (W_R/W_H). Table 3 shows the optimum design results using various weighting factor ratios. When $W_R/W_H=0.1$, J_R and J_H of the optimal design are both reduced. (In the other cases) J_H is reduced, but J_R is increased. The appropriate ratio should be determined in order to obtain the optimum design which can improve ride quality and handling performance simultaneously.

Table 4 compares the optimum design parameters. Table 4 shows that when the ratio (W_R/W_H) becomes larger, softer damper and air spring characteristic will satisfy ride quality performance. When the weighting factor for the ride quality becomes larger, the damping coefficients C_2, C_3, C_6, C_7 have smaller values, and the

Table 3. Result of optimization.

Cases		J_R	J_H
Optimum design	$W_R/W_H = 0.01$	2.5269	3.4136
	$W_R/W_H = 0.1$	2.4497	3.4264
	$W_R/W_H = 1$	2.4224	3.4522
	$W_R/W_H = 10$	2.4087	3.4710
Initial design		2.4788	3.4875

Table 4. Optimization result of design variables.

Design variables	W_R/W_H = 0.01	W_R/W_H = 0.1	W_R/W_H = 1	W_R/W_H = 10
P_f	6.62	6.69	6.97	7.21
P_r	5.97	6.15	6.29	6.82
H_f	299.6	295.4	290.2	280.2
H_r	372.3	364.3	347.4	342.5
C_2	351.4	342.7	328.6	307.4
C_3	597.6	548.4	495.4	472.4
C_6	241.2	198.8	191.6	188.6
C_7	502.6	362.5	299.5	303.4
d_1	0.77	0.73	0.68	0.79
d_2	0.34	0.37	0.42	0.38
d_3	0.68	0.71	0.70	0.69
d_4	0.45	0.42	0.38	0.7

pressure of air springs have smaller values, but the installation height of leveling value has larger values. These results show that the sensitive design variables are extension direction damping coefficients of the rear damper, and ride quality can be improved effectively by changing the damping coefficient of the rear damper.

6.2. Optimum Design using Global Criterion Method
Three cases of optimum design are performed using the global criterion method. Figure 7 compares the PSD

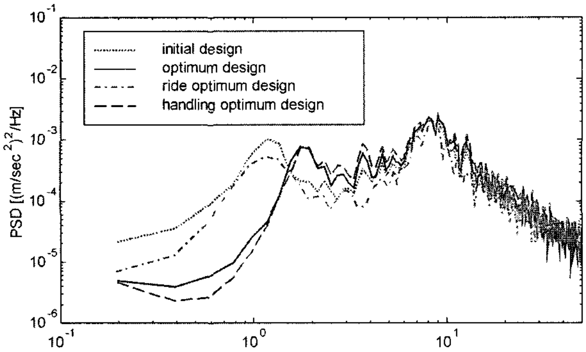

Figure 7. Comparison of PSD at driver's seat.

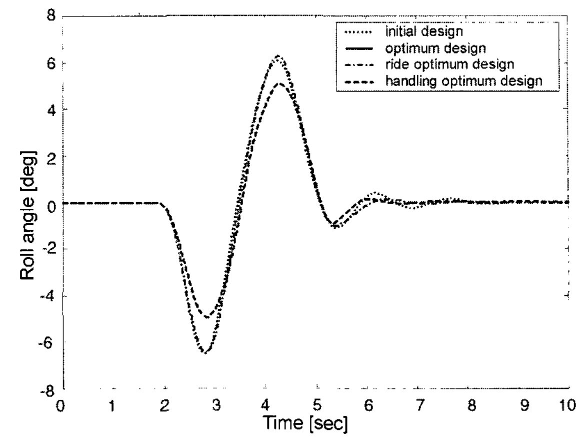

Figure 8. Comparison of roll angle.

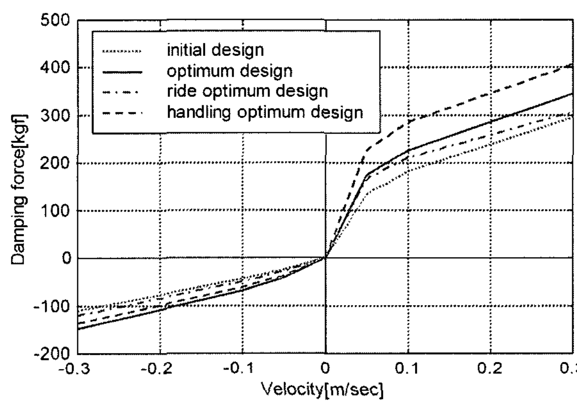

Figure 9. Comparison of damping curves (Front).

curves of accelerations at the 3 positions on the floor. Figure 8 shows the time history of the roll angle for the lane change maneuver. According to the Figure 7 and Figure 8, ride and handling performances can be improved simultaneously using the global criterion method.

Figure 9 shows the results of front damping curves for the initial design, ride quality optimum design, handling

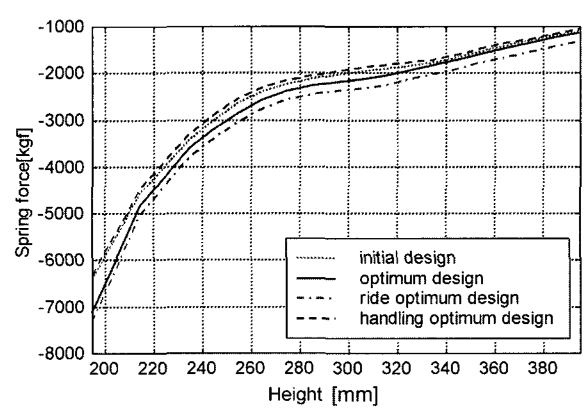

Figure 10. Comparison of spring curves (Front).

performance and global optimization. Figure 10 is comparison of the front air spring curves. Figure 10 shows that the global optimum solution for the air spring curves and the damping curves of extension direction are located between the optimum ride design and optimum handling design result. This means that the global solution is obtained in the range of Pareto optimum solutions (Rao, 1996).

7. CONCLUSIONS

The ride quality and handling performance of a full vehicle model for a bus, with a suspension consisting of air springs and hydraulic dampers, were improved simultaneously through optimum design procedures which use a utility function and the global criterion method.

When optimizing ride quality and handling performance simultaneously, the utility function method was very useful for investigating which is more sensitive with respect to changes of weighting factors. In this study, the damping coefficients and the air spring stiffness became more sensitive, as the weighting factor for ride quality has larger values.

The global criterion method was an effective way to solve multi-objective optimum design problems without weighting factors. The obtained spring curves and damping curves of global optimum design solution could be verified easily with Pareto optimum solutions.

For future work, systematic techniques which combine sensitivity analysis, optimization strategy, and algorithm will be required for effective optimum design tool.

REFERENCES

Bae, S. J. and Kim, K. T. (1996). Optimization of cab suspension parameters on heavy duty truck for improving the ride quality. *Hyundai Motors. Co. Conference Proc.*, Yongin, Korea. 164–170.

Boileau, P. E., Trucer, D. and Scory, H. (1989). Evaluation of whole-body vibration exposure using a 4th power method and comparison with ISO 2631. *J. of Sound and Vibration* **129, 1,** 143–154.

Choi, Y. H. (1991). A Study on Dynamic Design of Passenger Car Suspensions for Improved Ride and Handling Performances. *Seoul National Univ. Ph. D. Dissertation.*

Dahlberg, T. (1979). Optimization criteria for vehicle traveling on randomly profiled road – a survey. *Vehicle System Dynamics*, 239–252.

Dahlberg, T. (1980). Comparison of ride quality criteria for computer optimization of vehicles. *Vehicle System Dynamics*, 291–307.

Griffin, M. J. and Whitham, E. M. (1976). Duration of whole-body vibration exposure: its effect on quality. *J. of Sound and Vibration* **48, 3,** 333–339.

Griffin, M. J. (1986). Evaluation of vibration with respect to human response, *SAE Paper No.* 860047.

ISO 2631/1-1985(E), (1985a). Evaluation of human exposure to whole-body vibration – part 1: general requirements.

ISO 2631/3-1985(E), (1985b). Evaluation of human exposure to whole-body vibration – part 3: evaluation of human exposure to whole-body z-axis vertical in the frequency range 0.1 to 0.63 Hz.

ISO 2631/2-1989(E), (1989). Evaluation of human exposure to whole-body vibration – part 2: continuous and shock-Induced vibration in buildings.

ISO 4138, Road vehicles-steady state circular test procedure.

ISO 7975, Road vehicles-braking in a turn – open loop test method.

ISO 9816, Passenger cars-power-off reactions of a vehicle in a turn – open loop test method.

ISO/TR3888, Road vehicles-test procedure for a severe lane change maneuver.

Kang, S. H., Lee, K. H. and Kim, K. T. (1996). A study on optimization of ride & handling performance. *Hyundai Motors. Co. Conference Proc.*, Yongin, Korea. 171–177.

Naude, A. F. and Steyn, J. L. (1993). Objective evaluation of the simulated handling characteristics of a vehicle in a double lane change maneuver. *SAE Paper No.* 930826.

Peace, G. S. (1993). *Taguchi Methods: A Hands-On Approach to Quality Engineering.* Addison Wesley. New York, Singapore, Tokyo.

Rao, S. S. (1996). *Engineering Optimization: Theory and Practice 3rd Edition.* Wiley-Interscience. New York.

VR & D. (1995). DOT USERS MANUAL. Vanderplaats Research and Development Inc.

APPENDIX

Table A-1. Slope of S/N ratios of responses vs. control factors and correlation coefficients.

	$w_{R,1}\cdot J_{R,1}$	$w_{R,2}\cdot J_{R,2}$	$w_{R,3}\cdot J_{R,3}$	$w_{H,1}\cdot J_{H,1}$	$w_{H,2}\cdot J_{H,2}$	$w_{H,3}\cdot J_{H,3}$	$w_{H,4}\cdot J_{H,4}$
P_f	2.0766	1.7693	0.8152	1.1594	1.0014	0.5886	6.0589
P_r	1.3787	1.1626	0.9920	1.8251	1.3881	0.6062	6.1017
H_f	0.5538	1.9818	1.8586	6.1100	4.5815	0.6726	6.4835
H_r	1.1389	0.8986	0.8086	6.9247	7.3866	0.6051	7.9704
C_2	1.8972	2.5331	1.6432	0.2180	0.4164	0.3175	3.9700
C_3	1.9247	3.0296	2.3401	0.0760	0.2215	0.2740	2.1004
C_6	0.8318	0.6431	0.2339	0.2998	0.8227	0.3436	4.0631
C_7	1.8334	2.1360	1.7635	0.0857	1.0095	0.2792	2.1385
d_1	3.2063	4.1703	2.5870	0.3413	0.7574	0.3544	4.0118
d_2	3.2489	4.4501	3.1835	0.0832	0.8462	0.3264	3.8986
d_3	0.8022	1.0827	1.8091	0.3966	1.0789	0.3458	3.7846
d_4	1.6426	1.9007	1.5983	0.0555	0.7092	0.3098	3.8822
Correlation coefficients							
	0.9047	0.9151	0.9051	0.9366	0.9654	0.8834	0.9052