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Optimization of high speed EMU suspension parameters for vibration reduction †

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Abstract

Various designs of high speed EMU (electric multiple units) operating at maximum speed of 400 km/h or above are under active development in many countries. Driving at such extreme high speed, the EMUs that are equipped with multiple power units and non articulated bogies normally experience severe level of vibration. Noting one of the primary design issues of the vertical EMU vibration, the present work tries an optimization of vehicle suspension system using ten degree of freedom analytical EMU model. The objective function for the work is consisted of the rail power spectral density (PSD), transfer function of car body, and frequency dependent weighting function which may represent human riding comfort. The transfer function is formulated using an autocorrelation of the model in frequency domain. For the enhanced reliability of the optimization work, both the US FRA (Federal Railway Administration) rail PSD and the actual rail PSD acquired from Kyeong-bu high speed rail in Korea on which the particular EMU will be driven are used. The well proven BFGS (Broyden-Fletcher-Goldfarb-Shanno) method is adopted for the optimization. The optimized design is verified using a commercial dynamic system simulation code at various operating conditions.

Keywords: Optimization; High speed train; Suspension; Frequency domain; BFGS algorithm; Electric multiple units

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1. Introduction

Some of the high-speed trains which are under active development in many countries operate at the maximum speed of 400km/h or higher. Unlike the traditional high-speed train power car, the new version of vehicles developed in Korea is to be equipped with non-articulated bogies and distributed power plant. As a matter of fact, design engineers are interested in the study for the possibility of excessive vibrations caused by the distributed power generation systems, which are attached underneath the floor of each passenger car. Few of publications or previous research activities are available for this high-speed train designs and related design concerns. In addition, similar to other traditional high speed trains since rail irregularity and complicate contact situation between the rail and the wheel of a train are the major source of excitation, a comprehensive study on the overall vehicle dynamics should be carried for the reduction of the vibration of the distributed power system trains. Recognizing the vehicle dynamic characteristics under the extreme operating condition such as speed of 400Km/h which might affect the stability of dynamics system, the present paper tries to address technical and design issues of the optimization of carbody suspension system.

Design of suspension related to the sensitivity analysis. But it is hard to provide optimum value and consider stability of system [1]. So vehicle suspension design optimization problems have been heavily investigated in automobile design and manufacturing industry as the dynamic stability margin and the comfort of rider are major interests. Minimization of vertical acceleration of a car body is carried out by defining a objective function in time domain [2]. An objective function was formulated to minimize the vertical maximum acceleration of the car body in time domain by Haug [3, 5].

However, it is difficult to know much information of system in time domain. Therefore many researches were performed in frequency domain. Effects of vibration on ride comfort differs according to direction and frequency, based on information in UIC 513R and ISO 2631 [2, 4]. Analysis method in time and frequency domain was represented with various road about a car by Bruce [6]. Kim obtained a transfer function using a numerical method. And he optimized the vertical suspension of a car by using gradient optimal algorithm in the frequency domain [7]. Nishimura optimized the suspension of a high speed train for speeds below 300km/h in the frequency domain [8].

The optimization of the suspension is done in frequency domain as previous statement. But it is difficult to derive the

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transfer function theoretically. So Myers applied the method of a response surface model (RSM) to problems of complex optimization [8]. Park tried to optimize the suspension of the Korean High Speed Train by using design of experiments and neural network to consider the various design variables of a railway vehicle [10]. But these methods need real plant.

Therefore, the optimization of the suspension is carried out by decreasing the degrees of freedom of an equivalent model [6, 7]. Motion of a train was simulated with power spectral density (PSD) of rail irregularities by Hedrick [11]. Garge represented diverse methods of the modeling of a train [12].The effect of linear suspension to ride comfort of a train was tested using numerical simulation by Dukkipati [13].

Other existing research is one about active suspension which has been the main currents than that about passive suspension. In reality, however, a high speed train has a passive suspension for safety in relation with the malfunction of an active suspension. Although the active suspension system for a vehicle is the main trend in current vibration isolation research, the passive suspensions are still dominantly used for the high speed trains mainly due to the safety concerns. Hence a passive suspension system is to be considered in the present work. Noting the dominancy of vertical acceleration component in the overall ride comfort and considering the information given by the UIC 513R and ISO 2631 specifications, the present work focuses on the vertical vibrations for the formulation of objective function.

In the present work the optimum suspension design of a high speed train which operates at maximum speed of 400 km/h is acquired through the minimization of vertical vibration. The speed is faster than one of existing research in Korea. In 400 km/h speed, optimization of suspension has not been performed in Korea. The objective function for the optimization target consists of rail PSD, transfer function of the car body, and frequency weighting function which is defined by the definition of ride comfort. The optimal values of the primary and the secondary suspensions are derived and validated through a series of numerical simulations.

Fig. 1. Korean high-speed train (HSR–350x).

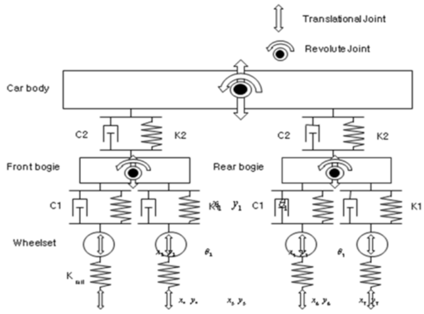


Fig. 2. Schematic diagram of 10 DOF system.

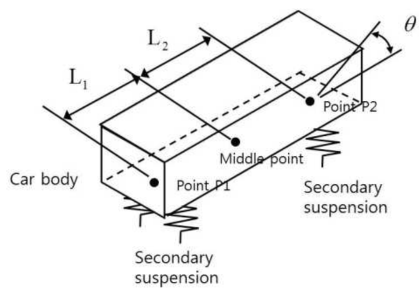


Fig. 3. Three points of car body.

2. Modeling of a vehicle

2.1 Equation of motion

In order to perform the EMU vehicle suspension optimization, one car body has been modeled in this article. Since the power car of the Korean High Speed Train which is shown in Fig. 1 has a non-articulated bogie and power units, it can be considered to represent a multiple power unit type vehicle. Most of design parameter sets are based from the Korean High Speed Train system. A model of 10 DOF system is sketched in Fig. 2 and the corresponding equations of motion are provided Eq. (1) [8].

2.2 Transfer function

Noting a typical process for suspension design optimization, a transfer function which describes a relationship between the excitation source and the acceleration of the car body is to be determined in this section.

To determine the relationship between input and output, a transfer function was derived. The input is the contact interaction between wheel and rail. The outputs are the vertical accelerations of the middle and upper secondary suspensions at the car body in Fig. 3. If the vertical acceleration of the middle of the car body is considered only, the effects of the pitching of the car body could be dismissed. The vibration of the car body is measured at various points to assess ride comfort; three points were selected. To obtain the transfer function, the equation of motion of the vehicle model is represented as Eq. (1) as a matrix and vector.

$$M\ddot{q}(t) + C\dot{q}(t) + Kq(t) = K_{ct}d(t) \tag{1}$$

Here,

$$M = \begin{bmatrix} M_{cb} & & & & & \cdots & & & & 0 \\ & J_{cb} & & & & & & & & \\ & & M_{fb} & & & & & & & \\ & & & J_{fb} & & & & & & \\ \vdots & & & & M_{rb} & & & & & \vdots \\ & & & & & J_{rb} & & & & \\ & & & & & & M_{w1} & & & \\ & & & & & & & M_{w2} & & \\ & & & & & & & & M_{w3} & \\ 0 & & & & & \cdots & & & & M_{w4} \end{bmatrix}$$

$$C = \begin{bmatrix} 2C_2 & 0 & -C_2 & 0 & -C_2 & 0 & 0 & 0 & 0 & 0 \\ 0 & 2C_2L_1^2 & C_2L_1 & 0 & -C_2L_1 & 0 & 0 & 0 & 0 & 0 \\ -C_2 & C_2L_1 & 2C_1+C_2 & 0 & 0 & 0 & -C_1 & -C_1 & 0 & 0 \\ 0 & 0 & 0 & 2C_1L_2^2 & 0 & 0 & C_1L_2 & -C_1L_2 & 0 & 0 \\ -C_2 & -C_2L_1 & 0 & 0 & 2C_1+C_2 & 0 & 0 & 0 & -C_1 & -C_1 \\ 0 & 0 & 0 & 0 & 0 & 2C_1L_3^2 & 0 & 0 & C_1L_3 & -C_1L_3 \\ 0 & 0 & -C_1 & C_1L_2 & 0 & 0 & C_1 & 0 & 0 & 0 \\ 0 & 0 & -C_1 & -C_1L_2 & 0 & 0 & 0 & C_2 & 0 & 0 \\ 0 & 0 & 0 & 0 & -C_1 & C_1L_3 & 0 & 0 & C_3 & 0 \\ 0 & 0 & 0 & 0 & -C_1 & -C_1L_3 & 0 & 0 & 0 & C_4 \end{bmatrix}$$

$$K = \begin{bmatrix} 2K_2 & 0 & -K_2 & 0 & -K_2 & 0 & 0 & 0 & 0 & 0 \\ 0 & 2K_2L_1^2 & K_2L_1 & 0 & -K_2L_1 & 0 & 0 & 0 & 0 & 0 \\ -K_2 & K_2L_1 & 2K_1+K_2 & 0 & 0 & 0 & -K_1 & -K_1 & 0 & 0 \\ 0 & 0 & 0 & 2K_1L_2^2 & 0 & 0 & K_1L_2 & -K_1L_2 & 0 & 0 \\ -K_2 & -K_2L_1 & 0 & 0 & 2K_1+K_2 & 0 & 0 & 0 & -K_1 & -K_1 \\ 0 & 0 & 0 & 0 & 0 & 2K_1L_3^2 & 0 & 0 & K_1L_3 & -K_1L_3 \\ 0 & 0 & -K_1 & K_1L_2 & 0 & 0 & K_1 & 0 & 0 & 0 \\ 0 & 0 & -K_1 & -K_1L_2 & 0 & 0 & 0 & K_2 & 0 & 0 \\ 0 & 0 & 0 & 0 & -K_1 & K_1L_3 & 0 & 0 & K_3 & 0 \\ 0 & 0 & 0 & 0 & -K_1 & -K_1L_3 & 0 & 0 & 0 & K_4 \end{bmatrix}$$

$$q(t) = \begin{bmatrix} y_{cb} & \theta_{cb} & y_{fb} & \theta_{fb} & y_{rb} & \theta_{rb} & y_{w1} & y_{w2} & y_{w3} & y_{w4} \end{bmatrix}^T$$

$$K_{ct} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 & K_r & K_r & K_r & K_r \end{bmatrix}$$

$$d(t) = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 & d_1(t) & d_2(t) & d_3(t) & d_4(t) \end{bmatrix}^T$$

The equation of motion in the time domain should be transformed to one in the frequency domain to obtain the transfer function. The relationship between the response and the input in the frequency domain are needed to do this.

The response is a Fourier frequency transformation of the impact unit function. The velocity and acceleration of the impact unit function can be transformed using Fourier frequency transformation in Eqs. (2)-(4). The equation of motion in the frequency domain can be represented using this relationship and linearity.

$$H^{(k+1)} = H^{(k)} + D^{(k)} + E^{(k)} \tag{2}$$

$$\dot{Y}(\omega) = i\omega H(\omega) \tag{3}$$

$$\ddot{Y}(\omega) = -\omega^2 H(\omega) \tag{4}$$

Since there are time delays among inputs, the relation among inputs should be considered. There is time delay between, rail displacement of first wheelset, and, one of the second wheelset. So the relationship between the rail displacement of the first wheelset and the rail displacement of the others can be expressed in Eq. (5).

$$d_i(t) = d_1\left(t - \frac{l_i}{v}\right) \qquad i = 2, 3, 4 \tag{5}$$

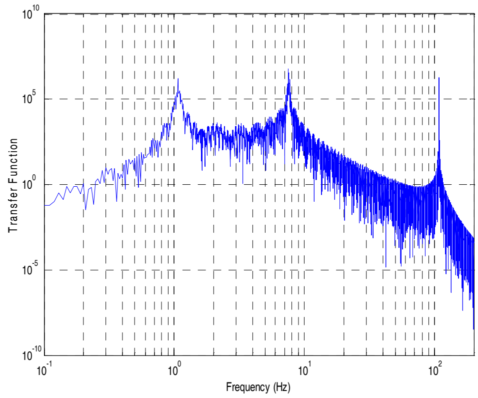


Fig. 4. Transfer function of car body.

To transfer these to PSD, autocorrelations were derived in Eq. (6). It was transformed to Eq. (6) by applying Fourier transformation to Eq. (7). Eq. (7) represents the relation among the rail displacements of the wheelsets.

$$D_i(t) = \left(t - \frac{l_i}{v}\right) D_1(t) \qquad i = 2, 3, 4 \tag{6}$$

$$\begin{aligned} S_{out,i}(t) &= \int_{-\infty}^{\infty} \left(t - \frac{l_i}{V}\right) D_1(t) e^{-i\omega t dt} \\ &= e^{-\frac{v}{l_i}\omega t} S_1(\omega) \qquad i = 2, 3, 4 \end{aligned} \tag{7}$$

Eq. (1) can be changed to Eq. (7) in the frequency domain using Eqs. (2)-(4). Then, the transfer function of the input and output can be obtained. Fig. 4 is the obtained transfer function.

$$H(\omega)\left(-\omega^2 M + i\omega C + K\right) = D(\omega) \tag{8}$$

2.3 Input PSD

Rail irregularity is expressed as a probability function. Rail irregularity is offered as a power spectral density by the Federal Railway Administration in USA [13] Its data based on enormous test data. Actual rail that high speed EMU run in is the Kyeong-Bu high speed line. But this is not a representative data. Therefore, the rail irregularities of both FRA and Kyeong-Bu high speed line were considered as inputs of the vehicle system.

Rail irregularity suggested by FRA is given by Eq. (9) and shown in Fig. 5 [13]. The irregularities of 300 km/h, 350 km/h and 400 km/h are represented in Fig. 5. When the vehicle speed increases, the shapes move right due to the increment of the spatial frequency caused by the increase of speed.

$$S(f) = \frac{B_{VorL}\,\omega_c^2}{(2\pi f)^2 \left\{ (2\pi f)^2 + \omega_c^2 \right\}} \tag{9}$$

Date of Kyeong-Bu high speed line is value between Si-Heung and Dae-Jeon. Fig. 6 represents the rail irregularity of

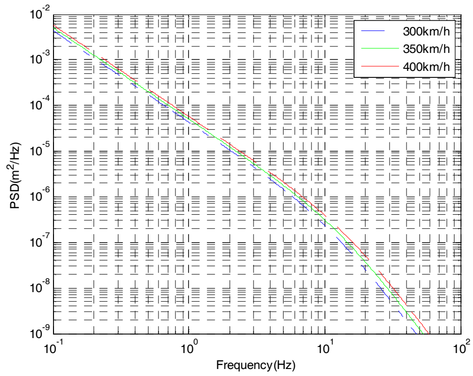
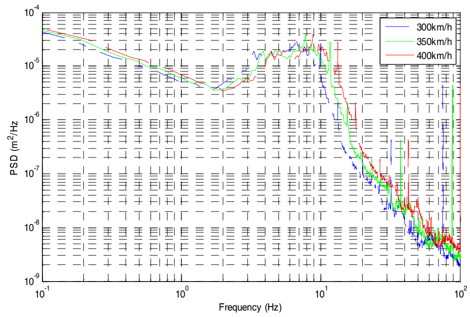
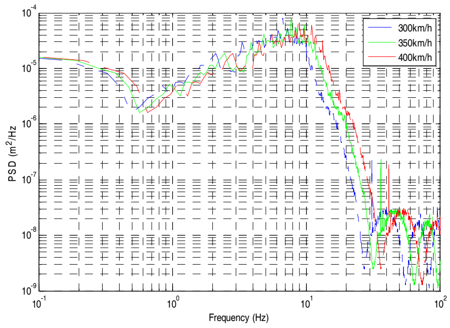


Fig. 5. Track irregularities of FRA.



(a) Lateral Irregularity



(b) Vertical irregularity

Fig. 6. Track irregularities of Kyeong-Bu high speed line.

the Kyeon-Bu high speed line in the vertical and the lateral direction. In the latter direction, the effect near 1 Hz is more effective than the one near 10 Hz. On the other hand, the effect near 10 Hz is dominant in the vertical irregularities.

3. Optimization of suspension

3.1 Objective function

Ride comfort is evaluated with respect to the vertical, lateral and longitudinal accelerations of the car body and bogie. And weighting function is multiplied by the acceleration of each direction. The objective function consists of the frequency weighting function, transfer function and input PSD in Eq. (10). It is an objective function of the vertical acceleration at the middle point of the car body. Design variables are vertical

Table 1. Optimal value.

Track	Suspension	Before	After	Δ
Kyeong -Bu	1st stiffness [N/m]	741000	759000	18000
	2nd stiffness [N/m]	616000	671000	55000
	1st damping [Ns/m]	10200	11500	1300
	2nd damping [Ns/m]	19400	23000	36000
FRA	1st stiffness [N/m]	741000	710000	31000
	2nd stiffness [N/m]	616000	616000	0
	1st damping [Ns/m]	10200	11500	1300
	2nd damping [Ns/m]	19400	23000	3600

parameters such as 1st and 2nd vertical spring coefficients and damping coefficients. It is difficult to determine the upper and lower limits of the design variables. And these limits are related to the stability of the system. These limits were set to the original value ± 10% of Korean High Speed EMU.

$$a = \int_{f_1}^{f_2} K(f)\left|\ddot{Y}_1(f)\right| S(f)df \tag{10}$$

In the case of a symmetric model, this objective function cannot exactly represent the effects of the rotation of the car body, so the accelerations of the points upper two bogie were considered in Eq. (13). For this transfer function, points P_1 and P_3 should be known. Eq. (11) is the relationship among the displacements of the three points and the rotation of the middle points of the car body.

$$y_{p2} = y_1 + L_1\theta_1 \tag{11}$$

The transfer function of points P1 and P3 can be found using Eq. (12), a relation among transfer functions. The final transfer function is given by Eq. (13). Each weighting factor of the objective function is 1 since effects of vibration at the three points are same.

$$H_{Pi} = H_{y1} \pm L_1 H_{\theta 1} \qquad i = 1: -,\ 2: + \tag{12}$$

$$a = \sum_{i=1}^{3} \int_{f_1}^{f_2} K(f)\left|\ddot{Y}_{Pi}(f)\right| S(f)df \tag{13}$$

3.2 Optimization algorithm

The optimization algorithm which minimizes the objective function is the BFGS method. Its efficiency was demonstrated by many researchers [15]. In this paper, the BFGS algorithm is used without its demonstration. It modifies the Hessian matrix. Eq. (14) represents the modification of the Hessian matrix which is the key point of this algorithm.

$$\mathrm{H}_{\mathrm{hes}}^{(k+1)} = \mathrm{H}_{\mathrm{hes}}^{(k)} + \mathrm{E}^{(k)} + \mathrm{F}^{(k)} \tag{14}$$

The optimal values obtained from the optimization algorithm are given in Table 1.

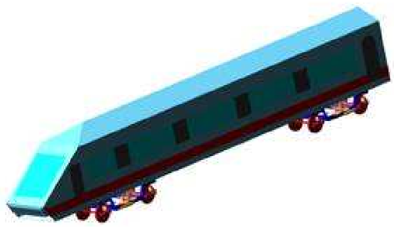


Fig. 7. ADAMS/Rail model of railway vehicle.

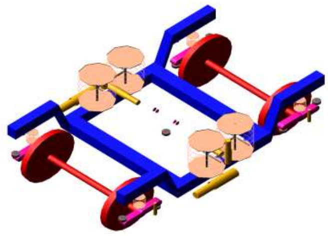


Fig. 8. ADAMS/Rail model of bogie.

4. Simulation of optimization model

4.1 Model of ADAMS/Rail

To verify the optimal value of the suspension derived by BFGS, an ADAMS/Rail model of the high speed EMU was designed. The specification of the ADAMS/Rail model is the same as that of the Korean High Speed Train in the way of method applied to 10 DOF model. A secondary spring consists of two coil springs, not an air spring characteristically. The center pivot was designed as a bump stop and busing elements. One volume of the vehicle is shown in Fig. 7. A model of its bogie is shown in Fig. 8.

4.2 Track irregularity

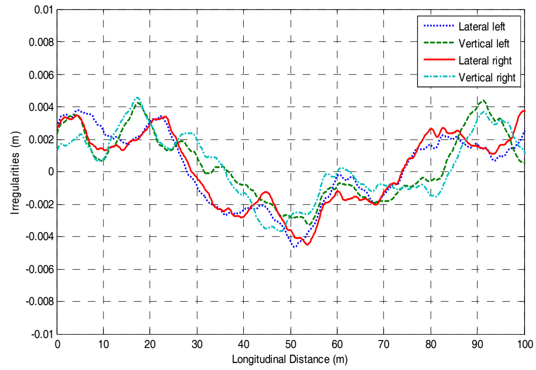
Since the track input data suggested by FRA is a function of frequency, it needs to be changed from one of the frequency domain to one of the time domain to simulate the ADAMS/Rail model in the time domain. The track input PSD of FRA can change the input data in the time domain using Eq. (15). Fig. 9 is the input data changed to data in the time domain. It represents the vertical displacement of the rail longitudinally. It is used as the input data of the wheel/rail interaction.

$$z(x)=\sum_{i=1}^{N}\sqrt{4S_z(\omega)\Delta\omega}\cos(2\pi\omega x-\beta) \quad (15)$$

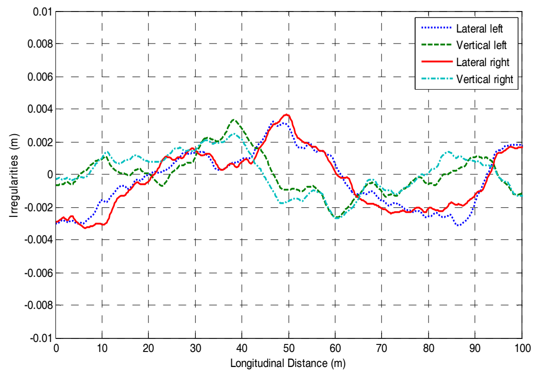
Fig. 9(a) represents the vertical displacement for the 6-class track . Fig. 9(b) represents the vertical displacement for the 5-class track. Displacement of the 6- class track is bigger than that of the 5- class track.

4.3 Optimal value

The optimal value of suspension derived by optimization



(a) 6 class



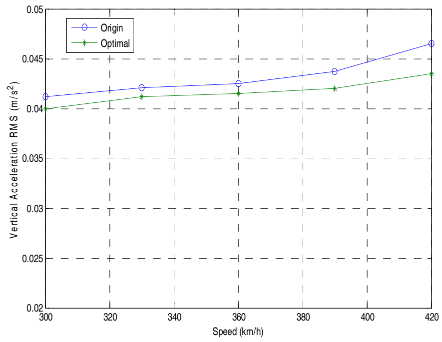
(b) 5 class

Fig. 9. ADAMS/Rail model of bogie.

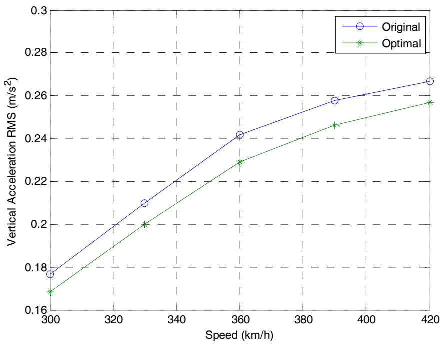
was applied to the ADAMS/Rail model. And a simulation was performed to verify the optimized suspension. The input is the track irregularity PSD of FRA and Kyeong-Bu high speed line. Results are shown in Fig. 10. It represents original and optimal vibration. We can see the reduction of vibration of the car body.

5. Conclusion

To improve the ride comfort and reduce the vibration of a high speed EMU with maximum speed 400 km/h, a vertical model of 10 DOF was designed and an objective function related to the ride comfort and vibration of the car body was defined in the frequency domain. The vibration of the upper bogie was considered, so there were rotation effects on the objective function. The optimal value of the suspension was derived by using an optimization algorithm. The derived value of suspension was applied to the ADAMS/Rail model, which was simulated with the track input data of FRA and Kyeong-Bu high speed line, to investigate the reduction of vibration. The optimal primary and secondary stiffnesses were 759000 N/m and 671000 N/m respectively. The optimal primary and secondary damping were 11500 Ns/m and 23000 Ns/m, respectively. In the future, the lateral degree of freedom, nonlinearity and flexibility of the car body will need to be considered.



(a) Track of Kyeong-Bu high speed line



(b) Track of FRA

Fig. 10. Vertical acceleration of car body.

Acknowledgment

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Nomenclature

M : Mass matrix
C : Damping matrix
K : Stiffness matrix
K_{ct} : Rail contact vector
$q(t)$: State vector
$d(t)$: Rail input vector
M_{cb} : Mass of a car body
J_{cb} : Moment of inertia of a car body
M_{fb} : Mass of a front bogie
J_{fb} : Moment of inertia of a front bogie
M_{rb} : Mass of a rear bogie
J_{rb} : Moment of inertia of a rear bogie
M_{wi} : i-th Mass of wheelset
C_1 : Primary damping
C_2 : Secondary damping
K_1 : Primary stiffness
K_2 : Secondary stiffness
L_1 : Half length between front bogie and rear bogie
L_2 : Half length between 1st wheelset and 2nd wheelset
L_3 : Half length between 3rd wheelset and 4th wheelset
y_{cb} : Vertical displacement of a car body
θ_{cb} : Pitch angle of a car body
y_{fb} : Vertical displacement of a front bogie
θ_{fb} : Pitch angle of a front bogie
y_{rb} : Vertical displacement of a rear bogie
θ_{rb} : Pitch angle of a rear bogie
y_{wi} : i-th vertical displacement of wheelset
K_r : Rail contact stiffness
d_i : i-th vertical displacement of rail
Y : Fourier response of displacement
H : Transfer function
H_{hes} : Hessian matrix
E : Correction matrix related to gradient change
F : Correction matrix related to search direction
D : Fourier response of rail displacement
v : Velocity of a train
l_i : i-th length between wheelsets
S : Power spectral density (PSD)
$K(f)$: Weighting function in frequency domain
y_{p1} : Vertical displacement of point 1
y_{p2} : Vertical displacement of point 2

References

[1] C. Park, Y. Kim and D. Bae, Sensitivity analysis of suspension characteristics for Korean high speed train, *Journal of Mechanical Science and Technology*, 23 (2009) 938-941.
[2] International Organization for Standardization(ISO), ISO Code 2631-1 (1997).
[3] J. H. Edward and S. A. Jasbir, Applied optimal design: Mechanical and structural systems, McGraw-Hill, USA (1979).
[4] International Union of Railways, UIC Code 513R (1994).
[5] J. H. Edward, Computer aided analysis and optimization of mechanical system dynamics, Springer-Verlag, USA (1984).
[6] B. D. Van Deusen, Analytical technique for designing riding quality into automotive vehicle, SAE Paper 670021, 155-162.
[7] H.-R. Kim and S. Choi, A study on the optimal design of automobile suspension system, *Transactions of the KSME*, A 11 (3) (1987) 437-443.
[8] K. Nishimura, N. C. Perkins and W. Zhang, Suspension dynamics and design optimization of a high railway vehicle, *Proc. of 2004 ASME/IEEE Joint Rail Conference*, Baltimore, Maryland, USA (2004) 129-139.
[9] M. Montgomery, Response surface methodology-process and product optimization using designed experiments, John Wiley & Sons, USA (1995).
[10] Y.-G. KIM, C.-K. PARK, H.-S. HWANG and T.-W. PARK, Design optimization for suspension system of high speed train using neural network, *JSME International Journal*, Series C 46 (2) (2003) 727-735.
[11] J. K. Hedrick, D. N. Wormley, A. K. Kar, W. Murray and W. Baum, Performance limits of rail passenger vehicles: Evaluation and optimization : U.S. DOT, Federal Railroad Administration, Report DOT, RSPA, DPC-50, 79, 32, USA (1979).

[12] V. K. Garg and R. V. Dukkipati, Dynamics fo railway vehicle systems, Academic Press, USA (1984).
[13] R. V. Dukkipati and J. R. Amyot, Computer aided simulation in railway dynamics, Marcel Dekker, USA (1988).
[14] D. J. Inman, Engineering vibration, Prentice Hall, USA (2007).
[15] J. S. Arora, Introduction to optimum design, Academic Press, USA (2004).

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