Model Updating of Flexible Vehicle Body Based on Experiment Modal Parameter

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Abstract. Due to the important contribution of flexible modal in rigid-flexible coupled multiple body system (MBS) dynamics, model updating of flex-body generation became a significant and necessary procedure. In this paper, a model updating method based on the measured frequency response function (FRF) is proposed to modify the flexible body in finite element analysis (FEA). To overcome the complexity of multiple measured responses and the huge cost of computation, the Kriging surrogate model with multi-objective function which considers the FRF's magnitude and shape of curves is proposed. Moreover, the FRF measurement of a high-speed railway carbody is conducted and the measured FRF is transferred into proposed model updating procedure. The results show that the proposed method is performed successfully on a high-speed railway vehicle after model updating and the simulated FRFs tend to coincide with the experimental FRFs. Finally, the updated model is transferred to the rigid-flexible MBS model to validate the influence of the model updating with a comparison of the vibration spectrum and ride comfort index. The results show that the updating procedure will make the prediction of vibration and ride comfort index closer to experimental results.

Keywords: Model updating · Multibody system dynamics · Kriging model · High-speed railway

1 Introduction

During the evaluation of dynamics and comfort of rail vehicles, it's necessary to consider the elastic vibration from the vehicle body, due to the design of rail vehicle body is gradually lightened, especially the high-speed EMUs in China, mostly adopt hollow aluminum profiles in design. Meanwhile, the frequency of the elastic vibration mostly covers a sensitive frequency range of the human body, which reduces the feeling of passengers [1–3]. Moreover, with the rapid development of computational multi-body dynamics, flexible bodies with complex structures could be obtained more quickly and completely from the finite element model, by which the impact from elastic vibration on passenger's feeling could be analyzed. As a result, the elastic vibration gradually attracts more attention during the design and analysis of railway vehicle dynamics, and it is applied in a lot of domain including vehicle dynamics [4, 5], noise and vibration controlling [6, 7], fatigue and wheel-rail wear [8–10].

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With the development of flexible body generation technology, it is probable to import the finite element model with a lot of degrees of freedom into MBS dynamics for rigid-flexible body coupling analysis, especially the promotion of model condensation and substructure model establishment method. But most flexible body generation methods are based on FEA models, which are substituted by beam, shell, volume, and mass elements with a series of omission on geometrics, material parameters and some neglected error, and these cause a significant uncertainty between the simulated model and measured model, in which the latter is often treated as a representation of a realistic vehicle body.

Although these researches improved the accuracy of the simulated model, there are still some problems: the direct utilization of the finite element model for optimization will make convergence speed slow, objective function will be evaluated repeatedly and the cost of calculation will be huge; The optimization process based on the modal information of the structure which of the car-body structure be complex, and the modal information of the structure estimated from the experiment will bring a huge gap between the optimized model and the real model. The model updating algorithm based on the FRFs provides a solution to avoid these problems.

In this article, the finite element model updating method based on acceleration FRF and Kriging metamodel is presented with a high-speed railway carbody. Firstly, a multi-objective function method is proposed, which consists of the response amplitude of the FRF and curve shapes from measured points. Then the Kriging metal model is used instead of the finite element model to avoid complicated calculation and is established by introducing the optimal Latin sampling method after factors and responses corresponding to design variables are normalized respectively. And last a detailed simulation model was replaced by the proposed Kriging model and calibrated with measured data. To validate the performance of the proposed method, a roller-rig modal experiment of a high-speed railway was conducted and the measured acceleration FRF (AFRF) data were in the proposed procedure and calibrated with the corresponding finite element model. What's more, the updated model was transferred to construct a rigid-flexible multibody dynamics model to present the influence of error from flexible body generation by demonstrating the vibration spectrum and the ride comfort index of passengers.

2 Methods

Usually, the finite element model is composed of thousands of elements, which will cost plenty of time during the calculation process, especially for structural dynamic analysis. In the process of model updating, the parameters of the structure are modified repeatedly and it seems impossible and unacceptable if every modification is solved once in finite element analysis. To avoid a lot of iterative calculation, the surrogate model is reliable to replace the finite element model to fit the relationship between the input and response of structure.

The Kriging model, with unbiased estimation at training sample points and highly nonlinear approximation capability, is well suited as a replacement of solving FRFs. Based on the Kriging model, assuming the actual relationship between the system response and the variables can be represented by the following:

$$y(\mathbf{x}) = F(\boldsymbol{\beta}, \mathbf{x}) + z(\mathbf{x}) = \left[f_1(\mathbf{x}) \cdots f_p(\mathbf{x})\right]^{\mathrm{T}} \boldsymbol{\beta} + z(\mathbf{x}) \tag{1}$$

where $z(\mathbf{x})$ is the realization of the stochastic process, $\boldsymbol{f}(\boldsymbol{x})$ is polynomial vector of training sample $\boldsymbol{x}$, and $\boldsymbol{\beta}$ is the coefficient of the linear regression. The $\boldsymbol{f}(\boldsymbol{x})$ term approximates the drift of the Kriging model and $z(\mathbf{x})$ approximates the local deviation of the Kriging mode. $z(\mathbf{x})$ follows normal distribution $\mathrm{N}\left(0, \sigma_{\mathrm{z}}^2\right)$ and has nonzero covariance which estimated by:

$$E[z(\mathbf{x})] = 0 \quad Var[z(\mathbf{x})] = \sigma^2 \tag{2}$$

$$\mathrm{Cov}\left[z(\mathbf{x}_i), z\left(\mathbf{x}_j\right)\right] = \sigma_z^2 R\left(\theta, \mathbf{x}_i, \mathbf{x}_j\right) \tag{3}$$

where $R(\theta, \mathbf{x}_i, \mathbf{x}_j)$ is the correlation function with hyperparameters θ and is a form of the Gaussian correlation function in general:

$$R\left(\theta_k, x_i, x_j\right) = \prod_{k=1}^{n} \exp\left(-\theta_k \left(x_i^k - x_j^k\right)^2\right) = e^{\sum_{k=1}^{n} -\theta_k \left(x_i^k - x_j^k\right)^2} \tag{4}$$

where ${x^k}_i, {x^k}_j$ are the kth elements of samples i and j.

The detailed estimation of θ_{k} and σ_{z} can be found in the reference [xxx], σ_{z} is the function of θ_{k}. In short, the θ_{k} and σ_{z} can be evaluated by solving the maximum likelihood estimated problem which has the form as:

$$\max_{\theta_k > 0}\left(-\frac{m}{2} \ln\left(\hat{\sigma}_z^{*2}\right) - \frac{1}{2} \ln(|\boldsymbol{R}|)\right), k = 1, 2, \ldots, n \tag{5}$$

When the Kriging model is constructed with the obtained θ_{k}, it can be used to predict the output response at untried location $\mathbf{x}_0$ with unbiased estimation. The predicted response is given by:

$$\begin{aligned} \hat{y}(\mathbf{x}) &= \mathbf{r}^{\mathrm{T}} \mathbf{R}^{-1} \mathbf{Y} - \left(\mathbf{F}^{\mathrm{T}} \mathbf{R}^{-1} \mathbf{r} - \mathbf{f}\right)^{\mathrm{T}} \hat{\boldsymbol{\beta}} \\ &= \mathbf{f}^{\mathrm{T}} \hat{\boldsymbol{\beta}} + \mathbf{r}^{\mathrm{T}} \mathbf{R}^{-1} (\mathbf{Y} - \mathbf{F} \hat{\boldsymbol{\beta}}) \end{aligned} \tag{6}$$

Meanwhile, the frequency response of the structure is taken as the research target and then used to establish the objective function based on the distance and shape error of the corresponding FRF curves. To quantify the difference between the simulated and measured AFRF, a correlation function is constructed as follows to describe the similarity between the two curves:

$$C = CSF_{i,j(\omega)} = \frac{2\left|\boldsymbol{H}_{i,j}(\omega)^T \widetilde{\boldsymbol{H}}_{i,j}(\omega)\right|}{\boldsymbol{H}_{i,j}(\omega)^T \boldsymbol{H}_{i,j}(\omega) + \widetilde{\boldsymbol{H}}_{i,j}(\omega)^T \widetilde{\boldsymbol{H}}_{i,j}(\omega)} \tag{7}$$

where $\boldsymbol{H}_i, \boldsymbol{H}_j$ means the transfer function between i-th point and j-th point.

Usually, the calibration is not based on only single AFRF, all the measured point should be considered. For the high speed railway carbody, there are 5 points from each section were used as the objective function, which can be as follows:

$$\begin{aligned} \mathrm{G}(X) &= \min_{X\in R}\left(5-\sum_{i=1}^{5}\lambda_i C_i(X)\right) \\ &= \min_{X\in R}\left(5-\sum_{i=1}^{5}\lambda_i \frac{2\left|\boldsymbol{H}(\omega)^T\,\widetilde{\boldsymbol{H}}(\omega)\right|}{\mathbf{H}(\omega)^T\mathbf{H}(\omega)+\widetilde{\boldsymbol{H}}(\omega)^T\,\widetilde{\boldsymbol{H}}(\omega)}\right) \end{aligned} \tag{8}$$

where $G(x)$ is the final equivalent of the weighted evaluation function, C_{i} is the similarity between the simulation curve and the test curve which is the weight factor to reflect the importance of each objective function, in this study, the weight of the five objective responses is taken as 0.2, the multi-objective optimization problem turns to a problem of minimization $F(x)$.

To avoid the huge cost of computation, the surrogate model is applied to the AFRF data-based model updating. During the calibration process, the established Kriging model is instead of FEM. Model updating result will be achieved based on the optimization results. The main procedure of the AFRF-based the FEA correction including:

First, the frequency response function of each measurement point of the structure should be obtained by impact testing, shaker testing, etc. The points chosen should be representative of the modal shape interested and be checked with coherence.

Secondly, the sensitivity of different variables to the similarity function is introduced to filter the representative variables to maximize the influence of the selected variables on the structure when building the metamodel. These representative variables and modelling errors are also used as input in the DOE analysis. It should be noted that if the selected design variables have different scales, the data should be normalized to avoid convergence or poor performance in the process of building the surrogating model.

Finally, based on the established simulation model, a suitable sampling method such as Latin hypercube should be used to construct the Kriging metamodel, which instead of the simulation model and transferred to the optimization process. By adjusting the main design parameters of the structure, the measured FRFs will consistent with the simulated FRFs. After the generation of the agent model, the accuracy of it should be checked. Then the updated model is used to calibrate the finite element until the updated error is acceptable.

3 Measurement and Validation

The measured railway carbody (Fig. 1) is made up of thin-walled aluminium alloy. The total length of the carbody is 25 m and the cross-section have 3.3 m in width. A modal shaker is set on the ground to excite the vehicle and showed in Fig. 1. On the carbody surface, 5 sections are selected along the length direction and 4 accelerometers are spaced on each cross-section evenly. A burst random signal in the range of 5–50 Hz was used as an excitation on the front wheelset which is naturally periodic and low signal-to-noise ratio (SNR). The measured data were recorded at a sampling frequency of 512 Hz; the sweep speed of the exciter was set to 0.05 Hz/s.

Fig. 1. The measured vehicle body and exciter

The measured vertical acceleration curves are shown in Fig. 3. The major vibration modes of the target vehicle calculated by experimental modal analysis are showed in Table 1. The time domain vibration signal is transformed into the frequency domain using the FFT analysis to obtain AFRF at each measured point and the results are showed in Fig. 3.

To demonstrate the effectiveness of the proposed method, an initial finite element model of the carbody is established in Optistruct (Academic license). The sidewall, roof and floor were modelled with shell elements, and the equipment under chassis is represented by mass elements. For the interior, the solid elements and an isotropic honeycomb material is introduced to simulate the vibration character. The main material parameters of each part are that the Young's modulus, Poisson's ratio and density of carbody are 69000 Mpa, 0.3 and 2700 kg/m^3 respectively. In total 1,310,000 elements and 128772 nodes were used in the initial model then the theoretical modal analysis was performed based on the initial finite element model (Fig. 2).

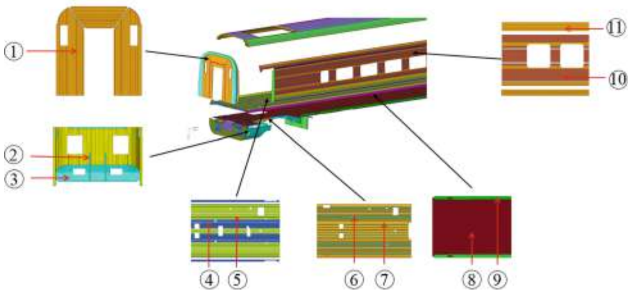

Fig. 2. Selected design variables in the vehicle FEA model

The boundary condition of the carbody is that 4 constraints are attached to air springs under carbody and each stiffness of the springs is set to 203 N/m as it in the measured models. The virtual model is excited by measured signal (along with Z and Y direction)

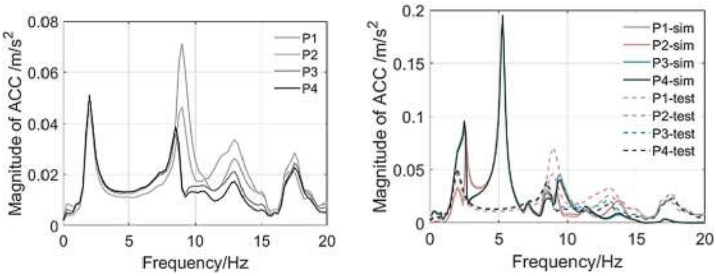

Fig. 3. Comparison of the magnitude of acceleration between the initial model and measured model

which was collected during the measurement at the same location. After the simulation, the analytical AFRF data in the Z direction corresponding to the AFRF can be obtained and presented in Fig. 3.

Then the sensitivity analysis is conducted taking Eq. (8) as target function and all 69 design variables including Young's modulus of the main car body and the interior floor, and the density of the aluminium material are directly treated as the input variables in DOE analysis.

According to the results of sensitivity analysis, there are 11 thickness design variables of the structure are considered and the density of the material and damping ratio which will influence the matrices of stiffness and mass when conducting the DOE analysis.

To calibrate the initial model of carbody, the proposed objective function that satisfies the errors between predicted response and measured during the sensitivity analysis is also introduced in optimization. Model updating result is achieved based on the optimization results. During this process, the scope of input variables is taken the same as the DOE factor bounds. A hybrid method mixed with particle swarm optimization (PSO) and the Hooke-Jeeves approach is introduced to search optimal results.

Due to the error of the establishment, a huge gap occurs in AFRF between the measured and initial model. After the updating procedure, the shape and peaks in each section tend to coincide with measured curves as shown in Fig. 4, which proves that the proposed procedure is efficient.

Besides the updated curves show an agreement with experimental results, the modes of carbody are aligned with experimental results as well. Table 1 lists the modal frequencies of the cabody obtained from the experiment, the initial model, and the updated FE model as well. The differences between the analytically predicted values and the experimentally measured values are also listed. The accuracy of some frequencies increased after updating, comparing with the initial mode's simulation result. For instance, the vertical bending mode's frequency is 14.06 in the calibrated model whose error is 3% less than the initial model's 10%. The relative errors of the frequencies of overall modes after model updating are smaller than 3% except for the vertical bending mode in phase with equipment. It should be mentioned that the vertical bending mode in phase with under chassis equipment is not updated even if the frequency of it is varied due to the stiffness decrease of the carbody. The natural frequency of the lozenging deformation mode is higher than the measurement model and the difference is 1.17 Hz, increased by about 14%.

Table 1. Mode frequency of initial, experiment and updated model

Mode shape	Calibrated/Hz	Initial/Hz	Measured/Hz
Vertical bending in phase with equipment	7.63	7.5	9.01
Lozenging deformation	8.91	7.09	8.26
Torsion	16.56	15.29	16.38
Vertical bending out phase with equipment	14.06	12.93	14.46

Figure 4 shows the difference of the acceleration responses at the centre of the floor and sidewall both in the time and frequency domain with the speed of 300 km/h. For the model with initial flexible body, peaks in vertical direction main focused at 8 and 14Hz while the frequency of the updated model mainly occurs among 12-16Hz. After updating the magnitude of curves drops lower and the elastic vibration occurs around 14 Hz which is close to the frequency of vertical vibration of the carbody. As for the response in the horizontal direction, it inclines when the frequency is over 10 Hz in the frequency domain, comparing with the initial model and rigid body dynamics model.

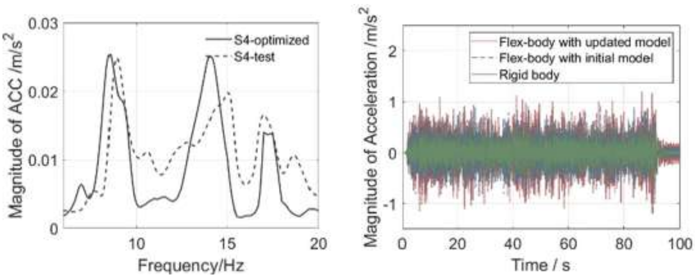

Fig. 4. Acceleration response comparison of center of the floor in the Z direction in frequency and time domain

4 Conclusions

This study was undertaken to propose a method to simplify the FEA model updating process based on the measured acceleration FRF and evaluate the performance of model updating in vehicle dynamics. The Kriging metamodel is introduced into the optimization and updating process by replacing the FEA model. During the establishment of the Kriging model, a multi-objective function method is proposed, which considered the vibration and curve shape of each measured section and a DOE sample method was introduced for reducing the number of training samples. The results of the proposed updating procedure show that there is a significant improvement on the AFRF after updating. So, the proposed method is eligible to update the FEA model of a high-speed railway vehicle.

References

1. UIC Code 513: Guideline for evaluation passenger comfort relation to vibration in railway vehicles. Paris: International Union of Railways (UIC) and European Committee for Standard (CEN) (1994)
2. ISO 8041: Human Response to Vibration - Measuring Instrumentation. Geneva, Switzerland (2005)
3. ISO 2631-1: Mechanical Vibration and Shock e Evaluation of Human Exposure to Whole-body Vibration - Part 1: General Requirements. Geneva, Switzerland (1997)
4. Shiiba, T., Eberhard, P.: Flexible multibody simulation of automotive systems with non-modal model reduction techniques. Veh. Syst. Dyn. **50**(12), 1905–1922 (2012)
5. Jaschinski, A.: Multibody simulation of flexible vehicles in interaction with flexible guideways. Veh. Syst. Dyn. **24**(sup1), 31–44 (1995)
6. Thompson, D.J.: A continuous damped vibration absorber to reduce broad-band wave propagation in beams. J. Sound Vib. **311**(3–5), 824–842 (2008)
7. Zheng, X., Zolotas, A., Goodall, R.: Combined active suspension and structural damping control for suppression of flexible bodied railway vehicle vibration. Veh. Syst. Dyn. **58**(2), 1–31 (2019)
8. Han, J., et al.: High-speed wheel/rail contact determining method with rotating flexible wheelset and validation under wheel polygon excitation. Veh. Syst. Dyn. **56**(8), 1233–1249 (2017)
9. Bezin, Y., et al.: An investigation of sleeper voids using a flexible track model integrated with railway multi-body dynamics. Proc. Inst. Mech. Eng. Part F J. Rail Rapid Transit **223**(6), 597–607 (2009)
10. Ling, L., et al.: Integration of car-body flexibility into train–track coupling system dynamics analysis. Veh. Syst. Dyn. **56**(4), 485–505 (2017)