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High Frequency Vibration Analysis of Container Flat Wagon Base on Rigid-Flexible System

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Introduction

With the development of international intermodal transport, railway container transport becomes more and more important. Higher speed and lighter weight is the development trend of wagon for container transportation. Lightening usually leads to many new problems, for example, global or local stiffness of the car body is not larger enough, which may result in large vertical-vibration for the container flat wagon. The vibration of the car body will become more serious as the car speed increases. Therefore, the flexible vibration model of car body should be taken into consideration. Article [1] used the receptance principles to treat the car body as a free-free beam. The finite element method and modal analysis results were adopted by article [2] to build a flexible car body model and the dynamic response of a passenger car was investigate. And article [3] investigated the dynamic performance of tank car and gondola car by considering flexible body model, and the results showed that the flexible body model is more accurate and reasonable, which was validated by running test.

A new type of container flat wagon (CFW) is investigated by building rigid-flexible body coupling dynamic system. Fig.1 shows a quarter of the CFW body mode. It is known that the long length of bogie center distance and car body load condition will have strong effect on the car body vibration. Larger vertical-vibration acceleration with high frequency appears near the center plate under fully loaded condition. The same phenomenon was founded in double-deck container flat car[4] and 150 ton depressed center flat car. Fig.2 shows the frequency distribution of two sampling points on car body (near the center plate and at the center of CFW body) during a running test. It is seen that the vibration near center plate is larger than the value in car body center and there is a vibration peak near 50Hz for the center plate.

Simulation

The dynamic simulation can be realized by using finite element software-ANSYS and multibody dynamic software-SIMPACK[5,6,7]. Fig.3 shows the flowchart of the simulation.

The car body finite element model is built by shell element with different thickness, and the containers are simulated by mass element that locations at the cen-

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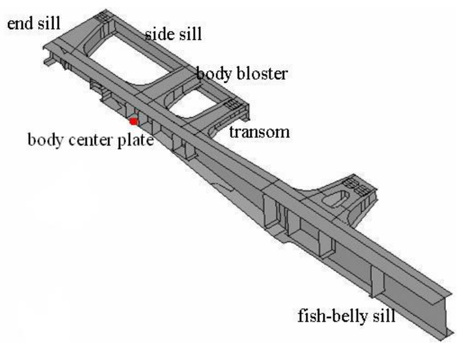

Figure 1: 1/4 CFW mode

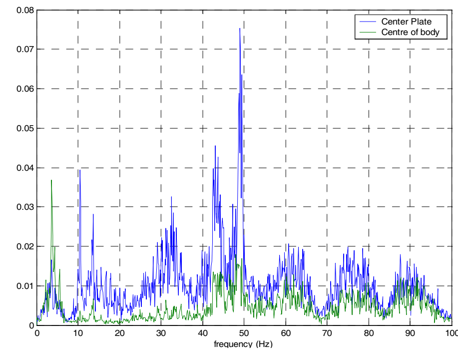

Figure 2: Frequency distributing of vertical acceleration near center plate and center of CFW body, v=100km/h

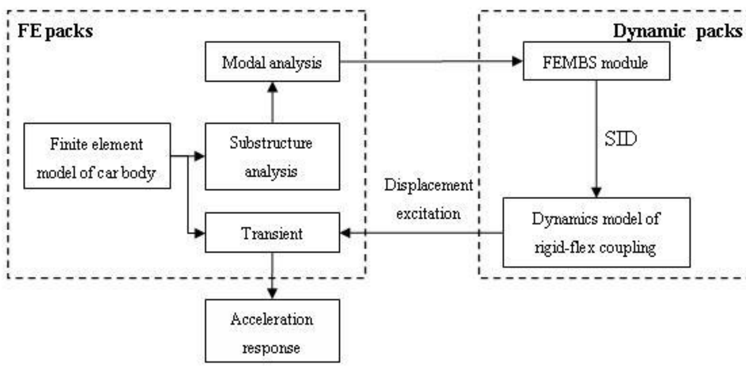

Figure 3: Flow chart of simulation

ter of the containers. By reducing the car body degrees of freedom using Guyan method, 40 lower-order natural modes are calculated by substructure method, among which 5 lower-order vertical bend modes included. Then standard input data (SID) can be obtained by using pre-processor FEMBS of SIMPACK yield. In this paper, the full vehicle dynamic model is set up by taking the wheelsets, frames, bolsters and containers as rigid bodies and the car body as flexible body. Fig.4 shows the topology diagram of the system, and Fig.5 illustrates the rigid-flexible body coupling system.

This type CFW uses three-piece bogie which includes many nonlinear factors, such as dry friction, damping clearance, stop clearance, two-stage stiffness spring. Thus the response of system can't be predicted by transfer functions due to these nonlinear factors. The displacement excitation for the carbody dynamic response calculation using ANSYS is obtained by using SIMPACK in time domain. The stochastic track irregularity is considered, and the car running velocity is 100 km/h.

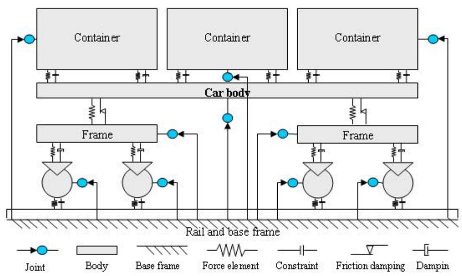

Figure 4: Topology diagram of system

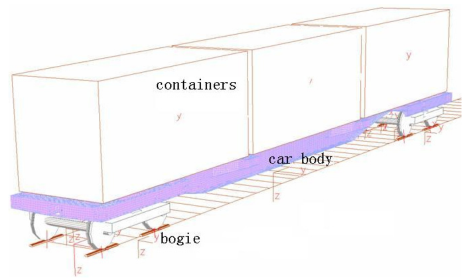

Figure 5: Rigid-flexible body coupling system

Fig.6 shows the displacements in positions of carbody front and rear center plates. In order to reduce the modal truncation error, the dynamic response of car body is calculated by ANSYS with full finite element model.

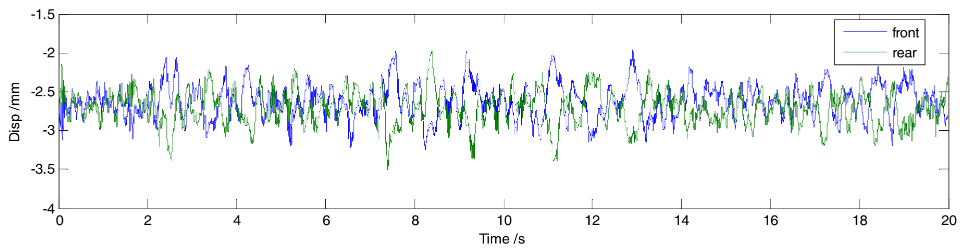

Figure 6: Displacements of car body front and rear center plate

The vertical accelerations in car body center plate and car body center are illustrated in Fig.7 (a) and (b) respectively. It is seen that the acceleration in car body center plate is too larger to be accepted and the maximum value exceeds 1.0g. The acceleration of car body center is much smaller.

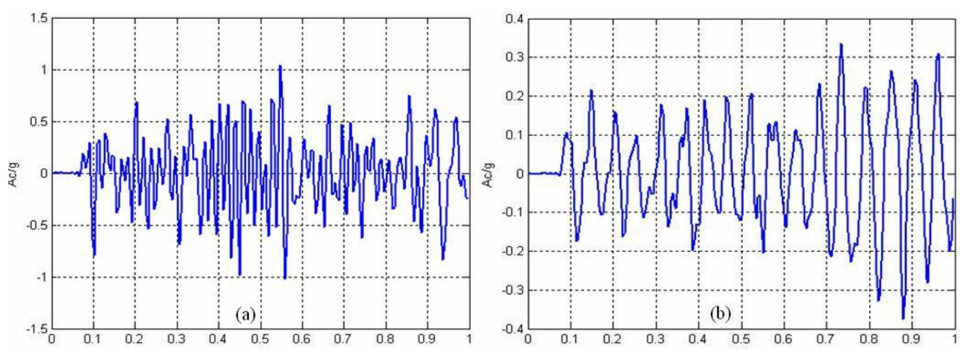

Figure 7: Vertical acceleration (a) center plate (b) center of car body

There are some reasons why the acceleration of car body center plate is very large. The dead weight of the wagon is only 8.9 tons, but the fully load of the container is 72 tons. Therefore, larger elastic deformation of car body near center plate exists. While the local bend stiffness around the middle of car body is large enough due to the fixed container.

In order to investigate the frequency distribution, the acceleration time history results are processed using FFT method. Fig.8 (a) and (b) show the acceleration spectrum of center plate and car body center respectively, which are similar with the running test result shown in Fig.2. Dominant frequency of center plate acceleration is 55Hz, and mostly distributes from 40Hz to 60Hz. It has wide frequency band and complicated frequency composition. The reasons are high order vertical bend vibration and larger dynamic interaction causing larger local elastic deformation. On the other hand, the center of car body has single frequency component, dominant frequency is 18 Hz, which is the first order vertical bending frequency of car body. The fourth order vertical bending frequency is 53Hz approaching dominant frequency of center plate, and center of car body is the node of this order mode, that is why only low frequency appears at the center of car body.

Small dead weight and larger load weight cause the uneven distribution of mass and stiffness, and local rigidity near the center plate is not lager enough. In this case, building rigid-flex coupling system dynamic model is necessary, and it is an availability way to solve the dynamic problems of the vehicle.

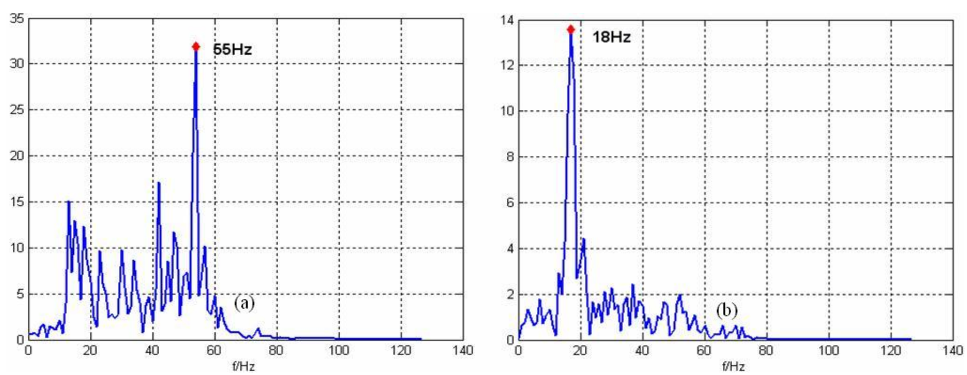

Figure 8: Vertical acceleration spectrum figures (a) center plate (b) center of car body

Improvement Advices

Comparing the simulation result and the test result, it is known that the simulation is reliable. According to the simulations, larger center plate vertical acceleration appears due to the small local rigidity near the center plate, and small frequency of the fourth order vertical bend and low damping ratio. There are two ways to strengthen the local rigidity. One is to add diagonal bracings between corbels and mid-beam, and the other is to increase the steel plate thickness of corbel. At the same time, prolonging the side sills throughout the car body can increase the whole rigidity, and enhance the frequency of the fourth order vertical bend.

Conclusion

Full rigid body model can't reflect the high frequency vibration, so rigid-flexible system dynamic model is necessary. The method used in this paper is FEM codes and MBS codes co-simulation. It is an availability way to solve the dynamic problems of vehicle system. The test phenomenon reappears again in the simulation. The reason of the problem is easy to be found. And many ways can be tried to solve the problem. In order to make the simulation more accurate, the container and car body coupling vibration should be considered in the future.

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