11th World Congress on Computational Mechanics (WCCM XI)
5th European Conference on Computational Mechanics (ECCM V)
6th European Conference on Computational Fluid Dynamics (ECFD VI)
E. Oñate, J. Oliver and A. Huerta (Eds)

FATIGUE ANALYSIS OF STRUCTURE OF GONDOLA CAR BODY BASED ON RIGID-FLEXIBLE COUPLING MULTI-BODY SYSTEMS

ZHONG YU-GUANG*, ZHAN YONG† AND ZHAO GE†

College of Mechanical and Electrical Engineering
Harbin Engineering University
Harbin 150001, China
e-mail: zhongyuguang@126.com

Keywords: *Gondola car body; Finite element analysis; Rigid-flexible coupling multi-body dynamics; Fatigue life.*

Abstract: At present, the traditional fatigue life calculation method on railway gondola mainly uses the measured dynamic load and the analysis results of static strength based on finite element method. The method didn't consider the impact of dynamic loads on the gondola structure and it's predicted results can not be accurate. Furthermore, the measured load spectrum should spend costly human and material resources.

To solve the problem, this paper presents a novel method in researching the fatigue life of the gondola car body structure based on finite element method and multi-body system dynamics theory. Firstly, the multi-rigid body system dynamics model of the gondola car is established considering the wheel-rail contact, nonlinear, orbit incentives and other factors. And then the rigid-flexible coupling system dynamics model is established by using structure analysis method. To obtain the ride quality of the gondola car model, simulation calculation is carried out according to the American five level track spectrum and it can be concluded that the rigid-flexible coupling model is more reasonable than the conditional method. To get the accurate the stress time history of the body structure, on the one hand the load time history of gondola car body can be obtained by simulation analysis in a fatigue cycle sample, on the other hand the corresponding quasi-static stress influence factors are calculated in software ANSYS. Through linear superposition of quasi-static stress influence factors and load time histories, the stress time histories of the body structure are acquired in Fatigue analysis software MSC.FATIGUE based on the quasi-static analysis method. On the basis of the method, the paper gets the whole life nephogram of the body structure. The C80B Gondola Car which is designed in China is taken as an example to verify the proposed method, the results show that the method to predict gondola car body structure based on the hybrid the dynamics simulation and finite element analysis is feasible.

1 INTRODUCTION

With the implementation of the high speed and heavy load policy in the railway freight transportation in China, the increasing using frequency and motivation form the track have heavily impacted on gondola running safety. It is clear that these complex influences can more easily lead to fatigue damage of railway gondola than ever. So, the gondola's fatigue life analysis should be an important part of the design process as well as the main evaluation standard for the new gondola car. Currently the fatigue life prediction of large and complex mechanical structure is to simulate the movement of the actual structure through the finite element analysis, and then use dynamic stress obtained to analyze and calculate fatigue damage of structural[1-4]. However, the fatigue analysis of the railway gondola body yet uses the traditional method of fatigue calculation which mainly uses the measured dynamic load and static strength obtained by finite element analysis to evaluate the fatigue life of dangerous parts. The method does not consider the impact of dynamic loads on the structure and spends a lot of human and material resources to gain actual load spectrum. We can use the existing load spectrum at present as a reference, but it can not be accurate predictions for its limitation [5,6]. With the development of computer technology, virtual prototyping technology is becoming mature and successfully applied in various fields. For the calculations of fatigue life on truck body structure, the predicting method based on computer simulation is becoming more welcome by the users. In the early stages of vehicle design, this method has a great advantage through using virtual simulation experiments to replace actual measured test for its short cycle and high economic efficiency.

2 THE MULTI-BODY DYNAMICS THEORY

In the mechanics modeling of multi-body dynamics, the most common model is the use of general form of kinetic equations based on generalized coordinates and multi-body system dynamics equations derived from Lagrange equations [7]:

$$M\ddot{q} + Kq + C_q^T \lambda = Q_F + Q_v \tag{1}$$

Corresponding constraint equations:

$$C(q,t) = 0 \tag{2}$$

Where: q is a generalized coordinate vector, M is the mass matrix; K is the stiffness matrix, C is the constraint matrix, λ is the Lagrange multiplier matrix, C_q^T is the Jacobian transposed matrix, Q_F is the generalized force corresponds to the active force; Q_V is the generalized force related to the second order item of speed.

The generalized coordinate q is represented by the block matrix of a reference coordinate q_r and elastic coordinate q_f:

$$q = \begin{bmatrix} q_r^T & q_f^T \end{bmatrix}^T \tag{3}$$

The formula (1) and (2) is converted to the form of block matrix:

$$\begin{bmatrix} M_{rr} & M_{rf} \\ M_{fr} & M_{ff} \end{bmatrix} \begin{bmatrix} \ddot{q}_r \\ \ddot{q}_f \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ 0 & K_{ff} \end{bmatrix} \begin{bmatrix} q_r \\ q_f \end{bmatrix} + \begin{bmatrix} C_{qr}^T \\ C_{qf}^T \end{bmatrix} \lambda = \begin{bmatrix} (Q_r)_F \\ (Q_f)_F \end{bmatrix} + \begin{bmatrix} (Q_r)_V \\ (Q_f)_V \end{bmatrix} \tag{4}$$

subject to

$$C\left(q_r, q_f, t\right) = 0 \tag{5}$$

The formula (4) and (5) are multi-body dynamics equations containing flexible body.

The multi-body dynamics equations are derivate from a simple multi-body system and it has been verified that the equations can be also applicable for a gondola car system. In fact, each multi-body system has its special place and it is difficult to be compute by the simple and universal way. Therefore, we should select the appropriate method to solve the problem according to the characteristics of our research object. In the multi-body system models on gondola, since the final analysis objects is the car body, its uniqueness lies in the treatment for the car body, such as flexibility which needs to be analyzed by the finite element. As a result, considering veracity of multi-body dynamics model, we need to attach importance to the research purpose of the system to establish a different multi-body dynamics model.

3 THE DYNAMICS MODEL ON GONDOLA SYSTEM

Based on the multi-body dynamics theory, this paper regards 80t car gondola as object to establish multi-body dynamics model and considers the Car body's flexibility. This paper mainly considers the following four aspects in modeling process: the relationship between wheel-rail contact, nonlinear, orbit incentives and the treatment of the body's flexibility. What is more, this paper analyzes its running quality and improves multi-body dynamics model.

3.1 Wheel-rail contact in the model

Since gondola car running along the track, wheel-rail contact will have influence on the running performance of car model. So, creep forces are calculated by using Hertz contact theory and the simplified theory of Kalker rolling contact under the premise of ensuring the accuracy of the calculation. At the same time, they should be modified by Shen's theory in order to improve the computational efficiency [8].

3.2 Nonlinear relationship in the model

Nonlinear relationship in dynamic model should be mainly caused by gaps and friction between the various connections, such as clearance and friction between the side frame pedestals, nonlinear characteristics of wedge friction damper and multi-level stiffness of central suspension. In the modeling process, this paper establishes the simulation of diffrent stiffness levels considering the segment linearization method on nonlinear components. Besides, appropriate switching speed is used to deal with the nonlinear problems on friction between center plate and side bearings.

3.3 Orbit motivation in the model

The motivation of track irregularity can be obtained from the actual site measurements and also simulated through existing track irregularity power spectral density function. According to the Association of American Railroads (AAR) standards, this paper takes advantage of the five standard track spectrum in the railway vehicle dynamics analysis. In addition, the power spectral density function is converted to the time domain waveforms directly in SIMPACK by using an inverse Fourier transform.

3.4 Flexible body treatment in the model

Because the vibration energy of the vehicle is mostly produced in the low frequency region and the vibration shape is characterized as the typical vibration. So the typical vibration of car body is the most devastating. This means that the low-frequency mode is the main part to need considering, so in this paper the first 8 modes are selected as the calculating base. In the multi-body system, a flexible body is described by the feature information and structure information, which need utilize the finite element method to calculate the modes of flexible body. To save simulation time and consider the restrictions on the numbers of degrees of freedom in SIMPACK, the paper needs to reduce freedom of the flexible body[9]. The common method uses substructure method in ANSYS to process finite element model so as to obtain the reduced modal of car body structure. The results are shown in Table 1.

Table 1: Body structure modes and substructure modes

Order	The frequency of empty car			The frequency of loaded car		
	Before reduction /Hz	After redcution /Hz	Error /%	Before reduction /Hz	After redcution /Hz	Error /%
7	4.4240	4.4197	0.097	9.7057	9.7208	0.155
8	10.652	10.638	0.131	10.725	10.743	0.167
9	13.489	13.483	0.045	14.700	14.700	0
10	17.524	17.581	0.324	15.333	15.386	0.344
11	23.797	23.992	0.813	17.566	17.690	0.701
12	24.808	24.690	0.476	20.710	20.711	0.005
13	25.290	25.747	1.774	20.815	20.816	0.005
14	25.471	26.161	2.637	21.919	21.929	0.046

The results show that substructure method meets modeling requirements of multi-body dynamics within the frequency range of 25Hz. The method not only increases calculation efficiency through reducing degrees of freedom also ensures the characteristics accuracy of the modal.

3.5 Multi-body dynamics model on gondola system

According to the basic theory of multi-body dynamics, the gondola's body, bolster, side frame, wheel are regarded as a rigid body in SIMPACK[10]. And then a multi-body dynamics model gondola (see Fig.1) is established through simulating the hinge and constraint relationships between bodies and simplifying componet style. The model contains: a car body, two bolsters and four side frames, eight inclined wedge, a 80t cargo, eight saddles and four wheelsets (24 rigid bodies). The body has five degrees of freedom, including a yaw, ups and downs, roll, nodding and shaking. Bolster has four degrees of freedom, including yaw, ups and downs, roll and shaking. Side frame has five degrees of freedom, including yaw, ups and downs, roll, nod and shake. Oblique wedge has two degrees of freedom, including yaw and ups and downs. Bearing saddle has one degree of freedom which is nod. Wheelset has four degrees of freedom, including yaw, ups and downs, roll and shake. Meanwhile, in order to record dynamic characteristics of the car body, the 53 sensors are installed on the model of car body.

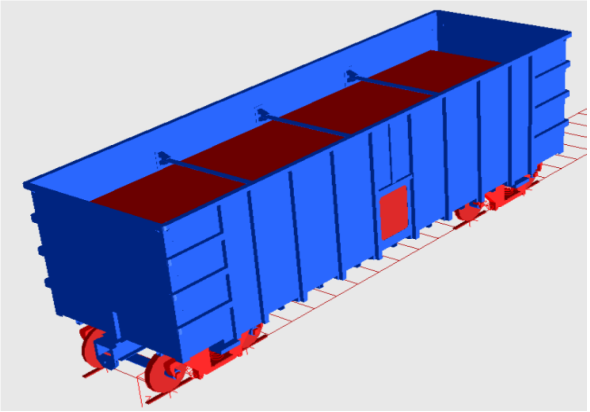

Figure 1 : A simplified multi-body system model for cargo as rigid body

If the gondola body is described as a flexible body, the other part is regarded as a rigid body and the connecting force element characteristics and nonlinear factors are also regarded as rigid body, then the rigid-flexible coupling multi-body systems on loaded car is shown in Figure 2. Similarity, a simplified rigid-flexible coupling system model for cargo as rigid is shown in Figure 3.

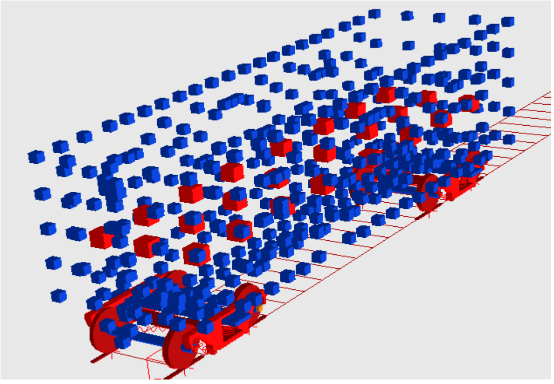

Figure 2: Heavy vehicle rigid-flexible coupling system model

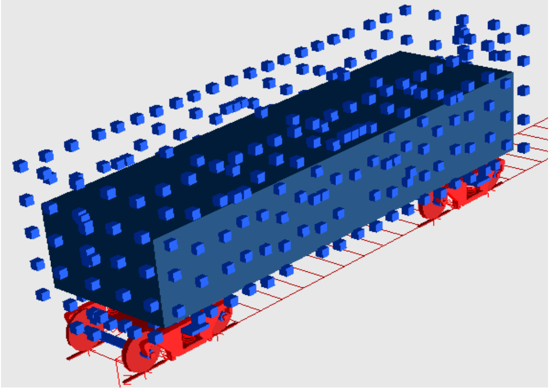

Figure 3: A simplified rigid-flexible coupling system model for cargo as rigid

4. OPERATION PERFORMANCE OF GONDOLA DYNAMICS MODEL

When the vehicle is running, external motivation and forces between wheel and rail have a significant impact on the body structure and its cargo.There are a variety of indicators to measure the operation performance in different countries, such as dynamic load factor, acceleration amplitude of vehicle body and stability of gondola car, etc. In this paper, the main indicators changes from the integrity of ensuring goods to damage problems caused by the vehicle wear and external motivation.More specifically, the main indicator for the study is the peak of acceleration and root mean square [11].

4.1 Dynamic load factor

In SIMPACK, the speed interval is 10km/h and then the speed from 50km/h to 120km/h is divided eight levels. Thus we can get the maximum acceleration amplitude in the vertical and lateral of loaded gondola body for each level. The result is shown in Figure 4.

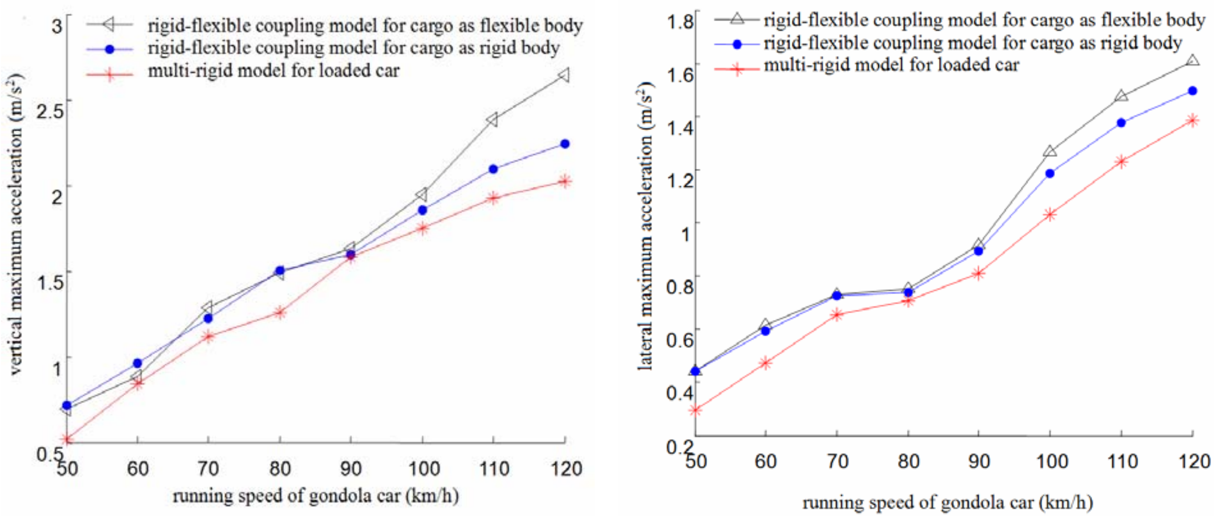

Figure 4: The maximum acceleration of the vehicle body at each speed level

Dynamic load coefficient is divided into vertical and lateral coefficient, and its value can be equivalent to the ratio of largest acceleration of vertical and lateral and gravitational acceleration. According to measure levels of dynamic load coefficient and the calculating result, it can be get some solutions: maximum vertical acceleration amplitude is 2.7 m/s^2 and its expresses that running quality is good, the vertical dynamic load coefficient is 0.275 and it expresses that running quality are good, maximum transverse acceleration amplitude is 1.66 m/s^2 and run quality are satisfied and the transverse dynamic load coefficient is 0.169 and running quality are satisfied.

4.2 Smoothness index

The smoothness index to assess the quality of a vehicle running in railway standards in China is modified from the empirical formula proposed by the German Sperling, and both have the same basic principle. The vehicles running smoothly empirical formula from China national standard GB5599-85 is:

$$W = 7.08 \sqrt[10]{\frac{a^3}{f} F(f)} \tag{6}$$

Where: f is the vibration frequency (Hz); a is the acceleration (g); W is the smoothness index; $F(f)$ represents the weighting coefficients corresponding to vibration frequency.

Sperling empirical formula is proposed based on a large number of single frequency test, but the actual vibration in the running of gondola car is random. Therefor, in calculating smoothness index of the gondola car, acceleration time history should be classified by frequency. And then the smoothness index of entire section can be obtained according to the computation results of every section's. Through the above empirical formula to calculate the gondola car body stability index, the result is shown in Figure 5.

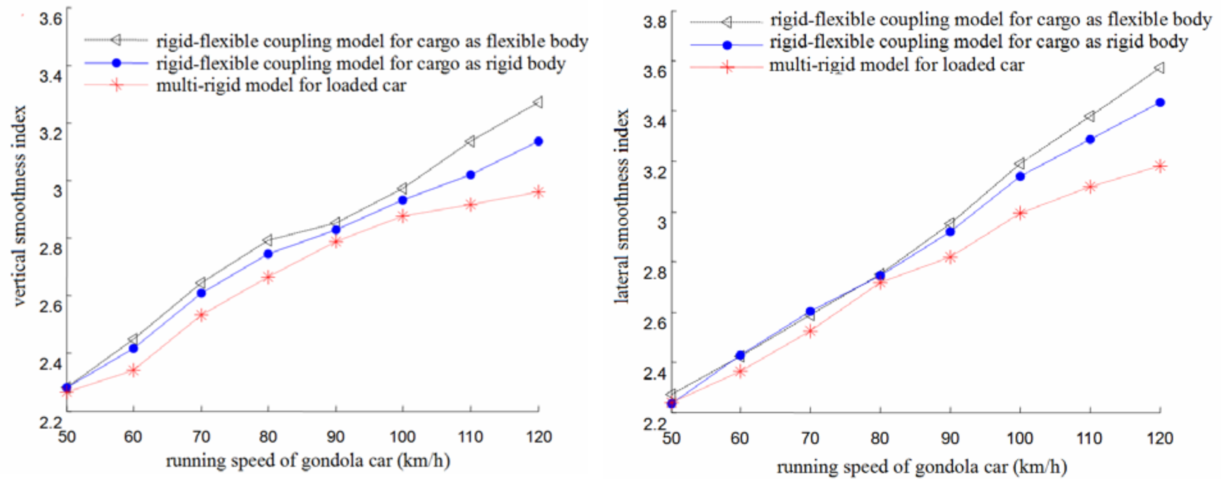

Figure 5 : The stability index of the gondola car body

Through analyzing of the above two figures, we know that the maximum vertical smoothness index is 3.27, and its stability rank belongs to the first rank and meeting the requirements; the maximum lateral index is 3.58, stability rank belongs to the second rank and meeting the requirements. Gondola car body's flexibility is influential in smoothness index and the smoothness index of rigid-flexible coupling body dynamics model should be greater than the model of rigid-flexible coupling multi-body dynamics at the same speed rank. Besides, such difference will increase with the speed.

5. THE FATIGUE LIFE OF CAR BODY STRUCTURE

There are three prerequisites to ensure the fatigue life of gondola car body: the description about the fatigue of materials, the stress time history of the car body parts and the rule of cumulative fatigue damage. On the base of establishing the rigid-flexible coupling multi-body dynamics model of the gondola car, this paper gets the load time history of the gondola car body in the full load conditions through the simulation on the sample line, and calculates stress time history of the car body by using the quasi-static finite element method combining with the related basic theory of the fatigue life. This paper establishes forecast model of the fatigue lift on the car body and forecasts the fatigue life of the gondola car body running on the sample line.

5.1 A simulation sample of route

This paper takes a regional main line in China (a total of 383.92 km) as simulation route, and makes inductive statistics according to the curve radius. Then, the paper needs to choose typical curve radius on behalf of each curve segment, such as R500, R800, R1000, R1600 and R2000. The simulation route is divided into 50 samples, and each curve segment of the sample is distributed according to above setting and the rest is straight line. This paper sets the curve segment length whose radius is 500m as 220m, then the other various typical radius curve segment is calculated as 1009m, 894m, 341m, 192m analogously according to the proportion. So, the paper gets that the length of each sample line is 7692m, including the curve line of 2656m. As a result, such a sample is regarded as a fatigue cycle in order to facilitate calculation of fatigue life.

The limit of speed is specified in the related standards when it passes the curve. Considering that the change of the speed is proportional to the amount of dynamic load in the same external drive and line condition, and dynamic load of a part is a major cause to generate fatigue damage, the design of this sample route is set up according to the maximum running speed of the gondola car.

5.2 The stress time history of the car body structure

The load time history can be calculated by dynamics simulation, when the gondola car body fatigue life runs the above line. The calculation results include the vertical, longitudinal and lateral load time history of both upper center plates and the vertical load time history of the four side bearing from rigid-flexible coupling dynamics model for the heavy car body. Furthermore, they also include vertical, longitudinal and lateral acceleration time history from the simplification model of rigid-flexible coupling dynamics for cargo as rigid. Now we take the half gondola car model as research object, its load time history is shown in the Figure 6.

Stress time history of the gondola car structure can be got by using quasi-static stress method, in other words, stress time history of any point of the gondola car structure is caused by dynamic load of the both upper center plates and dynamic load from cargoes and the four side bearing. The specific expression of the formula is:

$$\sigma(t)=\sum_{i=1}^{n}P_i(t)\frac{\sigma_i}{P_{i,FEA}} \tag{7}$$

Where: $\sigma(t)$ denotes the stacking stress time history of the dangerous parts in the gondola car body, $P_i(t)$ denotes the dynamic load time history corresponding to the i*th* load, σ_i denotes the stress of the dangerous parts caused by the i*th* load, $P_{i,FEA}$ denotes the i*th* static state load defined.

In this paper, the dynamic load time history of the gondola car body is simplified, and the dynamic loads of cargo impacting on end wall, side wall and baseplate are seen approximately as inertia force of cargo under the full load conditions. Then units load with the same direction is used to impact on the parts which is the joint and suffers outside force, in order to get the quasi-static stress influence factor of the car structure by the finite element analysis. Thirdly, this influence factor is used to multiply by the load time history of car body in turn, then it can get the stress time history of the gondola car body. Finally, a damage distribution diagram of the fatigue cycle sample on the car body structure can be obtained through the overlapping calculation of the software MSC. Fatigue combining with the materials properties.

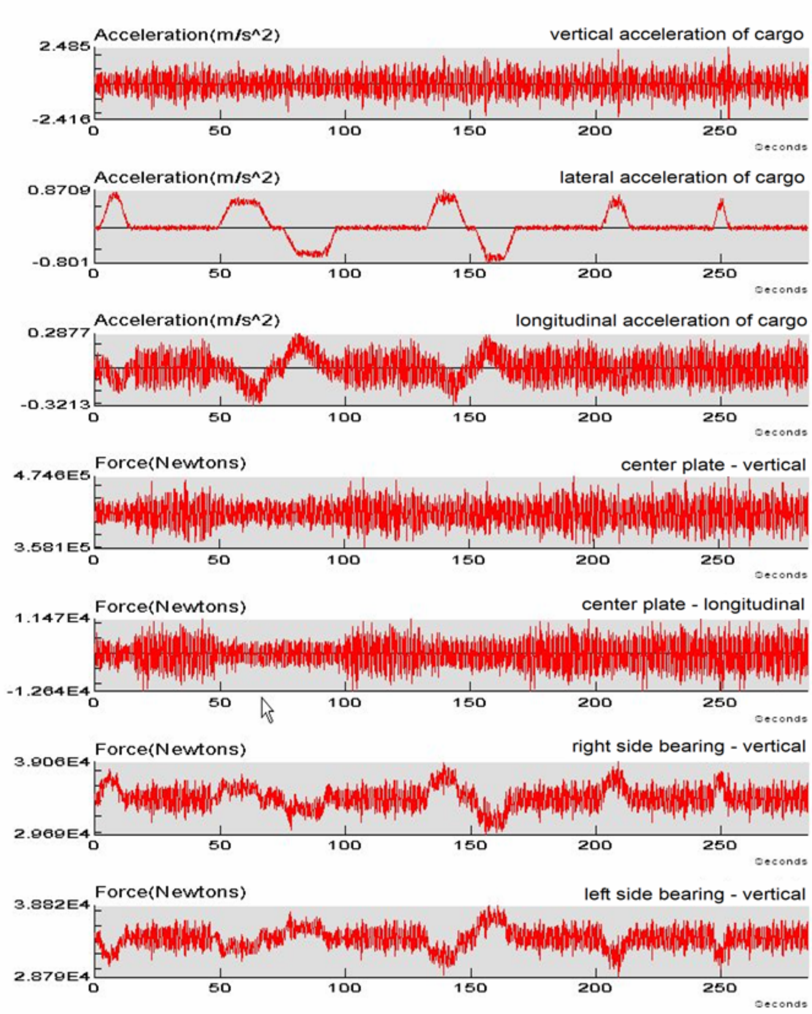

Figure 6: Simulation of load time history in the Sample line

5.3 The fatigue life of car body structure

The structure of gondola car body suffers from the long-term fatigue load, and its maximum stress don't exceed material yield limit in the any working condition. Thus the fatigue life analysis of car body structure belongs to the high-cycle fatigue problem and it should be calculated by the nominal stress method. As a result, the stress fatigue analysis (S-N) is selected in MSC. Fatigue. Because the finite element software ANSYS is seamlessly connected with the fatigue analysis software MSC. Fatigue, the quasi-static stress analysis destination file (*.rst) from the ANSYS is guided into MSC. Fatigue directly. Then the results of the finite element analysis can associate with the load time history achieved above to analyze the fatigue life on the basis of importing the materials properties. The cloud chart of fatigue life analysis is shown in figure 7 .

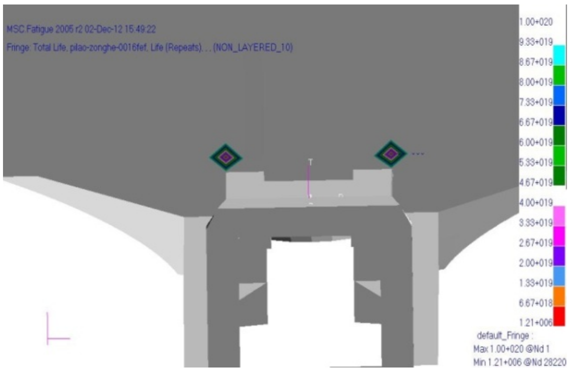

Figure 7: Cloud chart of fatigue life analysis

It can be seen that the minimum cycles which the car body can bear in one fatigue cycle sample is 1.21e6. Its corresponding node number is 28220 and its location is around the junction of the bottom of front-end wall and center sill. Each fatigue cycle length of sample lines is 7692m, so the fatigue life near the node number of 28220 is 9.307 million kilometers.

5.4 The result analysis of the fatigue life

In this paper, a mixed method of multi-body dynamics simulation and finite element quasi-static analysis is used to predict the fatigue life of railway gondola car body. It can be found from the cloud chart that the weak fatigue spot of the gondola car body mainly lies in two parts, the one is the junction of the bottom of front-end wall and center sill and other is the overlap region of bottom panel and center sill. The shortest lifetime of the gondola car body is about 9.31 million kilometers through above calculation. Compared to the 6.86 million kilometers from the referenced literature[6], the result has the same fatigue weak position, but the life value is 35% higher.

The above results shows that the dynamic model still need to be modified so as to improve the calculation precision. Because there is no uniform line load spectrum in China, this paper uses five standard track spectrum in AAR standard which is similar with most of track in China. On the one hand, although this paper seeks to improve the accuracy in the dynamics simulation process of gondola car(for example, the gondola car body is regarded as elastic parts), its results are impossible to be perfect and its error is unavoidable because the gondola car system, especially bogie part, is very complex. On the other hand, this paper mainly considers straight line route and curve line route in the actual running, and does not consider special route such as crossroads or climbing in detail. In the process of considering the acting force between cargo and the gondola car body structure, cargo is simplified as a rigid body and interaction force is ignored among granular material, the handling methods have certain influence on the results. Prediction for fatigue life of the gondola car body is a complex and multidisciplinary problem, there is only simple prediction and a preliminary research in this paper because of the limited time and energy.

6 CONCLUSIONS

According to the theory of multi-body dynamics, the gondola car multi-rigid-body dynamics model is established. On the basis of the model, the rigid-flexible coupling multi-body dynamics model of gondola car is got. Based on the five track spectrum of AAR, this paper simulates the gondola car in typical working conditions and gets acceleration time history curve on the vertical direction and horizontal direction of the car body. The computation result shows that it is according with the evaluation standards of railway wagon in China and the car’s running quality is good.

Simulating a fatigue cycle samples based on the rigid-flexible model of loaded car, the eight load time histories of the gondola car body(including the force and acceleration) are obtained. Furthermore, the corresponding eight quasi-static stress influence factors are also obtained by using the finite element analysis software ANSYS. Based on the quasi-static analysis method, this paper gets stress time history of gondola car body structure with the software MSC. Fatigue. Finally, the full life cloud of car body structure can be getted combining with the fatigue characteristic curve of the material of car body structure. Through the analysis, the minimum cycles that car body can bear is 1.21e6 in a fatigue cycle sample (9.307 million kilometers) and the location is near the junction of bottom of car body front-end wall and center sill. Compared to the gondola car body fatigue life from reference literature[6], the two results have the same fatigue weak position, but the life values differ by 35%.

REFERENCES

[1] Mohammad Reza Ghazavi, Majid Taki.Dynamic simulation of the freight three-piece bogie motion in curve. Vehicle System Dynamics,2008,46 (10):955-973P

[2] Sung II SEO, Choon PARK, Ki Hwan.Fatigue strength evaluation of the aluminum carbody of urban transit unit by large scale dynamic load test. JSME international journal, 2005, 48(1):34-46

[3] Xie Jilong, Zhang Yan, Xie Yunye. Dynamic Response and Fatigue Strength of Depressed Center Flat Frame. Journal of Mechanical Engineering (2010), 46(16):16-22. (in Chinese)

[4] Meng Jin, Zhou Pinghe, Hu Zhigang. Fatigue Life Prediction of Autobody Structure Based on Multr body Dynamics and Finite Element Method. China Journal of Highway and Transport (2010),23(4):113-119 (in Chinese)

[5] Zheng Xiaoyan, Xie Jilong. Compilation and Verification of Fatigue Load Spectrum of Freight Car Body. Railway Locomotive & Car(2008), 28(3):11-14 (in Chinese)

[6] Ran Kun. Study on C80B Body's Load-Stress Transfer Relations and Fatigue Damage. Beijing Jiaotong University, Master's Thesis(2011): 5-87 (in Chinese)

[7] Hong Jiazhen, Liou Zhuyong. Modeling Methods of Rigid- Flexib1e Coupling Dynamics. Journal of Shanghai Jiaotong University(2008), 42(11):1922-1926 (in Chinese)

[8] Ding Guofu, Zhang Weihua, Liou Boxing. Study on Dynamical Simulation of Running Locomotive. Acta Simulata Systematica Sinica(2004),16(8): 1697-1700 (in Chinese)

[9] Wang Wenjing, LI Qiang, Liou Zhiming. Dynamic Stress Simulation of the Flexible Bogie. Chinese Journal of Construction Machinery(2004),2(4):384-389 (in Chinese)

[10] Liao Bingrong, Luo Ren, Wang Zhe. SIMPAKE Dynamics Analysis Advanced Course. Southwest Jiaotong University Press(2010):97-117 (in Chinese)

[11] Chen Lei, Lv Kewei. Research of Ride Quality Assessment of Railway Freight Cars. Journal of the China Railway Society(2006),28(3):111-115 (in Chinese)