An Analysis on Strength of Balancing Shaft Bracket Based on Inertial Release and Submodel

Bao-shan Shen¹, De-bo Chen¹, De-qing Geng¹ and Zheng-wen Yun²,³

¹ Xuzhou XCMG Automobile Manufacturing Co.Ltd,Xuzhou 221004, Jiangsu;
² Jiangsu Xuzhou Construction Machinery Research Institute, Xuzhou Construction Machinery Group, Xuzhou 221004, Jiangsu;
³ State Key Laboratory of Intelligent Manufacturing of Advanced Construction Machinery, Xuzhou Construction Machinery Group, Xuzhou 221004, Jiangsu
E-mail:shenbaoshan12 @163.com

Abstract: A rigid-flexible coupling digital model of whole vehicle is established to reproduce the failure mode of the balance shaft bracket accurately. Then the method of inertia relief and submodel is applied to calculate the contact nonlinear strength analysis in 7 working conditions. Finally, the calculated result was compared with the failure mode from market and test results, and an improvement measure was proposed. The results show that the method can predict the reliable performance of the component quickly and accurately.

1. Preface
The balancing shaft assembly (as shown in Figure 1) is used as the key component of the heavy truck, such as the thrust rod transmits the braking force and driving force of the vehicle, the suspension system as a whole carries the complex forces from the vehicle body and the ground, so it plays an important role in the stability and safety of the vehicle[1-3].

![Figure 1. Structure of the balance shaft in the vehicle](image)

1. Bridge 2. Balance Shaft Support 3. Balance Shaft Shaft Housing 4. Riding Bolt 5. Leaf Spring 6. Rear axle

Poor road conditions and serious overload of heavy-duty dump truck loading make the balance shaft brackets bear a large and complex load. In order to improve the reliability of a 6x4 dump truck's balancing shaft bracket, the authors calculated the strength of the bracket in 7 working conditions and proposed an improvement measure.
2. Principle of inertial release and submodel method

2.1. Principle of inertial release
A free body can withstand a group of external forces that are static and unbalanced, and produce rigid acceleration. If the rate of change of the external force is sufficiently small compared with the rate of change of the natural frequency of the object, the external force and inertial force can be considered to be balanced. Unbalanced forces can be applied during the analysis. If the effect of the inertial force is considered, the condition where inertial force does not occur must be defined, that is "inertia release"[4]. That is to say, the inertial force of the structure is used to balance the external force received. So we don't have to put too much emphasis on boundary conditions.

2.2. Submodel Method Principle
The submodel is based on Saint-Venant's principle, that is, if the actual distributed load is replaced by an equivalent load, the stress and strain change only near the position applied by the load, then the more accurate results can be obtained within the submodel away from the stress concentration position[5]. The method in the finite element analysis is similar to the "parameter method"[6] in beam theory.

3. Establishment of a rigid-flexible coupling vehicle model

3.1. generation of flexible body[7]
The flexible body is generated by the finite element analysis software, and the result of the finite element modal analysis of the component is converted into a modal neutral file (MNF file)[8], which is used as an input file for multi-body modeling.

3.1.1. Establishment of Finite Element Model of Frame Assembly. The shell elements are used for the longitudinal beams, beams, and supports. The tetrahedral element is used for the leaf spring bracket and the rear bracket of the driver's cab. The rbe2+beam element simulation is used for rivets and bolt connections; Use the rbe2 main point as the outer connection point at the connection point between the frame and the front leaf spring, the rear leaf spring and the bridge, the balance shaft support and the thrust rod, frame, and container. The established finite element model is shown in Figure 2.

3.1.2. Establishment of finite element model of leaf spring. An external connection point is established at the installation position of the leaf spring, and the sheet unit contact of the leaf spring is simulated through the gap unit, and the value of the vertical stiffness obtained from the finite element model is kept consistent with the clamping rigidity required by the drawing by adjusting the tangential rigidity of the unit. The finite element model is shown in Figure 3.

Figure 2. Frame assembly finite element model
3.1.3. Output flexible body file. The outline of the frame and leaf spring are drawn by Plot elements and the MNF files are generated by an analytical software. In order to enable the MNF file to be quickly imported into multi-body software, the authors make the files contain the Plot elements and the information of stiffness and quality only.

3.2. The establishment of a rigid-flexible coupling model for complete vehicle
The MNF of the frame and the leaf springs are imported into software and the other components (front and rear axles, tires, transmission systems, shock absorbers, limiting blocks, box, etc.) required and reasonable connections are established based on the assembly relationship of the vehicle, as shown in Figure 4.

3.3. Output the force and torque of outside point

3.3.1. Working condition establishment. According to the design manual[9] and analytical norms to establish the analysis conditions shown in Table 1. The Fz function is used to describe the friction force of each tire under braking conditions, so that the total friction force received by each tire is equal to the inertia force of the whole vehicle, which better reflects the distribution of forces on different tires under different working conditions[10].

| Operating conditions        | Gravity acceleration/g | Note                                |
|-----------------------------|------------------------|-------------------------------------|
| Vertical condition          | 0                      | 0 -3                                |
| Turning condition           | 0                      | 0.3 -1                              |
| Braking condition           | -0.6                   | 0 -1                                |
| Left front wheel elevation  | 0                      | 0 -1                                |
| Right rear wheel elevation  | 0                      | 0 -1                                |
| Raise 0 degrees             | 0                      | 0 -1                                |
| Raise 5 degrees             | 0                      | 0 -1                                |

Figure 4. Rigid-flexible multibody model of complete vehicle
3.3.2. **Output of external force of the frame.** All the conditions are calculated, after comparing the tire wheel load respectively obtained from model calculation and software in the conditions of bending, steering, braking and the other. and output the force and torque of the frame outside the joint. Due to the large amount of output, it is not listed here.

4. **Calculation of static strength**

The method of inertial release and submodel is used in this paper to solve the problem that the method of inertial release cannot be used for contact nonlinear calculation.

4.1. **Calculation of globe model**

The force and torque calculated above are applied to the external connection points of the frame (including the balance shaft bracket) model. Then the model of frame strength is calculated by the inertial release method which is used as a global model.

4.2. **Calculation of submodel**

4.2.1. **Determining the boundary of the sub model.** Node-based submodel technique is applied in this paper, which means that the displacement of the boundary node of the submodel is obtained by interpolation of the displacement result of the global model node. In order to ensure the correctness of the submodel, by comparing the displacement and stress cloud diagram near the boundary of the global model and the submodel, the nodes on the left and right longitudinal beam section are selected as the submodel boundary, as shown in figure 5.

4.2.2. **Displacement actuation and strength calculation.** The contact relationships between balance shaft bracket and frame were established, after imposing displacement driving to the boundary of the sub model, then calculating the sub model.

5. **Calculation results verification and improvement scheme**

5.1. **Comparison of failure modes**

In the calculation results of the strength of the 7 working conditions, the value of the stress in vertical condition is maximum. The stress cloud charts and the position of actual failure are shown in figure 6, 7.

The figure above shows that the high stress zone of the component is damaged first under vertical impact conditions, and then gradually extends to the entire crack, and the calculation is in good agreement with the actual failure mode.
5.2. Improvement scheme

In order to minimize the cost of the mould caused by structural changes, the authors carried out the strength analysis of the five existing structures. The balancing shaft bracket with connecting rod structure is better, the structure and stress cloud chart as shown in figure 8, 9.

The vertical condition is still the worst, but the maximum principal stress of the bracket is less than the tensile strength of QT500. When the new structure is verified by real vehicle, the improvement is obvious.

6. Conclusion

Through the multi-body analysis of rigid-flexible coupling and the finite element analysis of inertial release and submodel method, the contact nonlinear strength calculation of the balance shaft bracket is carried out in the vehicle, which solves the problem that the inertial release method cannot make contact nonlinear, and greatly improves the calculation accuracy. At the same time, it avoids the problem of high computing resources and long computing time caused by contact nonlinear calculation of the finite element model of the whole vehicle including the balance shaft bracket (small mesh size and two order tetrahedral mesh). Therefore, it is better to reproduce the failure mode of components, and has a good application value to predict the performance of the bracket.

Acknowledgment

This work was partially supported by the Fund Project: National Science and Technology Support Plan (2015BAF07B02).

Reference

[1] Li jie, Zhang zhe and Zhu yijie 2011 Establishment and Simulation of Balanced Suspension Leaf Spring Model Journal of Chongqing University 34 (6) 31-35.
[2] Zhang junrong, Li jianlin and Deng yong 2008 Strength Design of Thrust Bar for 40t Heavy-duty Truck Balanced Suspension *Automotive Technology* 3 19-22.

[3] Zhang jianzhen, Chang lianxia and Ma wensong 2009 Finite Element Analysis of Failure Mode and Effect of Balanced Suspension *Automotive Technology* 10 9-12.

[4] Zhang yongchang 2004 *The Theoretical Foundation and Application of MSC Nastran Finite Element Analysis* Beijing: Science Press pp 337~339.

[5] Zhou yu 2008 Sub-model based fine calculation of railway vehicle structural strength Dalian: *Dalian Jiaotong University* 9~10.

[6] Chen tieyun and Chen bozhen 1991 *Ship Structure Mechanics* Shanghai: Shanghai Jiaotong University Press.

[7] Li chulin 2008 *Hyperworks Analysis Application Example* Beijing: Machinery Industry Press pp 3~48.

[8] Jiang yingchun 2008 Finite Element Analysis of Automobile Suspension Based on Rigid and Flexible Coupling Hefei: *Hefei University of Technology* 22~32.

[9] Automotive Engineering Handbook Editorial Board 2001 *Automotive Engineering Handbook Design articles* Beijing: People's Communications Press pp 97~117.

[10] Shen baoshan, Wang yiwen and Lu yongneng 2016 Analysis of Frame Strength of New Energy Freight Cars Based on Rigid and Flexible Coupling *Automotive Engineer* 12 31-33.