Strength Analysis of Torsion Beam of a MPV vehicle Based on ADAMS

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Abstract. This paper is based on HyperMesh and MotionView to establish a flexible body file of the rear torsion beam of a MPV vehicle, and then according to the vehicle parameters, bushing, shock absorber and spring and other parts properties and hard point information, the flexible file of torsion beam is input into ADAMS to establish the vehicle dynamic suspension model after rigid and flexible coupling; according to the load transfer, the ADAMS software is used to calculate the force at each connecting point of torsion beam under different working conditions, and the force at each point is added to the finite element model of torsion beam to analyze the stress distribution of torsion beam under each working condition. According to the post-processing cloud diagram, the risk points of torsion beam under various working conditions are found, which provides the direction and train of thought for the structural optimization of torsion beam in the later stage.

1. Introduction
As an important part of vehicle chassis, suspension plays an important role in driving comfort and handling stability. Torsion beam suspension is widely used in rear suspension system because of its simple structure, easy disassembly, less space and less mass under the spring. The rear torsion beam is an important bearing part in the rear suspension system, which needs to bear the force and torque transmitted by the road surface through the wheel, spring, limit block and shock absorber, and is easily broken. It is necessary to analyze and verify the strength of the back torsion beam in the design stage to ensure that the torsion beam has enough strength.

Traditional strength analysis and evaluation method of automobile parts is bench test. Although this method is the most direct and effective one, it has a long test cycle and high cost[1]. In case of strength failure, it is necessary to re-design changes and re-perform bench test. Therefore, domestic and foreign automobile manufacturing enterprises adopt the method of computer virtual simulation to analyze the strength of parts, the strength analysis of the key lies in the parts of the accuracy of the joint point load, but the torsion beam stress distribution is complex, it is difficult to get the joint load directly from the experiment, so the connection point load of the beam can only be extracted by the method based on multi-body dynamics. The accuracy of the multibody dynamics model directly affects the accuracy of the results, so the establishment of a precise multibody dynamics model of torsion beam becomes the key factor[2].

2. Establishment of Finite Element Model of Torsion Beam
Three-dimensional CATIA digital model is introduced into the finite element analysis software HyperMesh, and the torsion beam is divided into tetrahedral meshes according to the specified grid quality standard. The weld seam is simulated by the thick grid between the components. The torsion
beam is connected with other components by RBE2 rigid connection unit, and the plate-shell Shell element is used to mesh the main parts of the torsion beam, and 29239 elements are obtained; Each component is given material properties (as shown in Figure 2) and the finite element model of the torsion beam is shown in Figure 1. It includes components such as cross member, longitudinal arm, longitudinal arm reinforcement, spring tray, cushion block support, shock absorber support, flange support, flange, bushing, wire harness support, etc.

3. Dynamics Model Building
The precision of dynamic model is the main factor affecting load extraction. To establish an accurate vehicle dynamic model, it is necessary to consider that the highly deformed components can not simplify the rigid body in the dynamic model, and that the flexible body is used instead of the rigid body to do the rigid-flexible coupling vehicle dynamic model[3].

3.1. Flexible body file for torsion beam
The torsion beam not only has anti roll, but also anti braking nod. In the actual motion, the torsion beam has a large amount of deformation, so the torsion beam can not be simply reduced to rigid body, so it is necessary to improve the accuracy of the dynamic model by flexible torsion beam, and the analysis results are more in line with the reality.

This paper uses the finite element analysis software to discretize the torsion beam into small meshes (the finite element model generated in chapter 2). MotionView software is used for modal calculation to generate the first 15 order modal neutral file MNF (modal neutral file), which is directly read into the ADAMS to build the flexible body, as shown in figure 3.
3.2. Dynamic model of torsion beam in Adams/Car
The dynamic model of the torsion beam of a MPV vehicle is built in the Adams/Car based on the hard point, the suspension system parameters of the rear torsion beam, the distribution information of the axle load, the connection relation of each component and so on, and will be generated. A file is imported into the Adams/car software instead of rigid body torsion beams, and at each connection point to establish interface part as a rigid body and flexible body transition parts, and finally after assembly and debugging into rigid flexible coupling after the torsion beam suspension system, as shown in figure 4.

4. Load determination
Static load analysis of the suspension system ADAMS simulate the force situation of the suspension system under specific working conditions, and provide boundary conditions for component strength analysis. In order to simulate the actual working conditions of the vehicle, considering the comprehensiveness of the analysis, combined with the actual working conditions of the whole vehicle, the analytical working conditions are formulated.

4.1. Development of analytical working conditions
In the process of acceleration, deceleration and steering, the axle load will be transferred, so it is necessary to solve the input load at the tire according to the axle load transfer in the course of vehicle motion.

4.1.1 Bumpy conditions.
Simulation of vehicle bumps: when bumps occur, the wheels are impacted by the vertical load of the road surface, and the vertical force of the left and right tire joints can be obtained by the formula of the force on the ground point of the tire:

\[ F_z = \frac{W}{2} \times 3G \]  

(1)

4.1.2 Braking conditions.
Simulated vehicle braking: when braking occurs, the tire is subjected to the longitudinal resistance of the ground, under the action of inertial force, the axle load is suspended and transferred forward, and the tire ground point force has the formula (2) to obtain the longitudinal force and the vertical force of
the left and right tire connection place:

\[
\begin{align*}
F_x &= -(W/2 - \Delta W/2) \times a_x \\
F_z &= (W/2 - \Delta W/2) \times G \\
\Delta W &= (M \times \frac{a_x}{G} \times H) / T
\end{align*}
\]

(2)

4.1.3 Acceleration conditions.
Simulation of vehicle acceleration: when acceleration occurs, the tire is driven by the ground longitudinal force, under the action of inertia force, the axle load is suspended backward, the tire grounding point force formula (3) can be obtained left and right tire wheel center position force:

\[
\begin{align*}
F_x &= -(W/2 + \Delta W/2) \times a_x \\
F_z &= (W/2 + \Delta W/2) \times G \\
\Delta W &= (M \times \frac{a_x}{G} \times H) / T
\end{align*}
\]

(3)

4.1.4 Right turn.
Simulation of vehicle steering: when the steering occurs, the tire is subjected to the lateral force of the ground to the left, and under the action of inertial force, the axial load is transferred to the left wheel, and the force on the ground point of the tire is obtained by formula (4),(5):

Location of left tyre

\[
\begin{align*}
F_{\text{lefty}} &= (W/2 + \Delta W) \times a_y \\
F_{\text{leftz}} &= (W/2 + \Delta W) \times G \\
\Delta W &= (M \times \frac{a_y}{G} \times H) / L
\end{align*}
\]

(4)

Right tire ground

\[
\begin{align*}
F_{\text{righty}} &= (W/2 - \Delta W) \times a_y \\
F_{\text{rightz}} &= (W/2 - \Delta W) \times G \\
\Delta W &= (M \times \frac{a_y}{G} \times H) / L
\end{align*}
\]

(5)

In the formula: W full load quality of vehicle, \(\Delta W\) axle load transfer, \(a_x\) for longitudinal acceleration, \(a_y\) to the lateral acceleration, H vehicle full load center of mass height, the T is wheelbase, and the L is wheeltrack.

The dynamic analysis is shown in Table 1 below.

| Serial number | Dynamic analysis          | Acceleration |
|--------------|---------------------------|--------------|
|              |                           | X | Y | Z  |
| 1            | Bumpy conditions          | 0 | 0 | 3g |
| 2            | Brake conditions          | -1g| 0 | 0  |
4.2. Output load
The established vehicle dynamic model is used to analyze the above conditions, and the loads at each stressed point of the torsion beam (as shown in Fig. 5) are extracted as boundary constraints of the finite element analysis model. The load at each connection point of the torsion beam (body to torsion beam mounting point, hub to torsion beam mounting point, shock absorber to torsion beam mounting point, spring to torsion beam mounting point, eight points in total due to left-right symmetry) is simulated and analyzed and output.

|   | Accelerated conditions | 0.5g | 0   | 0   |
|---|------------------------|------|-----|-----|
| 4 | Right turn             | 0    | 0.7g| 0   |

5. Finite Element Simulation Analysis
The loads extracted under typical limit conditions are input into the HyperMesh finite element model as boundary conditions of the finite element model[4], as shown in Figure 6.

The force values extracted from the dynamic analysis are loaded into the finite element model and the stress of the torsion beam under each condition is calculated by using Nastran solver. The stress cloud diagram is shown in Fig. 7 to Fig. 10.
As can be seen in the cloud diagram of the calculation results, the bumpy conditions Maximum stress 529.5 MPa; maximum stress 238.8 MPa during braking condition; maximum stress 168.8 MPa during acceleration condition; maximum stress 335.9 MPa at steering condition. It can be found from the yield strength table that the maximum stress of the cross member under rough conditions does not satisfy the yield strength of the material. It is recommended to select the appropriate cross member material or increase the thickness of the cross member[5].

The maximum stress is 495.9 MPa when the thickness of the cross member is optimized to 3mm and then the torsion beam is simulated under the bump condition, which is obviously better than the simulation results before optimization; as shown in Figure 11.

Figure 11. Optimized bumpy conditions

6. Results and discussion
Taking the rear torsion beam of a MPV vehicle as an example, the strength analysis of the torsion beam is carried out by using ADAMS, HyperWorks and MotionView to simulate the four typical
working condition limits of the whole vehicle, and the strength analysis is carried out. The concrete conclusions are as follows:

1) Under the condition of bumping, the maximum stress appears at the beam, and the risk point under the condition of torsion beam is found, which plays an important reference value for the optimization and development of torsion beam.

2) The flexible torsion beam in ADAMS is established by HyperWorks and Motion View, which makes the model more accurate and the simulation results more credible.

3) It is of great significance to shorten the development cycle, reduce the development cost, reduce the number of tests and ensure the quality of products by using simulation software analysis.

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