Analysis and Research of Stiffness Based on Body-in-White NVH

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Abstract: In this paper, the BIW of a car are analyzed by the torsional stiffness and bending stiffness and based on the mechanical model of the stiffness, which determined the boundary conditions and constraint loads. Consequently, the CAE model is established and analyzed of the stiffness calculation and experiment validation. It find out the resonance point that caused by the lack of rigidity of the car body to improve the local stiffness and achieve the goal of advancing the NVH performance of the whole car.

1. Introduction

The automobile body is the main bearing object in the movement process. Due to its numerous component parts, complex structure and complex working conditions, the main working conditions include driving inertia force, braking inertia force, turning inertia force, uneven road surface reaction force and the engine assembly load of different position[1-2]. For example, the weight of the driver and the passenger causes the body floor to bear gravity, the torque generated by the gravity of the side plate and the random load produced by the uneven pavement cause the body to turn around. In the design of modern car body structure, if the stiffness of the car body is insufficient, the vibration frequency of the body will cause the structural resonance, weakening the connection strength of the structural joints, and eventually causing a large deformation of doors, windows, door frames, trunk opening, engine hood, etc. Finally, it caused by the death of the door card, the broken glass, the tight seal, the leakage of the air, the leakage of rain, interior decoration and so on. Therefore, it is very important to analyze the stiffness of the car body, improve the body structure design, and improve the body stiffness analysis and research[3].

In this paper, the mechanical analysis of the bending stiffness and torsional stiffness of a certain type of vehicle under static load is carried out, and a finite element model is established by setting a certain boundary constraint form and loading method under the actual load of the vehicle body. Through CAE calculation and analysis and stiffness test verification, the resonance point of the vehicle body caused by insufficient rigidity of the vehicle body was found, and the design requirements for improving the local stiffness of the vehicle body were achieved.

2. Mechanical Analysis of Body Stiffness

2.1. Mechanical model of body torsional stiffness

When the front body of the vehicle encounters uneven road conditions, the front and left suspension brackets generate torque M to the vehicle body. In order to simulate the actual working conditions, the
force condition of the vehicle body is simplified. The three translational degrees of freedom are 0, the
two translational degrees of freedom at the left and right suspension of the front wheel are 0, and
the front wheel has only the freedom of movement in the z direction. At the same time, the rotational
freedom of all suspensions is limited. The body torque M is reduced to two equal and opposite forces
F acting on the left front suspension and the right front suspension. The mechanical model is shown as
Figure 1[4].

\[
\alpha = \arctan \frac{z_1 - z_2}{L}
\]

(1)

The white body is simplified as a beam structure, and its torsional stiffness KT is

\[
K_T = \frac{F_1 \times L_1 - F_2 \times L_2}{d\alpha/dx}
\]

(2)

Where G is the shear elastic modulus of the beam, and Ip (I=πd4/32) is the polar moment of inertia.

2.2. Mechanical Model of Bending Rigidity of Vehicles

According to the bending mechanics of material mechanics, the mechanical model of bending stiffness
at this time is simplified to a simple beam with a certain change in cross-sectional area (as shown in
Fig. 2)[5-7], and the part fixed at the front and rear suspensions of the body-in-white are simplified to a
fixed hinge and a living hinge support, a concentrated load in the Z direction is applied at positions
where the front suspension and the rear suspension cross the longitudinal beam.

First, the static force method is applied to obtain the binding force R1 and R2 of beams at A and C,
as shown in Figure 2, from the static equilibrium formula.
The moment equation is used to establish the bending moment equation of the beam.

\[
\begin{align*}
M_1(x) &= \frac{F_b}{l} x, \quad 0 \leq x \leq a \\
M_2(x) &= \frac{F_b}{l} x - F(x - a), \quad a \leq x \leq l
\end{align*}
\]  

The second-order derivative of beam bending moment is obtained the deflection equation of body

\[
\begin{align*}
\frac{d^2w}{dx^2} &= -\frac{F_b}{l} x, \quad 0 \leq x \leq a \\
\frac{d^2w}{dx^2} &= -\frac{F_b}{l} x + F(x - a), \quad a \leq x \leq l
\end{align*}
\]  

Formula (5) is integrated to obtain the deflection and corner equation of the body

\[
\begin{align*}
EI\theta_1 &= -\frac{F_b}{2l} x^2 + C_1 \\
EI\theta_2 &= -\frac{F_b}{2l} x^2 + \frac{F}{2}(x - a)^2 + C_2 \\
EIw_1 &= -\frac{F_b}{6l} x^3 + C_1 x + D_1 \\
EIw_2 &= -\frac{F_b}{6l} x^3 + \frac{F}{6}(x - a)^3 + C_2 x + D_2
\end{align*}
\]  

Among them, \(C_1, C_2, D_1,\) and \(D_2\) are integral constants. Because the deflection at the two positions \(A\) and \(C\) should be zero and the axis of the beam after bending should be a continuous smooth curve, the deflection and corner at the junction between the AB segment and the BC segment must be equal to each other.

\[
\begin{align*}
\text{When } x = 0, & \quad w_1 = 0 \\
\text{When } x = l, & \quad w_2 = 0 \\
\text{When } x = a, & \quad w_1 = w_2, \quad \theta_1 = \theta_2
\end{align*}
\]  

The continuous condition of formula (8) is substituted into formula (6) and (7)

\[
\begin{align*}
D_1 &= D_2 = 0 \\
C_1 &= C_2 = \frac{F_b(b^2 - l^2)}{6l}
\end{align*}
\]  

The corner and deflection equation of body is obtained from equation (9)

\[
\begin{align*}
\theta_1(x) &= -\frac{F_b}{6EI} (x^2 - b^2 + l^2) \\
\theta_2(x) &= -\frac{F}{6EI} [3x^2 - 3l(x - a)^2 + b(b^2 - l^2)]
\end{align*}
\]
$\begin{align*}
  w(x) &= -\frac{Fb}{6EI} \left( x^3 - b^2 + l^2 \right) \\
  w_2(x) &= -\frac{F}{6EI} \left[ bx^2 - l(x-a)^3 + b(b^2 - l^2)x \right] 
\end{align*}$

(11)

Vt-vo: E is the elastic modulus of the beam; I (I=πd⁴/64) is the moment of inertia of the solid circular beam.

3. Finite element analysis of the stiffness of the body in white

3.1. CAE calculation and analysis of body torsion stiffness

A finite element model was built based on the analysis of the torsion mechanics of the body-in-white. The model has a total of 524,187 units, in which the triangle unit accounts for 4% of the 2D unit and the FE model (including solder joints) weighs 301.1 kg. The boundary conditions are the translational freedom of the front bumper midpoint Z, the freedom of translation of the leaf spring mounting points X, Y, and Z, and the application of a torque of 2000 Nm to the front left and right damper seats. In the center of the left and right front suspension support points, an equal-sized direction is applied along the Z direction and the opposite and concentrated load is 1850 N (as shown in Fig. 3). The body's torsional stiffness displacement cloud diagram (as shown in Fig. 4) is obtained by Hypermesh analysis. It can be seen that the maximum deformation is 3.844mm, and the maximum torsion angle of the left and right front suspension of the body is given by formula (1).

$$\alpha_{max} = \arctan\left(\frac{Z_1 - Z_2}{L}\right) = 0.244^\circ$$

The maximum torsional stiffness is given by formula (2).

$$K_{\alpha_{max}} = \frac{F_1 \times L_1 - F_2 \times L_2}{d\alpha/dx} = 8928.57 N \cdot m/deg$$

Under this operating condition, the full-load finite element analysis was performed to obtain the stress cloud diagram of the vehicle body (as shown in Fig. 5). The maximum stress was 195 MPa, which satisfies the maximum torsional stiffness requirement and meets the requirements of the body structure design.
3.2. CAE calculation and analysis of body bending stiffness

The boundary condition of the bending stiffness of the body of the white body is the Y, Z translational freedom of the front and right damper seats before the restraint, and the plate spring installation points X, Y, Z translational freedom, and the position of the front and rear suspension and the longitudinal beam in the position of the front and rear suspension and the Z axis negative to the 1500N force (as shown in Figure 6).

The flexural stiffness displacement map (as shown in Fig.7) is obtained from the body bending finite element model analysis. It can be seen that the maximum bending deflection \( w_{\text{max}} \) at the concentrated load is 3.125 mm. At this time, the bending stiffness of the body is given by formula (11).

\[
EI = - \frac{F_b}{6w_{\text{max}}^2} (x^3 - b^2 + l^2) = \sum_{w_{\text{max}}}^F = 4800 \text{ N/mm}
\]

![Fig.7 Bending stiffness displacement map](image)

![Fig.8 Bending stiffness stress diagram](image)

The bending stress of the body shows that the maximum stress is 1087 MPa and the longitudinal distribution curve of the body is smooth, which means that the overall bending stiffness of the body meets the basic requirements.

4. White Body Rigidity Test Validation

The installation of the test vehicle is based on the size of the test vehicle to make the front suspension bracket. The front suspension bracket is fixed on the front suspension limit block reinforcement board, and the degree of freedom in each direction of X, Y, Z needs to be restrained. Fasten the front suspension bracket bolts during installation and adjust the height of the front suspension bracket until it is level. The spiral screw structure is adopted to achieve height adjustment. The built-in displacement sensor and force sensor are installed on the rear suspension bracket by means of rotating the screw structure to achieve torque loading. Displacement sensor is arranged in the lower part of the body longitudinal beam every 200mm, the range is adjusted to the middle, and a total of 13 measuring points are arranged. At the same time, the sensor is installed on the front windscreen, front and rear door openings and the back door opening, respectively, and number them, and finally draw the bending deformation curve through the X coordinate value and Z-direction displacement of these measurement points.

4.1. Torsional test of body stiffness

Torsional clockwise and counterclockwise torques are applied to the front suspension damper and the body joint, respectively, and each load is divided into four times, respectively 1020 N·m, 2040 N·m, 3060 N·m, 4080 N·m. Torque loading is performed by manual loading, and the force of each front suspension loading bracket is recorded, and the distance from the suspension fixing point is accumulated to obtain the applied torque. Before each load, record the readings of each sensor in the wind window. After the torque is loaded to the corresponding value, the readings of each sensor of the wind window are recorded at the same time. The deformation of each measurement point is obtained and the white body deformation and torsion angle test results are obtained. And use charts to describe the corresponding relationship between the loading torque and the torsion angle of each measurement.
point, the change of the torsion angle of each measurement point, and the calculation result of the relative torsion angle between the axes (as shown in Figure 10, 11).

From Fig.10, the experimental torsional stiffness $KT=(15070.1 + 15826.0)/2= 15448 \text{ N}\cdot\text{m/deg}$ is greater than the calculated torsional stiffness value $9925.57 \text{ N}\cdot\text{m/deg}$. At a torque of $2040 \text{ N}\cdot\text{m}$, the maximum torsion angle at different measurement points is $0.156$ less than the calculated torsion angle of $0.224$, which indicates that the design of the vehicle body structure has been tested to verify the need for torsional stiffness.

4.2. Body stiffness bending test

After commissioning and installation of the measuring equipment and sensors, use the crossbar to connect the left and right installation points in the engine installation position, and the engine simulation load is applied to the crossbeam. As shown in Figure 11. Ensure the balance of the front seat load during the loading process. In the process of exerting the load, the vibration of the vehicle body is avoided as far as possible, so as to ensure the stability of the measured value.

The bending stiffness test was performed a total of three times. The first time is the pre-experiment, the purpose of the pre-experiment is to eliminate the influence of the installation gap, and the second and third times are the formal tests[10]. Before the experiment, the sensor is cleared and the displacement sensor reading is recorded. The test process and steps are as follows:

Working condition 1: loading engine load $1500 \text{ N}$, wait for one minute after loading, record force value sensor and displacement sensor readings. The total load is $1500 \text{ N}$ at this time.

Working condition 2: Load the front seat load to $3000\text{N}$, load the rear seat four-occupant load $6000\text{N}$, wait for one minute after loading, record the force sensor and displacement sensor readings. The total load at this time is $1500 + 3000 + 6000 = 10500 \text{ N}$;

Working condition 3: Load the mid-seat three-occupant load $4500 \text{ N}$ and record the displacement sensor readings. The total load at this time is $1500 + 3000 + 4500 + 6000 = 15000 \text{ N}$;

Working condition 4: Unload the rear seats with a load of $6000 \text{ N}$. After unloading, wait for one minute and record the force sensor and displacement sensor readings. The total load at this time is $1500 + 3000 + 4500 = 9000 \text{ N}$;

Three formal tests are repeated in the upper order, and the bending stiffness displacement curve of the body is drawn according to the average value of the two test, and the bending stiffness of the body is calculated according to the average maximum deformation amount (as shown in Figure 12).

The maximum actual bending stiffness of the test $KB = F/\Delta Z = 6000 / 0.363 = 16534 \text{ N/mm}$ is greater than the calculated flexural rigidity value of $4800\text{ N/mm}$. When the body is loaded $1500 \text{ N}$, the maximum deformation of the bending stiffness is $0.156 \text{ mm}$, which is less than $3.125$ of the calculated...
value of CAE, which meets the basic requirements.

5. Conclusion
From the test and CAE analysis results, the torsion stiffness calculation value is less than the actual test torsion stiffness value, the torsion angles of different measuring points are all less than the calculated torsion angles. Although the deformation of the main opening is small, the diagonal deformation of the front windshield and back door is still larger than 0.1%, which indicates that the corresponding four joint stiffness is weak, and the stiffeners should be added here.

In flexural condition, according to the test results of flexural rigidity and flexural rigidity, the numerical results are in good agreement.

The test results verify the correctness of the CAE calculation results, which provides a theoretical basis for the optimization of the rear body structure stiffness. The test method has played a very important engineering significance for the study of the body stiffness.

project name:
1) Research and application of automotive NVH based on the BIW (XDJK2014C004);
2) Study of high Precision Portable temperature detector test device (Qian Kehe LKB[2013]22 number);
3) Analysis and Application of auto body stiffness based on the NVH (Qian Kehe LH[2014]7527 number);
4) Design of power system for pure electric vehicle based on Advisor (Qian Kehe LH[2014]7526 number).
5) Joint Foundation Project of Guizhou Province, Bijie City and Guizhou University of Engineering Science (LH[2017]7012)
6) Joint Foundation Project of Guizhou Province, Bijie City and Guizhou University of Engineering Science (LH[2015]7583)

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