Chapter 44
Fatigue Analysis of Car Body Structure Based on Transient Response

Pengbo Wang

Abstract Fatigue simulation is executed for a car body. Loading spectra are collected by real-vehicle test on enhanced road surfaces, and the load history for each interface point is acquired by using multibody dynamics virtual iteration. Based on a finite element model and the mode superposition method, transient response of the car body is simulated. And the stress history is obtained using the modal stress recovery technique. The S-N method is applied to predict fatigue damage distribution of the full car body. In the fatigue analysis, material Haigh diagrams are defined to introduce the mean stress influence and the critical cutting plane method is applied to treat the multi-axial stress. Based on the presented numerical scheme, fatigue analysis and structural improvement for the car body can be carried out at the earlier stage of the vehicle development project.

Keywords Car body · Road loading spectra · Transient response · Mode superposition method · Fatigue analysis

44.1 Introduction

Automotive body is the main load bearing part of the vehicle, which supplies a necessary space for the occupants and packages, and acts as an installation base for the powertrain and the suspension system. Automotive body bears various dynamic loads,thus its fatigue resistance is important [1, 2]. The fatigue life of the automotive body can be measured by durability experiment for a certain number of body samples. But the durability experiment, which consume much money and time,can be executed only after the prototyping bodies are manufactured, and the

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experimental conclusions may be affected by various accidental factors. While using modern computer simulation technologies, the car body life can be estimated before the durability test to reduce the huge cost of prototyping manufacturing and physical sample experiment.

In automobile travel process, the vehicle body bears multiple loads including the vertical load from road surface roughness, the lateral load from steering or lateral wind and the longitudinal load from acceleration or brake. Those loads are too complex to be calculated with theoretical method and can only be measured by physical or virtual experiments.

In conventional body fatigue simulation, firstly acceleration signal or force signal is collected at wheel center locations, secondly a full vehicle multi-body dynamic model is build to implement virtual iteration, in which the dynamic interface forces of the body are determined by decomposing the wheel center signal, thirdly the finite element method (FEM) is applied to obtain the dynamic stress distribution of the body, and lastly fatigue analysis is executed to get the fatigue life or damage value of the vehicle body [3].

Though virtual road and numerical vehicle provide a method to simulate automobile driving and get the load history [4, 5], the real road experiments with physical prototype are in much better accordance with the actual situation,so real vehicle road test is still the most universal method for present automobile companies to obtain road loading spectra.

The finite element analysis (FEA) on automotive body dynamic stress can adopt quasi-static method or transient method. The former applies a unit static load on the FEM model to replace the actual dynamic load, and executes a static analysis to get the stress field, which is the stress affecting factor for the corresponding load component. Each stress affecting factor multiply the corresponding load history, and the sum of all the products is the stress history of the body structure [6, 7]. Because the quasi-static method ignores the effect of loading frequency and is inadequate when structure resonance occurs due to external excitation at certain frequencies, it is only suitable for the auto parts whose natural frequency is much higher than the loading frequency [8]. Usually a body structure has a lowest natural frequency on the order of tens of Hertz, and therefore its dynamic stress should be calculated by using the transient method.

Research on the body stress in actual driving process proves that the structure stresses are below the material yield limit in most cases. So the body fatigue life can be calculated with high-cycle fatigue assumption [5].

This paper measures the wheel-center acceleration signal of a car with real vehicle road test, obtains the time-domain load data at the body interface points by multibody virtual iteration, calculates the stress history of the car body structure by using transient FEA, and executes fatigue simulation based on the S-N method to estimate the fatigue life and identify the high-risk positions.

44.2 Road Load Data Measurement and Interface Force Generation

44.2.1 Real Vehicle Road Test

The wheel-center acceleration signal of a concept car is measured on 11 enhanced test roads as shown in Fig. 44.1, which includes stone block road, gravel road, cobble stone road et al. The measurement does not require that the test vehicle completely conforms to the new designed model,and it just need to modify a similar reference vehicle to ensure that its rear and front axle loads, wheelbase, track width and chassis hard points are close to the design values. Therefore, the road test can be done at the concept design phase of the automobile developing process.

The whole road test work involves test scheme determination, preparation of parts, measurement on the proving ground, data check and data processing, and finally supply the road load data which will be used in a subsequent virtual iteration to derive interface forces and moments between the body and the chassis parts [3].

Data processing of the original signal from road test is a necessary work before the virtual iteration. Operations such as peak removal, drift compensation, translation and filtering are completed using professional software to correct the measurement errors, then the road load data are appropriately split and combined according to different roads and vehicle durability test requests.

44.2.2 Virtual Iteration

Because the road test can directly measure the wheel-center acceleration rather than the body interface point loads, a multibody model must be built to derive the interface forces and moments from the wheel-center signals. If the wheel-center acceleration signal is directly applied on the multibody model, the dynamic problem will become unsolvable because of unbalanced force system. Before interface force generation, the wheel-center displacement should be calculated by virtual iteration according to the measured wheel-center acceleration by virtual iteration.

Fig. 44.1 Enhanced test roads on the proving ground

The principle of virtual iteration is shown in Fig. 44.2. The multibody model of the whole vehicle acts as a signal input and output system, for which the wheel-center displacement A and acceleration U are respectively the input and output signals. A frequency spectrum analysis of the output signal with applying white noise as input signal provides a transfer function $H(f)$, which means the ratio of output to input under sinusoidal excitation at different frequency. By the transfer function's reciprocal $1/H(f)$, the input can be reversely solved from the output. Because the whole vehicle multibody model is a nonlinear system but the transfer function is based on linearization treatment, the input signal is solved iteratively to make the calculated output signal approach the tested output signal, and finally a relatively accurate solution of input signal is obtained.

The transfer function is defined as

$$H(f) = A(f)/U(f) \tag{44.1}$$

By Fourier transform, the measured time-domain signal of wheel-center acceleration from the road test is converted to frequency-domain signal $A_{\mathrm{m}}(f)$, from which the initial trial solution of wheel-center displacement $U_0(f)$ is derived using the transfer function $H(f)$.

$$U_0(f) = A_{\mathrm{m}}(f)/H(f) \tag{44.2}$$

The trial solution $U_0(f)$ is applied to the multibody model as input signal,and the wheel-center acceleration $A_0(f)$ is obtained as output response signal. The calculated output $A_0(f)$ is compared with $A_{\mathrm{m}}(f)$. If the former agrees well with the latter, $A_0(f)$ is the final solution of the wheel-center displacement,and can be used to derive the body interface loads for the subsequent fatigue analysis. If there is a mismatch between them, the first iteration is executed to obtain an updated wheel-center displacement solution.

$$U_1(f) = U_0(f) + (A_{\mathrm{m}}(f) - A_0(f))/H(f) \tag{44.3}$$

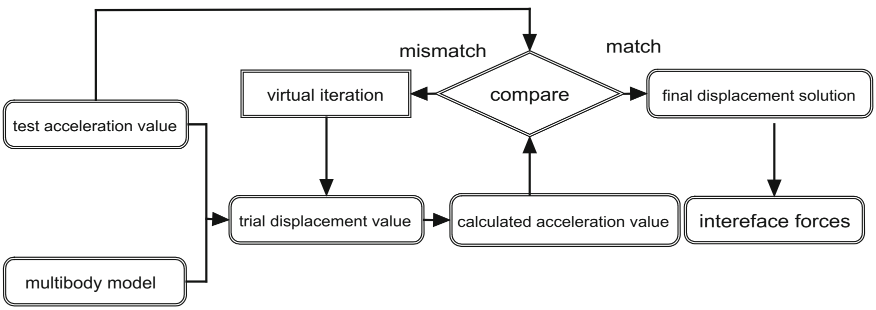

Fig. 44.2 Multibody virtual iteration

The updated trial solution $U_1(f)$ is applied to the multibody model to get a new output response $A_1(f)$, which is also compared with $A_{\rm m}(f)$. The above procedure is executed iteratively until the calculated output response is in good agreement with the tested signal.

The comparison between the calculated results and the tested data involves time-history signal comparison and power spectral density (PSD) comparison. Usually determining when the iteration can be over depends on subjective evaluation. If the calculated results are acceptable, the virtual iteration process is ended, and the final solution of the wheel-center displacement is input to the multibody model to generate interface loads between the body and the chassis parts. The time history of the interface loads in global coordinate system will be used in the subsequent body FEM analysis and fatigue simulation.

44.3 Transient Dynamic FEA

44.3.1 BIW Finite Element Model

After generating the body interface loads, a finite element model of the body in white (BIW) as shown in Fig. 44.3 is built and a transient dynamic analysis is performed to obtain the stress spatial distribution and time history.

Shell elements is used to simulate the BIW, where 4-node elements occupy a major percentage and 3-node elements account for less than 5%. The whole model contains 1,897,845 nodes and 1,866,466 elements. In driving process, any mass added to the BIW will influence the dynamic response of the whole structure, so

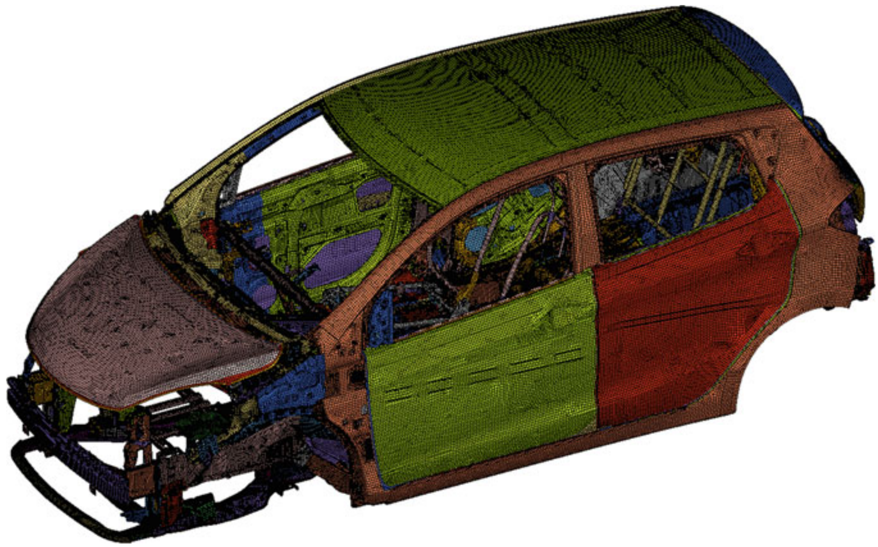

Fig. 44.3 Finite element model of the BIW

distributed masses or concentrated masses are built and connected to the BIW model to simulate the interior and exterior trim parts, luggage and passengers before the dynamic analysis.

44.3.2 Transient Analysis Based on Mode Superposition

Two different numerical schemes can be used in transient dynamic FEA: the direct integration method and the mode superposition method [9]. Since the mode superposition method uses structural mode shapes to reduce the solution space and uncouple the equations of motion, usually it is more efficient than the direct integration method for a problem involving a large-scale model and a large number of time steps. So this paper adopts the mode superposition method in transient analysis.

The transient analysis based on mode superposition is a natural extension of the conventional modal analysis.

As a first step, transform the variables from physical coordinates to modal coordinates by

$$\{\mathbf{u}\} = [\Phi]\{\xi\} \tag{44.4}$$

where $\{\mathbf{u}\}$ is the displacement vector of all the nodes, $[\Phi]$ is a matrix consisting of mode shapes and $\{\xi\}$ is the vector of modal coordinates. Equation (44.4) represents an equality if all modes are used; however, because all modes are rarely used, the equation usually represents an approximation.

To proceed, ignoring the damping and write the equation of motion as

$$[\mathbf{M}]\{\ddot{u}\} + [\mathbf{K}]\{\mathbf{u}\} = \{\mathbf{p}\} \tag{44.5}$$

where $[\mathbf{M}]$, $[\mathbf{K}]$ and $\{\mathbf{p}\}$ are respectively the mass matrix, stiffness matrix and external load vector.

Substitute Eq. (44.4) into Eq. (44.5) and multiply by $[\Phi^T]$, resulting in

$$[\Phi^T][\mathbf{M}][\Phi]\{\ddot{\xi}\} + [\Phi^T][\mathbf{K}][\Phi]\{\xi\} = [\Phi^T]\{\mathbf{p}\} \tag{44.6}$$

where $[\Phi^T][\mathbf{K}][\Phi]$ and $[\Phi^T]\{\mathbf{p}\}$ are respectively the modal mass matrix, modal stiffness matrix and modal force vector.

Since the generalized mass and stiffness matrices are diagonal matrices, the modal equations of motion are uncoupled and can be written as a set of single degree-of-freedom system equations as

$$m_i\ddot{\xi}_i(t) + k_i\xi_i(t) = p_i(t) \tag{44.7}$$

where m_i, k_i and p_i are respectively the i-th modal mass, modal stiffness and modal force.

Once the single degree-of-freedom equations are solved, individual modal responses are computed by solving the above single degree-of-freedom equations, nodal displacement responses are recovered as the summation of the modal responses by using Eq. (44.4). Then the stress and strain responses are derived from nodal displacements.

If the stress time history of the whole BIW is output to the result file, which means the full-field stress is output at each time step, this will make the result file too large to treat. Under conventional computation conditions, usually the local stress output and fatigue analysis are executed for only the pre-estimated high-risk areas of the BIW [10].

In order to complete a fatigue analysis of the whole BIW structure,this paper does not directly output the stress history in the transient dynamic FEA, but use modal stress recovery to calculate the dynamic stress [11]. The FEA supplies only the modal stresses and modal coordinates, and the stress time history of the whole BIW is calculated as the summation of the modal responses in the following fatigue analysis, as Eq. (44.8).

$$\{\sigma(t)\} = \sum_{i=1} \xi_i(t)\{\sigma_i\} \tag{44.8}$$

where $\{\sigma_i\}$ is the modal stress of the i-th modal shape.

44.4 Fatigue Analysis of the BIW

44.4.1 S-N Curve Method for Low-Cycle Fatigue

The FEA results shows that the dynamic stresses of the BIW are lower than the corresponding material yield limits, which means that it should be consider as a low-cycle problem. Therefore, the S-N curve method, also known as the stress-life method is selected to solve the fatigue problem. An S-N curve for a material defines alternating stress value versus the number of cycle required to cause failure.

The S-N curve method is based on uni-axial fatigue theories. However, because the BIW bears random loads when driving, the amplitude and direction of each principal stress are both changing with the external loads. Therefore, equivalent stress hypotheses are employed to assess the multi-axial stresses with the aid of a uni-axial reference stress.

The conventional equivalent stress hypotheses, such as the maximum shear strain energy criterion and the maximum principal stress hypothesis, whose application is limited in proportional loading, are not applicable for complex random loading. In order to allow a fatigue analysis for non-proportional loading cases, the cutting plane method is applied. It transforms the multi-axial stress to an equivalent stress at the crisis cutting plane with a modified maximum shear strain

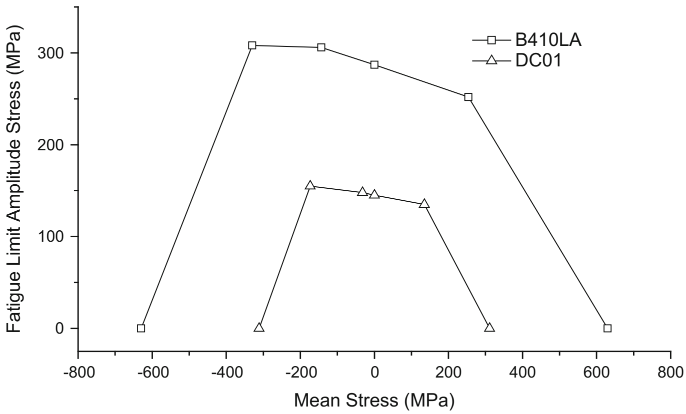

Fig. 44.4 Example of Haigh diagram

energy criterion, and then the uni-axial fatigue theories can be employed to obtain the fatigue damage and life [12].

Usually the S-N curves are defined by means of fully reversible cyclic loading, which means the mean value of the alternating stress is zero. In most actual cases, the mean stress of the BIW is non-zero. For the fatigue life, a mean tension stress is beneficial and a mean compression stress is harmful. The mean stress influence is taken into consideration in this paper with the aid of the Haigh diagram, which defines the relationship between the fatigue strength limit amplitude and the mean stress value, as shown in Fig. 44.4.

In addition, fatigue life of the structure is also affected by stress gradient, surface roughness and technological surface treatment. The influence of these factors can be considered by properly modifying the S-N curve.

An S-N curve is derived with constant-amplitude loading cycles. However, the BIW bearing randomly changing stress histories, for which the division and amplitudes of loading cycles are difficult to determine. In this paper the loading cycles is counted by means of the rain flow counting method, which converts a complex variable amplitude loading history to a series of simple constant-amplitude loads.

44.4.2 Miner Linear Damage Accumulation

The Miner linear damage accumulation rule is widely used in current automobile industry to calculate structural fatigue damage. The Miner rule can be formulated as

$$\mathrm{D} = \sum_{i=1}^{l} D_i = \sum_{i=1}^{l} \frac{n_i}{N_i} \tag{44.9}$$

where n_i is the actual number of cycles with the stress amplitude S_{ai} and mean stress S_{mi} in the loading history, and N_i is the corresponding number of cycles required to cause failure.

The damage contribution of a single stress cycle is given as the reciprocal of $N_i(1/N_i)$, and the total damage D is the summation of the individual damage contribution induced by each single loading cycle.

When the total damage $D = 1$ is reached, the structure reaches its life limit and failure occurs.

44.4.3 Results of Fatigue Analysis

The fatigue analysis simulates a 7500 km durability test on 11 enhanced roads of the proving ground, the numbers of travel repetition for different roads range from 90 to 270.

On the basis of the modal stresses and modal coordinate responses given by the transient dynamic FEA, the fatigue analysis calculates the stress time history of the whole structure and then obtains the damage value for a single travel on each road. The accumulation of the individual damage values gives the total damage of the complete road test, which is shown in Fig. 44.5.

The result indicates that the maximum damage occurs at the rear floor but its value 0.048 is far less than 1.0, which demonstrates excellent anti-fatigue performance of the new designed car body.

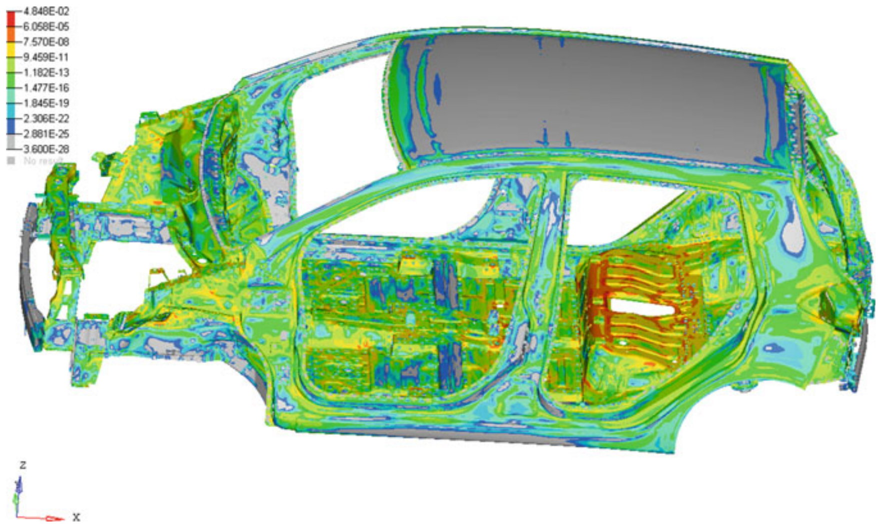

Fig. 44.5 Damage contour of the BIW

44.5 Conclusions

This paper proposed a numerical scheme of vibration fatigue simulation for vehicle bodies, which covers measurement of road load data, interface force generation by virtual iteration, transient dynamic FEA, and fatigue analysis based on stress-life method. Using the presented scheme, fatigue damage analysis, high-risk position identification and structural improvement can be carried out for the designed car body at the earlier stage of the vehicle development project.

The road load data are measured by real vehicle road test that provides more reliable data than virtual road test with numerical vehicle. The stress history of the BIW is calculated by transient dynamic FEA, which can consider the effect of loading frequency and obtain more accurate results than the quasi-static method.

The stress time history is calculated by modal stress recovery method, which avoids outputting a huge result file of BIW stress in the transient FEA, and make it practicable to complete a fatigue analysis for a whole BIW structure under multiple load cases.

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