

Fatigue Life Estimation of Front Subframe of a Passenger Car Based on Modal Stress Recovery Method

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ABSTRACT

In this paper, the dynamic stress of the front subframe of a passenger car was obtained using modal stress recovery method to estimate the fatigue life. A finite element model of the subframe was created and its accuracy was checked by modal test in a free hanging state. Furthermore, the whole vehicle rigid-flexible coupling model of the passenger car was built up while taking into account the flexibility of the subframe. Meanwhile, the road test data was used to verify the validity of the dynamic model. On this basis, the modal displacement time histories of the subframe were calculated by a dynamic simulation on virtual proving ground consisting of Belgian blocks, cobblestone road and washboard road. By combining the modal displacement time histories with modal stress tensors getting from normal mode analysis, the dynamic stress time histories of the subframe were obtained through modal stress recovery method. According to the Palmgren-Miner hypothesis and $S-N$ curve, fatigue life of the subframe was assessed, while the Goodman method was employed as a mean stress correction method. Additionally, an attachment bracket of the subframe was modified to extend the fatigue life of the subframe.

CITATION: Wang, D. and Jiang, R., "Fatigue Life Estimation of Front Subframe of a Passenger Car Based on Modal Stress Recovery Method," *SAE Int. J. Mater. Manf.* 8(3):2015, doi:10.4271/2015-01-0547.

INTRODUCTION

The front subframe is an important part of passenger car chassis system, which has a significant influence on vehicle properties like ride comfort, handling stability, NVH performance and safety. As a main load-bearing component, the front subframe of a passenger car has to bear and transfer various forces and torques caused by the road surface roughness and engine vibrations while traveling over a rough road [1,2]. As a result, these random loads may lead to fatigue failure of the subframe during the vehicle service life. Therefore, fatigue life prediction of the subframe is one of the key requirements in design and development process of passenger cars [3]. Moreover, the fatigue strength of automotive components has been traditionally evaluated by conducting physical tests on actual road, which is time-consuming and needs loads of resources. In order to overcome these disadvantages, virtual durability tests are employed to investigate the fatigue durability of vehicle components by using computer simulation [4,5,6,7].

A significant amount of effort and research on the analysis and prediction of fatigue failure of vehicle components have been done. Petracconi et al. [8] in their research on fatigue life of a rear tow hook pin assembly of a passenger car conducted a comparison between fatigue tests and computer simulations. It was found that the computational methodology was accurate enough to make good estimates of the rear tow hook assembly failure. Sathish et al. [9] evaluated the fatigue damage of a truck cabin caused by proving ground excitation through linear superposition method with static results from finite element model and load histories from multi body

simulation. The virtual results were compared with physical test results, which indicated that the virtual fatigue damage locations were matched well with physical test. Jing et al. [10] predicted the fatigue life of a vehicle driving axle housing under random load based on the stress results of finite element analysis by means of software MSC. Nastran and the random loading history obtained by multi-body dynamic analysis using software SIMPACK. The damaged positions obtained through numerical simulation were in good agreement with the test rig results. Shao et al. [11] proposed a novel analysis method to estimate the fatigue failure life of the drive axle housing of a mining dump truck based on dynamic strain measurement from practical mine road surface conditions combined with finite element analysis. Tekeli et al. [12] investigated the fatigue durability of an engine bracket by utilizing the direct transient analysis technique considering both of the engine and road loads simultaneously. In these researches, the dynamic stress situations experienced by the components, as a main parameter to calculate fatigue life, were mostly obtained from simulated transient analysis or tested strain data. However, it is not practical to conduct transient dynamic stress analysis for the entire durability schedule due to excessive simulation time. Instead of calculating stress at every time step directly, the dynamic stress is obtained by combining the modal stress tensors with the modal coordinates in the modal stress recovery (MSR) method. This makes the solution very fast compared to direct transient approach. Since the MSR approach assumes that the elastic deformations of all degrees of freedom are approximated by a linear

combination of suitable modes, it is possible to calculate dynamic stress with a very good accuracy as long as enough number of modes are retained.

In this study, the modal stress recovery method is introduced to obtain the dynamic stress of the front subframe of a passenger car. The fatigue life of the subframe is then estimated using the S - N method. In addition to that, a structural improvement of the subframe attachment bracket is performed to extend fatigue life of the subframe.

THEORETICAL METHODS

Modal Stress Recovery Method

In the flexible body modeling, the elastic deformation of a part is represented by a linear combination of mode shape and modal coordinate [13], that is

$$\mathbf{u} = \Phi \mathbf{p} \quad (1)$$

where $\mathbf{u}$ is the elastic deformation of the FE model; Φ is the modal matrix that includes both normal and constraint modes; $\mathbf{p}$ is the vector of modal coordinates.

The generalized coordinates of the flexible body include coordinates for rigid body motion and modal coordinates, which can be defined as

$$\mathbf{q} = (x, y, z, \phi, \theta, \varphi, \mathbf{p})^T = (\mathbf{R}, \mathbf{\Psi}, \mathbf{p})^T \quad (2)$$

where x, y, z are the location of local coordinate system under global coordinate system; ϕ, θ, φ are the Euler angle of local coordinate system under global coordinate system; $\mathbf{p}$ is the modal coordinate; $\mathbf{R}, \mathbf{\Psi}$ are the vector expression of rigid body coordinates and $\mathbf{p}$ is the vector expression of modal coordinate $\mathbf{p}$

According to Lagrange equation, the equation for a flexible body, in terms of the generalized coordinates is

$$\mathbf{M}\ddot{\mathbf{q}} + \dot{\mathbf{M}}\dot{\mathbf{q}} - \frac{1}{2} \left(\frac{\partial \mathbf{M}}{\partial \mathbf{q}} \dot{\mathbf{q}} \right)^T \dot{\mathbf{q}} + \mathbf{K}\mathbf{q} + \mathbf{G} + \mathbf{C}\dot{\mathbf{q}} + \left(\frac{\partial \mathbf{\Omega}}{\partial \mathbf{q}} \right)^T \lambda = \mathbf{Q} \quad (3)$$

where $\mathbf{M}$ is the flexible body mass matrix; $\mathbf{K}$ is the generalized stiffness matrix; $\mathbf{G}$ is the generalized gravitational force; $\mathbf{C}$ is the modal damping matrix; $\mathbf{\Omega}$ is the algebraic constraint equations; λ is the Lagrange multiplier for the constraints; $\mathbf{Q}$ is the applied generalized forces.

By solving the Equation (3), the rigid body coordinates and modal coordinates can be obtained as a result. Then the dynamic stress σ could be calculated based on modal stress recovery method by combining the modal stress and modal coordinates, which can be expressed as

$$\sigma = \mathbf{E}_\sigma \mathbf{p} \quad (4)$$

where $\mathbf{E}_\sigma$ is the modal stress matrix.

Fatigue Life Prediction Method

The vehicle component failure is mainly attributed to the stress based high-cycle fatigue as characterized by the S - N curve [14], which is described as

$$\sigma_a = \sigma'_f (2N_f)^b \quad (5)$$

where σ_a is the stress amplitude; σ'_f is the fatigue strength coefficient; b is the fatigue strength exponent; N_f is the number of cycles to failure.

The S - N fatigue data is obtained using a fully-reversed stress cycle in the laboratory. However, actual loading applications usually involve a mean stress on which the oscillatory stress is superimposed. Thus, an equivalent alternating stress to a zero mean stress can be calculated by using the Goodman method [15]:

$$\sigma_e = \frac{\sigma_a}{1 - (\sigma_m / \sigma_u)} \quad (6)$$

where σ_e is the equivalent stress amplitude; σ_m is the mean stress; σ_u is the ultimate tensile strength.

According to Palmgren-Miner rule, the cumulative damage can be calculated as [16,17]

$$D = \sum_{i=1}^k \frac{n_i}{N_{fi}} \leq 1 \quad N = \frac{1}{D} \quad (7)$$

where k is the number of stress series of the variable amplitude loading; n_i is the number of cycles applied at a load corresponding to a lifetime of N_{fi} ; D is the cumulative damage and N is the fatigue life.

VEHICLE MODELING

Finite Element Modeling of Subframe

The front subframe of the passenger car is made of a hydroforming tube with a stamped steel cross member welded at the rear, and 9 mounting brackets welded to the tube. The finite element (FE) model of the subframe is set up by using shell elements, which gives a good relationship between stress results and simulation time consumption. Weld connection between structural parts are simulated using rigid elements. The finite element model contains 68807 nodes and 68051 elements, which is presented in Figure 1.

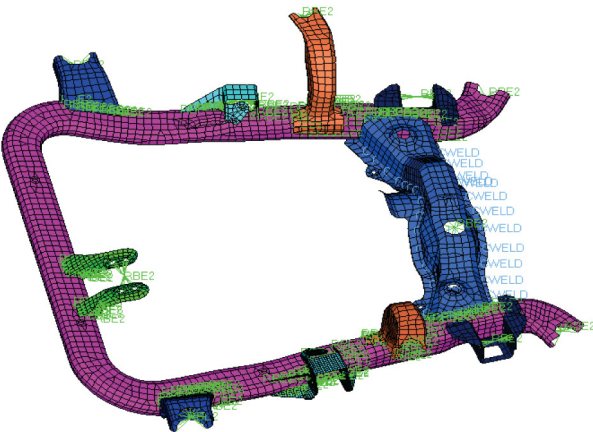


Figure 1. Finite element model of a front subframe

Modal Analysis

The modal analysis of the subframe FE model in free hanging state was conducted based on Block Lanczos approach. The first 6 orders elastic modes were extracted and listed in Table 1, while the fore 6 rigid body modes were excluded. Simultaneously, a modal test was performed on the subframe using impact hammer modal testing shown in Figure 2. The layout of accelerometers for structural response observation was shown in Figure 3. The natural frequencies obtained from FE model and modal test were compared in Table 1, which verified the validity of the subframe FE model. The 7th and 8th modes which were expected to have the most significant effect on the degree of deformation and stress distribution were illustrated in Figure 4. It was confirmed that the analytical mode shapes were in good agreement with the measured ones.

Table 1. Comparison between analysis and test modal frequencies

Mode number	Natural frequencies (Hz)		Relative error (%)	Mode shape
	Analytical	Experimental		
1~6	0	0		Rigid mode
7	86.70	89.03	-2.69%	1st Torsion
8	143.01	137.20	4.06%	1st Bending
9	163.77	155.09	5.30%	
10	224.45	217.86	2.94%	
11	234.85	239.18	-1.85%	
12	274.81	289.86	-5.48%	

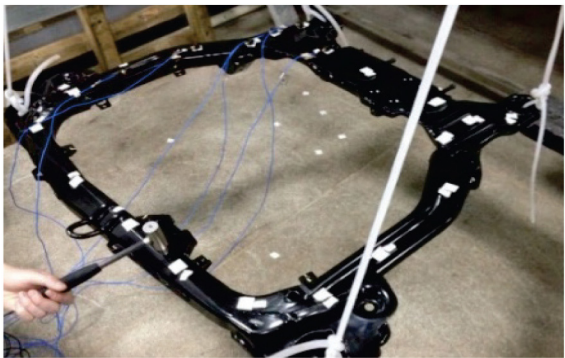


Figure 2. Modal test of subframe

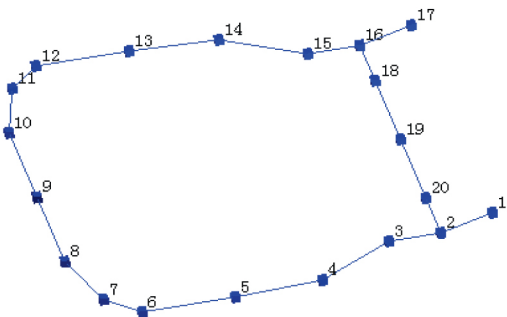
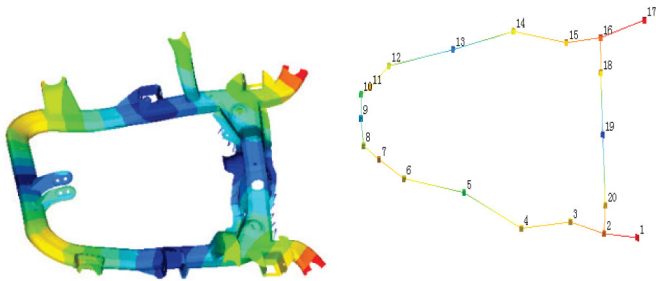
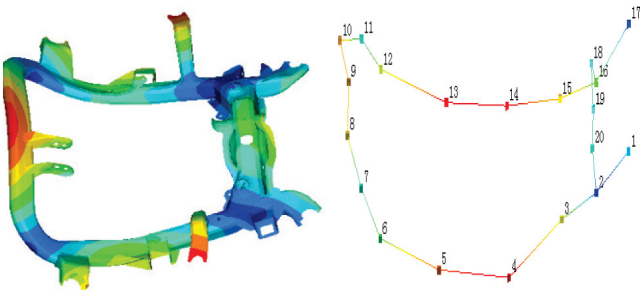


Figure 3. Layout diagram of accelerometer



(1). Mode 7



(2). Mode 8

Figure 4. Mode shapes of the subframe