

Automotive Structural Durability Design Using Dynamic Simulation and Fatigue Damage Sensitivity Techniques

2010-01-0001

Published
04/12/2010

Hong Su
Summitech Engineering of Tempo Group

Copyright © 2010 SAE International

ABSTRACT

It is well known that product durability is related to three major factors, the structure, load and material. That is, the durability performance of an automotive product is not only depended on the structure configuration, but also related to the load dynamic characteristics (such as, profiles and frequency spectrum), and the material fatigue properties as well. Due to the fact that the automotive vehicle loads are dynamic in nature, one of the major technical challenges to the product durability design is how to quantify the fatigue damage sensitivity.

This paper presents a procedure of automotive structural design for durability performance based on the structural dynamic simulation and fatigue damage sensitivity techniques. A methodology for calculating the fatigue damage sensitivity is introduced. Dynamic stresses of an automotive structural system under multiple simultaneously applied proving ground road loads are first simulated, using the finite element model and dynamic simulation, in time domain response analysis. The high stress of each component is then sorted out and converted into the corresponding power spectral density (PSD) result in the frequency domain by using the fast Fourier transformation technology. The fatigue damage and its damage sensitivity of each component are in turn computed as functions of the dynamic stress PSD properties and material fatigue parameters, with respect to the design variables selected. In this way, the design variables that impact the durability life the most can be identified, as to achieve an optimal durability design with the given conditions.

INTRODUCTION

Reliability and performance are fundamental requirements on a durable product. In the automotive industry, the reliability is directly related to the durability of a vehicle [10, 11], since the operation and performance of all parts of the vehicle are under the repetitive dynamic loading conditions, such as the chassis product under loads from uneven road surfaces, a suspension due to shocks from potholes, engine vibrations from unbalanced inertial parts, and extreme temperature cycling, and so on. Vehicle durability means that an automobile can be driven over a continuous and designed long useful life, without requiring an inordinate degree of maintenance, such as a repair or change for failed parts.

In current practice of the automotive industry, in order to meet the durability and reliability targets, prototype vehicles are built during phases of product development, and vehicle durability tests are performed on roads of proving grounds (PG). Vehicle PG road load tests are essentially an accelerated test, and defined in the PG road load durability test schedule. A proving ground durability test typically includes several road load events, with different road surface profiles, travelling speeds and durations, which will, in theory, introduce the equivalent durability damage as compared to that of the vehicle life duration, based on the statistics of the vehicle customer usage.

With advancement of the computer hardware and software, CAE virtual tests of automotive products provide a way to speed up the product development, to minimize prototype tests, and to reduce the development time and costs [16, 17]. A virtual durability test method simulates the automotive structural configuration and proving ground road loads conditions the same way as in a physical lab test, by employing a set of CAE dynamic simulation and durability

**Documents**

Document Sharing Hub

techniques. It predicts the durability life of a component within the automotive product assembly, based on the simulated transient stress response and its material fatigue damage model, at each critical location of the component. The goal of the CAE virtual durability test is to help design engineers to identify and correct any potential structural integrity issues at early stages of the product development, before a prototype is made and tested.

However, in order to realize an optimal durability design of an automotive structure product, engineers of the product development are not only required to evaluate the fatigue damage and durability life, but also need to know the relative influence and trend of the durability performance, with respect to all the key design variables of the automotive product. Due to the fact that the automotive vehicle loads are dynamic in nature, one of the major technical challenges to the product durability design is how to quantify the fatigue damage sensitivity. It is well known that product durability life is related to three major factors, the structure, load and material. The durability performance of an automotive product is not only depended on the structure configuration, but also related to the load dynamic characteristics (such as, profiles and frequency spectrum), and the material fatigue properties as well.

In this paper, a procedure of automotive structural design for durability performance, based on the structural dynamic simulation and fatigue damage sensitivity techniques, is presented. A methodology for calculating the fatigue damage sensitivity, a long with a set of related CAE virtual test technologies, is introduced. The basic engineering theories and key practical technologies are reviewed to establish the background of the proposed procedure, especially the properties of simulated stress response results, in terms of their power spectral density functions in the frequency domain.

Dynamic stresses of an automotive structural system are first simulated, using the finite element model and dynamic simulations, in terms of time domain response to multiple simultaneously applied loads measured from the proving ground. The high stress of each component is then sorted out and converted into the corresponding power spectral density result in the frequency domain by using the fast Fourier transformation (FFT) technology. The fatigue damage and its damage sensitivity of each component are in turn computed as functions of the dynamic stress PSD properties and material fatigue parameters, with respect to the selected design variables in the finite element model of the automotive product. In this way, the design variables that impact the durability life the most can be identified, as to achieve an optimal durability design of each component of the automotive structure, with the given conditions and constraints.

Application examples of the proposed durability design procedure, as to automotive chassis and driveline structure systems, are illustrated. The predicted durability life results in frequency domain from the example are first confirmed with that of stress history results from the time domain. And the fatigue sensitivity results are then further verified from the dynamic simulation runs, final durability prediction data, and quasi-static stress simulation for comparison, as well. The given examples and their results also indicate that the proposed procedure provides a means to clearly reveal the insight on the trends in dynamic durability damage with respect to the different design variables, which are not that obvious and sometimes even contradictory to the results based on the traditional quasi-static approach.

BASIC BACKGROUND OF DURABILITY TECHNIQUES

STRUCTURE MODEL IN TIME DOMAIN

For a general nonlinear dynamic system of an automotive product, for example, such as a vehicle axle structure system with nonlinear springs, rubber mounts and bushings, which is under the dynamic proving ground road loads and different temperature conditions, the mathematical model can be established as a system of nonlinear differential equations in the time domain [1, 2].

$$[M]\{\ddot{x}(t)\} + [C(x, \dot{x}, T)]\{\dot{x}(t)\} + [K(x, \dot{x}, T)]\{x(t)\} = \{P_s(t)\} \quad (1)$$

Where $[M]$ is the mass matrix of the dynamic system; $\{x(t)\}$ is the generalized coordinate vector, as a function of time, t . $[C]$ is the damping matrix, as a function of response $\{x\}$, time t and temperature, T . $[K]$ is the stiffness matrix, as a function of response $\{x\}$, time t and temperature, T . $\{P_s(t)\}$ is the dynamic force vector due to the vehicle loads. The solution $\{x(t)\}$ to the above system of equations is generally obtained by using the finite element integration simulation technique, corresponding to each time step.

For product development in automotive industry, vehicle dynamic loads to automotive structures are mainly interface reactions between wheel tires and road surfaces from proving grounds, which are in general constructed based on statistics of vehicle customer usage, loading conditions, reliability target, and confidence level for the designed vehicle life. A typical set of vehicle proving ground load tests are usually defined in the reliability test schedule, which includes several road load events, with different road surface profiles, travelling speeds and durations.

For an automotive structure, there are commonly three ways to do product design validation tests, that is, (1) the proving

ground test, (2) the lab test, such as on an MTS 329 road load simulator, and (3) the CAE virtual test by using the computer simulation techniques. The proving ground test requires a wholly assembled prototype vehicle. While the lab test needs a prototype structure built and measured loads from the proving ground. And the CAE virtual test only needs information data of the design model geometry and the measured loads, as well. All the test measured data and the computer simulated data of an automotive structure under the proving ground road loads are typically in time history format, in terms of acceleration and stress (or strain) response results [6, 11]. The acceleration results are usually utilized to identify the dynamic behaviors of the structure system. While stress or strain results will then be used for further durability evaluation based on the material fatigue damage properties.

The advantage of the computer virtual tests is simply due to the fact that its structure model is generated based on the design geometry data, loading data and material information, not any physically built parts are required. It employs the same measured proving ground loadings as the test lab, but without expensive test equipment and fixtures. In addition, it is much more efficient to make changes to the design and to run numerous what if study tests, as to find the best design solution under the given load conditions. All the time domain simulations of a structure can be realized by employing general commercial finite element solver software.

FATIGUE EVALUATION IN TIME DOMAIN

In the process of the structure durability analysis, all stress results of each component are first ranked and sorted out for the elements with the highest stress levels. The selected element dynamic stress time history is the result of the component from the time domain transit simulation under the given proving ground road loads. The representative stress time history is then further evaluated, along with the material fatigue data and damage model, for prediction of durability life of the component.

Material Fatigue Damage Model

The material fatigue properties can usually be measured as S-N curves, which define the relationship between the stress amplitude level, S_A , versus the mean cycles to failure, N . For most high cycle fatigue durability problems ($N \geq 10^4$), the S-N curves can be expressed as a simplified form [7]:

$$B = N S_A^m \quad (2)$$

Where B and m are the material properties varying with material type, and its loading and environment conditions, such as mean stress, surface finishing, and temperature.

Accumulated Damage Evaluation

In the time domain, the expected value of accumulated damage, $E[AD]$, due to dynamic fatigue stress loading on an automotive structure component, is evaluated based on the Palmgren-Miner's rule, and expressed as [8]:

$$E[AD] = \sum_i \frac{n_i(S_A)}{N_i(S_A)} \quad (3)$$

Where $n_i(S_A)$ is the number of cycles applied at stress amplitude level S_A , which is related to the simulated stress time history response; while $N_i(S_A)$ is the number of cycles to failure at stress amplitude S_A , which is based on the material fatigue properties, such S-N curves, and i is the stress load case number. The geometrical and surface conditions of all critical components will be also considered for the modified material damage model.

It should be noticed that in order to compute the accumulated damage in equation (3), the fatigue damage effects of all the stress levels are to be calculated with respect to the material fatigue data and curves. And the number of cycles of each stress range, corresponding to all the peaks and valleys, with respect to variable amplitude load histories, needs to be counted by employing a proper cycle counting algorithm, such as the Rain-flow Cycle Counting method by Matsuishi and Endo [9].

FATIGUE EVALUATION IN FREQUENCY DOMAIN

On the other hand, in the frequency domain, the expected value of accumulated damage, $E[AD]$, due to fatigue random loading is also evaluated based on the Palmgren-Miner's rule, and expressed as [4, 15]:

$$E[AD] = \sum_i \frac{n_i(S_A)}{N_i(S_A)} = \int_0^\infty \frac{p(S_A)N}{N(S_A)} dS_A \quad (4)$$

Where $n(S_A)$ is the number of cycles applied at stress amplitude level S_A , $p(S_A)$ is the probability density function of the stress amplitude, while N is the mean cycles to failure, $N(S_A)$ is the number of cycles to failure at stress amplitude S_A .

STRESS RESULTS IN TWO DOMAINS

In order to facilitate evaluation of the durability life and fatigue sensitivity of a structure, using an alternative approach in the frequency domain, the relationship between the stress time history and the stress power spectral density

should be first established, by using the Fourier transformation technique.

Fourier Transformation Technique

Stress response results in a vehicle structure under proving ground road loads are random in nature [3]. It has been established that most road surface irregularities are normally distributed and may be accurately described by a stationary random process [5, 6]. For a stationary ergodic random phenomenon, the ensemble averages are equal to the time averages. The statistical properties of a stationary ergodic process can then be computed from a single time history of sufficiently long period. The time average of a random variable $x(t)$ is equal to the expected value of $x(t)$, as defined as:

$$E[x(t)] = \lim_{T \rightarrow \infty} \frac{1}{T} \int_0^T x(t) dt \quad (5)$$

The mean square value of $x(t)$ is found by applying the average mean operation to variable $x^2(t)$ over a time interval of T .

$$E[x^2(t)] = \overline{x^2} = \lim_{T \rightarrow \infty} \frac{1}{T} \int_0^T x^2(t) dt \quad (6)$$

Correlation function is a measure of the similarity between two random quantities in a time domain τ . For a single record $x(t)$, the autocorrelation $R(\tau)$ of $x(t)$ is the expected value of the product $x(t)$ and $x(t+\tau)$:

$$\begin{aligned} R(\tau) &= E[x(t)x(t+\tau)] \\ &= \lim_{T \rightarrow \infty} \frac{1}{T} \int_0^T x(t)x(t+\tau) dt \end{aligned} \quad (7)$$

When $\tau=0$, the above definition reduces to the mean square value.

$$R(0) = \overline{x^2} = E[x^2(t)] \quad (8)$$

A random process can be described in frequency domain in terms of power spectral density functions. It can be shown that the power spectral density functions are related to the correlation functions by Fourier transform pairs [3]. For a single random record $x(t)$, the relationship can be shown in the following equations:

$$S(\omega) = \int_{-\infty}^{\infty} R(\tau) e^{-i\omega\tau} d\tau \quad (9)$$

And

$$R(\tau) = \frac{1}{2\pi} \int_{-\infty}^{\infty} S(\omega) e^{i\omega\tau} d\omega \quad (10)$$

Where $\omega=2\pi f$. From equations (7) and (10) it is clear that auto-correlation $R(\tau)$ is an even function of τ , and the auto PSD $S(\omega)$ is real. When $\tau=0$, the mean square value is related to PSD $S(\omega)$, from (8) and (10), by

$$\overline{x^2} = R(0) = \frac{1}{2\pi} \int_{-\infty}^{\infty} S(\omega) d\omega \quad (11)$$

BASIC SPECTRAL PROPERTIES

Random Vibration Properties

Under random vibration loads, the fatigue damage of a structure is estimated based on the statistical properties of the response stress PSD function. The statistical characteristics of the random vibration response stress can be obtained through the moments of the PSD function. The n^{th} spectral moment, m_n of the stress PSD function $S(f)$, frequency f in unit of Hz, is defined by the following equation [15]:

$$m_n = \int_0^{\infty} f^n S(f) df \quad (12)$$

Properties of a continuous stationary Gaussian process can be related to the above n^{th} moments, m_n of the PSD function. The root mean square (RMS) value, σ of the stress PSD function is

$$\sigma = (m_0)^{1/2} = \left(\int_0^{\infty} S(f) df \right)^{1/2} \quad (13)$$

The average rate of the zero crossing with positive slope, $E[0]$, which is also called as equivalent frequency in unit time, is expressed as:

$$E[0] = \left[\frac{m_2}{m_0} \right]^{1/2} \quad (14)$$

Frequency Fatigue Damage Model

The expected value of the accumulated damage, $E[AD]$, due to fatigue random loading is evaluated based on the Palmgren-Miner's rule as equation (4), and