

CAE Virtual Durability Tests of Automotive Products in the Frequency Domain

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ABSTRACT

Both NVH and durability performance of automotive products are mainly related to their structural frequency characteristics, such as their resonant frequencies, normal modes, stiffness and damping, and transfer function properties. During the automotive product development, product design validation test loads for NVH and durability are, therefore, often specified in the frequency domain, in terms of either swept sinusoidal vibration or random vibration in power spectral density function. This paper presents a procedure of CAE virtual design validation tests for durability evaluation due to the frequency domain vibration test loads. A set of frequency domain simulation techniques and durability evaluation methodologies, for material fatigue damage due to either random or sinusoidal vibration loads, are introduced as well. Finite element models of automotive products are developed along with their nonlinear frequency dynamic stiffness and damping elements and properties, such as those related to the mounts and rubber bushings. The dynamic stress simulation is realized by utilizing the frequency response analysis technique. Statistical properties are employed to account for the scatter nature of material fatigue S-N raw data, and a damage model for durability performance is then established by using the reliability and tolerance interval techniques. The durability life evaluation is based on the simulated dynamic stresses and the newly defined material fatigue damage model. Two examples of virtual durability evaluation tests of automotive products are also provided to illustrate applications of the proposed procedure and techniques, with respect to the random and swept sine vibration loads, respectively.

INTRODUCTION

It has been well established [1, 2] that the noise, vibration and harshness (NVH), and dynamic durability performance of a product are essentially related to its frequency characteristics, such as their resonant frequencies, normal modes, stiffness and damping, and transfer function properties between its inputs and outputs. Product dynamic loads and engineering response data, such as force, acceleration, stress and

strain commonly measured in terms of time histories, can be conveniently converted into the frequency domain for revealing their respective frequency spectrum, insight information and inherent properties [3], by using the fast Fourier transform (FFT) technique. For durability design validations of automotive products the vibration test loads are, therefore, often specified in the frequency domain, in terms of either swept sinusoidal vibration function or random vibration function in terms of power spectral density (PSD) [4,10,11].

In this paper, a procedure of virtual design validation tests for durability evaluation due to frequency domain vibration loads, by employing a set of computer aided engineering (CAE) simulation and durability techniques, is presented. The proposed virtual durability test procedure incorporates several frequency domain modeling and simulation technologies. The frequency domain test loads under the vehicle test environments, due to either the road load interface loads with the vehicle or the engine vibration loads, are simulated in the random or swept sine vibration format. The finite element models of automotive products are developed along with their nonlinear frequency dynamic stiffness and damping properties, as related to mounts and rubber bushings in vehicle structures. Dynamic stress response results are then simulated by using the frequency response analysis technique. The statistic properties are employed to account for the scatter nature of the material S-N fatigue raw data, and a damage model for durability evaluation is then established using the reliability and tolerance interval techniques. The durability evaluation is based on the dynamic stress results and the newly defined material damage model. The predicted durability life results of an automotive product under the frequency vibration test are then obtained for the given reliability parameters.

Two examples of durability evaluation of automotive products are provided to illustrate the applications of the proposed procedure and techniques. One example is on an axle structure system under a random vibration load based on the measured proving ground data. While the other is an engine suspension system under a swept sine vibration load, which was originated from dynamic

running conditions of an I4 gas engine. The virtual durability test results in the frequency domain are presented in terms of the estimated life for the given reliability target and a confidence level. It has shown that the virtual test approach can reveal the insight relationship into the design parameters, product design weak spots and durability life, provide a guidance to design improvement, and help to achieve our goal for only one successful physical design validation test.

VEHICLE LOADS IN THE FREQUENCY DOMAIN

There are generally two types of measured vehicle load raw data [10] employed in automotive industry for product development according to their original sources, namely, the proving ground load data for vehicle body mounted products, and the engine load data for engine mounted products. It has been well known that both the time and frequency domain data are mathematically related by using the fast Fourier transform technique [3, 4]. The commonly measured vehicle load data in terms of time histories are conveniently converted into the frequency domain for their respective frequency spectrum and then frequency load specifications.

In order to completely define the properties of a load in terms of the total damage [11], both the measured load data and the load schedule information are needed. The measured load data quantifies the load level and its frequency contents for each load event; while the load schedule information tells how the loads are applied to the product, such as duration and conditions. Therefore, the complete proving ground data and information include the measured vibration loads from vehicle due to road surface profile interfaces, and the corresponding schedule of road surface events, with defined vehicle speed, duration and road test schedule of a given type of vehicle. The engine load data, on the other hand, are measured vibrations mainly due to unbalanced inertial forces of the centrifugal and reciprocating components, corresponding to different engine speed RPM, in terms of harmonic contents of vibrations and a schedule with the engine duty cycle definition.

An engineering procedure along with its related methods for determination of a justified vibration test load specification in the frequency domain for automotive products, based on the measured vehicle load data, has been introduced in [11]. The resulted vibration test load specification for an automotive product can be presented in either swept sinusoidal or random vibration profile format, and both will have an equivalent durability damage level for the given test duration and reliability parameters.

BASIC EQUATIONS OF A STRUCTURE MODEL

In the automotive industry, vehicle structural systems commonly employ many plastic parts and rubber bushing components. A vehicle structural system, in general, demonstrates strong nonlinear response characteristics, with respect to the dynamic load level,

excitation frequency and temperature conditions, due to those nonlinear components. In order to model and simulate such a nonlinear structural system for virtual design validation (DV) tests of the automotive products, the modeling technique by using an array of locally linearized systems [13] is introduced. A brief development of the basic equations of the locally linearized model of a nonlinear automotive system is summarized as follows.

STRUCTURE MODEL IN TIME DOMAIN

For a general nonlinear dynamic structural system of an automotive product, such as a vehicle axle structure or engine suspension system with rubber mounts and bushings, which is under the dynamic proving ground or engine vibration loads and different temperature conditions, the mathematical model can be established as a system of nonlinear differential equations in the time domain [5, 6].

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

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 the system due to dynamic vibration loads in the frequency domain, the steady state solution to the system of equations (1) is extremely difficult, if is not impossible, in terms of CPU time and resources.

FREQUENCY DOMAIN MODEL

An alternative approach to the above problem is to solve the system of equations in the frequency domain, in terms of transfer functions, using the Fourier transformation technique. A localized nonlinear model of a dynamic system, corresponding to the equation (1), in the frequency domain can be expressed as follows [14]:

$$(-[M]\omega^2 + i[C_{eq}(X, \omega, T_i, P_j)]\omega + [K_{eq}(X, \omega, T_i, P_j)])\{X(\omega)\} = \{P_s(\omega)\} \quad (2)$$

where $\omega = 2\pi f$, f is the frequency (Hz), $i = \sqrt{-1}$, $(i = 1, 2, \dots, L)$, $(j = 1, 2, \dots, M)$, L is the total number of temperature cases, M is the total number of load level cases; $[C_{eq}]$ is the equivalent local damping matrix, and $[K_{eq}]$ is the equivalent local stiffness matrix, as a function of response, frequency, temperature, and load level.

$\{X(\omega)\}$ and $\{P_s(\omega)\}$ are the structure response and load vectors in frequency domain, respectively.

RANDOM VIBRATION MODEL

When the input loads are in random vibration format and expressed as a matrix of the loading power spectral density (PSD) functions, $[S_p(\omega)]$, the system of time domain differential equations of motion of the structure in (1) is then reduced to a system of algebra equations as follows.

$$[S_x(\omega)]_{n \times n} = [H(\omega)]_{n \times m} [S_p(\omega)]_{m \times m} [H(\omega)]_{m \times n}^{*T} \quad (3)$$

where m is the number of multiple input loads; n is the number of output response variables. The T denotes the transpose of a matrix. $[H(\omega)]$ is the transfer function matrix between the input loadings and output response variables.

$$[H(\omega)] = (-[M]\omega^2 + i[C_{eq}]\omega + [K_{eq}])^{-1} \quad (4)$$

The random response variables $[S_x(\omega)]$, such as displacement, acceleration and stress response, in terms of power spectral density functions, are obtained by solving the system of the linear algebra equations in (3).

By employing equations (2) and (3), the solution to nonlinear differential equation (1) in the time domain has been translated to a solution to the system of localized linear algebra equations, in the frequency domain, for the given loads and conditions, in terms of sine sweep or random vibration format, respectively.

MATERIAL FATIGUE DAMAGE MODEL

MATERIAL FATIGUE DATA

The material fatigue properties are usually measured as S-N curve, which defines 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 curve can be expressed as a simplified form:

$$B = N S_A^m \quad (5)$$

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.

RELIABILITY PARAMETERS

It is well known [7] that material fatigue test data of sample pairs (S_i, N_i) are randomly scattered on their stress (S) and number of cycles (N) plots. A mean S-N

curve can be estimated by using the least square analysis of all test data sample pairs of (S_i, N_i) on log-log scales, based on a data sample size of W , such as illustrated in Figure 1. The material fatigue model in equation (5) is therefore only a medium S-N curve, estimated from the limited sample size (W). It is also known that the fatigue life $\log(N)$ of a set of S-N fatigue test data has a random distribution, corresponding to a given fatigue strength level $\log(S)$.

If we assume that $\log(N)$ is normally distributed for given $\log(S)$, and its variance is a constant, the uncertainties in S-N estimators can be then accounted for by using the tolerance interval technique. That is, the new design S-N curve can be shifted to the left (safe side of data) by amount of margin, which is determined from reliability requirements and statistical properties of the fatigue sample data. The new design S-N curve for the durability evaluation and the test load equivalency can then be defined as [16]:

$$\log(B_0) = \log(B) - k_{p,\gamma}(W) D_s \quad (6)$$

where B_0 is the new fatigue strength coefficient, replacing B in equation (5), D_s is the standard deviation of the test sample data in $\log(N)$, and $k_{p,\gamma}(W)$ is the factor for a one-sided tolerance interval, corresponding to a proportion p of the population (reliability), confidence γ and sample size W . For a given set of reliability parameters, that is, the sample size W , confidence level γ and reliability goal p , the factor $k_{p,\gamma}(w)$ can be found in the table of factors for one-sided tolerance limits for normal distributions, such as in [18].

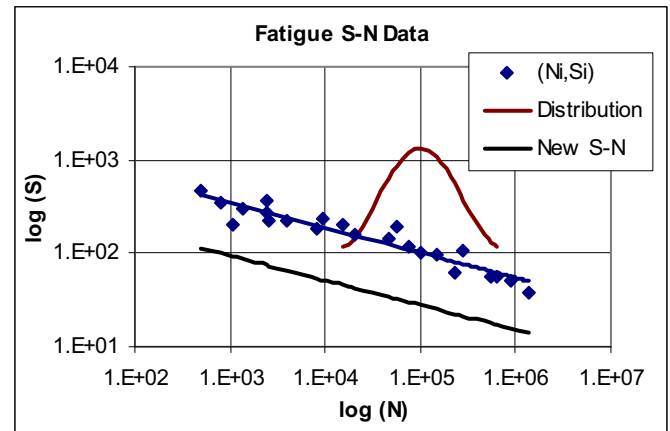


Figure 1: A reliability model for S-N fatigue data

MATERIAL FATIGUE IN FREQUENCY DOMAIN

RANDOM VIBRATION FATIGUE

Under random vibration loads, the fatigue damage of structures 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