

Improving the Accuracy of Dynamic Vibration Fatigue Simulation

Kurt Munson, Frederic Kihm, and Andrew Halfpenny
 HBM Prencia

ABSTRACT

Finite Element Analysis (FEA)-based structural simulations are typically used to assess the durability of automotive components. Many parts experience vibration in use, and resonance effects are directly linked to many structural problems. In this case, dynamics must be included in the structural analysis. Dynamic FEA can be more realistic than static analysis, but it requires knowledge of additional characteristics such as mass and damping. Damping is an important property when performing dynamic FEA, whether transient or steady state dynamics, as it governs the magnitude of the dynamic stress response and hence durability. Unfortunately the importance of damping is often overlooked; sometimes a default damping value is erroneously assumed for all modes. Errors in damping lead to errors in the stress response, which in turn lead to significant changes in the fatigue life estimates. For example: 3% versus 5% damping ratio can lead to fatigue life estimates varying over an order of magnitude. Therefore, accurate fatigue life predictions rely heavily on realistic damping ratios. Damping must be evaluated from physical tests, in which the component is excited and responses are measured. The modal parameters for the structure's modes of vibration are then extracted using experimental modal analysis techniques. The purpose of this paper is first, to explain the critical role that damping plays in fatigue damage; and second, to recommend best practices for improving FEA-based stress and durability analysis. Methods for quickly and easily determining damping ratios will be introduced. An example of an exhaust system will be used to illustrate the importance of using experimental modal analysis to adjust the properties of a FE-based dynamic structural analysis in order to obtain fatigue life results that correlate with testing.

CITATION: Munson, K., Kihm, F., and Halfpenny, A., "Improving the Accuracy of Dynamic Vibration Fatigue Simulation," *SAE Int. J. Engines* 10(2):2017, doi:10.4271/2017-01-0333.

INTRODUCTION

Durability is critical to meeting product performance, reliability, and warranty goals. Fatigue is a progressive failure mode caused by stress cycles. These stress cycles can be driven by dynamics and resonance under some conditions, so prediction of fatigue life requires an understanding of resonance.

Prediction of product durability via analytical means like CAE-based simulation is important to product development. FEA is often used as part of this analysis, and can account for dynamics. However, these analytical stress results need to be correlated to real structural behavior in order to accurately predict fatigue life.

Often a damping ratio is assumed when performing the FE structural analysis. This can lead to errors in the fatigue life estimation. Experimental techniques can be used to properly quantify the damping and therefore improve the accuracy of the durability analysis.

The purpose of this paper is first, to explain the critical role that damping plays in fatigue life estimation; and second, to recommend best practices for improving FEA-based stress and durability analysis. This includes experimental techniques for determining modal damping and correlating analytical results with the real world.

DYNAMICS, VIBRATION, AND RESONANCE

Many automotive components and structures experience vibration in use. Dynamic excitation like engine vibration and road roughness can occur over a broad frequency range and excite significant structural modes. Resonance effects are directly linked to many structural problems.

Single Degree of Freedom Systems

Mechanical structures can oscillate elastically when exposed to dynamic excitation. Take for example a single lump mass with a flexible connection to the ground, as shown in [Figure 1](#). There is only a single mass, and it can only move in one direction, so this is called a single degree of freedom (SDOF) system [1].

The motion of the mass is governed by [Equation 1](#):

$$F(t) = m\ddot{x}(t) + c\dot{x}(t) + kx(t) \quad (1)$$

All physical structures have natural frequencies at which they freely respond if they are disturbed from equilibrium [2]. This is the eigensolution to the equation of motion seen above, and is called a resonant frequency. The response can be represented in the frequency domain as the frequency response function (FRF), which has both

gain and phase. Gain is the ratio of response to excitation, while phase shows any lead or lag between excitation and response. [Figure 2](#) shows the FRF for a lightly damped SDOF system.

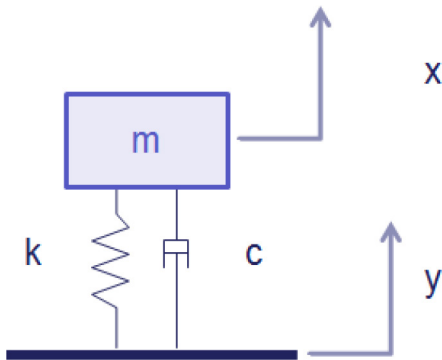


Figure 1. Single degree of freedom (SDOF) model

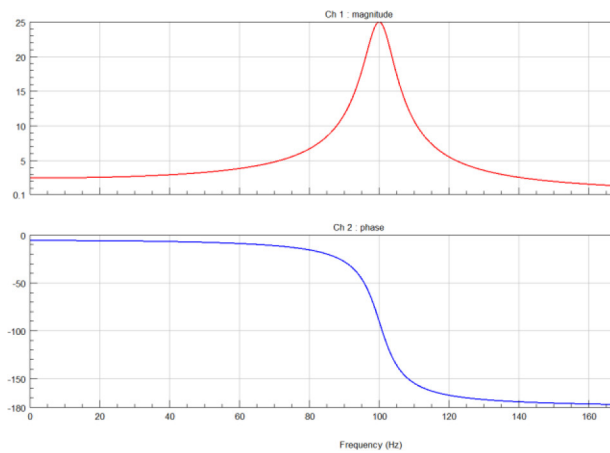


Figure 2. Frequency response function (FRF) of a lightly damped SDOF system.

For a damped system, the natural frequency of the response is dependent on stiffness k , mass m , and damping ratio ξ in [Equation 2](#):

$$\omega_n = \sqrt{k/m} \sqrt{1 - \xi^2} \quad (2)$$

Note that for lightly damped systems, the natural frequency is not heavily influenced by the damping ratio.

When excited at this resonant frequency, the magnitude of the response is dependent on stiffness and damping. As damping changes, the natural frequency changes very little, but the gain's magnitude changes significantly. [Figure 3](#) shows the gain of the FRF for 2% and 5% damping:

Each natural frequency has a corresponding damping ratio. Damping values are empirical values that must be obtained experimentally [2].

Damping is proportional to the rate at which the FRF's peak decreases away from the natural frequency [3]. Mass and stiffness govern the frequency at which resonance occurs, while damping

controls the magnitude of the response. Damping is important when investigating system dynamics as it governs the magnitude of the dynamic stress response and hence durability.

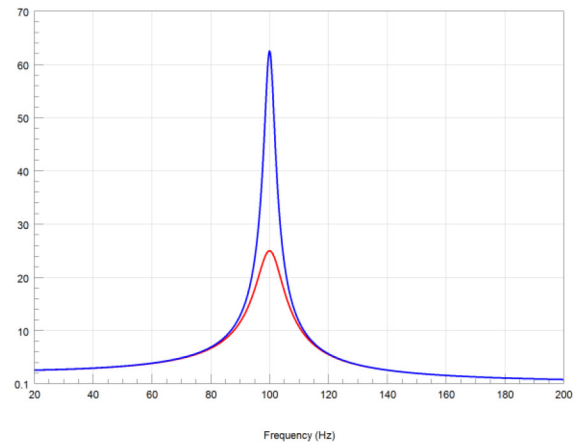


Figure 3. FRF gain for two different damping ratios: 2% (blue) vs 5% (red)

Multiple Degree of Freedom Systems

More complicated structures are referred to as multiple degree of freedom (MDOF) systems, in which multiple masses can move in different amounts and directions under excitation. These relative motions at a given natural frequency create a deformed shape, also known as the eigenvector or mode shape. Nearly all structures have multiple natural frequencies [2].

MDOF systems have more complicated FRFs that contain multiple resonant frequencies or modes of vibration, one for each degree of freedom. [Figure 4](#) shows a system with two degrees of freedom and hence two resonant frequencies in the FRF. Each vibration mode has its own natural frequency, mode shape, and damping ratio.

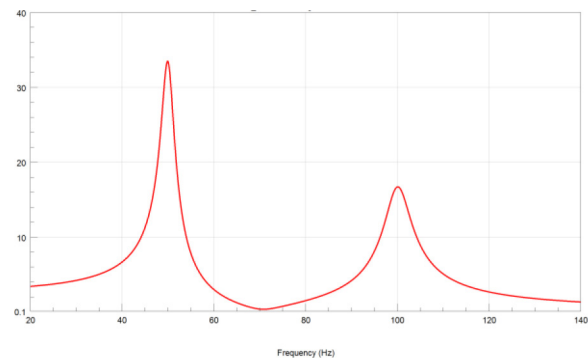


Figure 4. FRF showing two resonant frequencies

This MDOF example can be viewed as the superposition of dynamic behavior of a number of SDOF systems [3]. If the modes are not closely spaced and are not heavily damped, then at a given resonant frequency, the structure behaves predominantly as an SDOF system. This means the modal parameters can be determined for each mode separately, one after another. If the modes are close, more sophisticated methods that account for modal interaction can be used to extract modal parameters [4].

STRUCTURAL SIMULATION FOR STRENGTH AND DURABILITY

Structural simulation via FEA is often used to assess the strength and durability of automotive components. This finite element stress analysis can be done with a number of different techniques: linear or nonlinear, static or dynamic, transient or steady state, time or frequency domain. These predicted stresses from FEA can then be used in conjunction with fatigue life prediction algorithms to estimate product durability.

Dynamic FEA can predict more realistic stresses than quasi-static FEA if the frequencies of excitation align with the frequencies of resonance. Up to about 1/3 of the natural frequency, response can be approximated as quasi-static, as gain is close to unity (shown in Figure 5). If the highest frequency of excitation remains less than 1/3 of the system's lowest natural frequency, static FEA can suffice. Otherwise, dynamic FEA is needed.

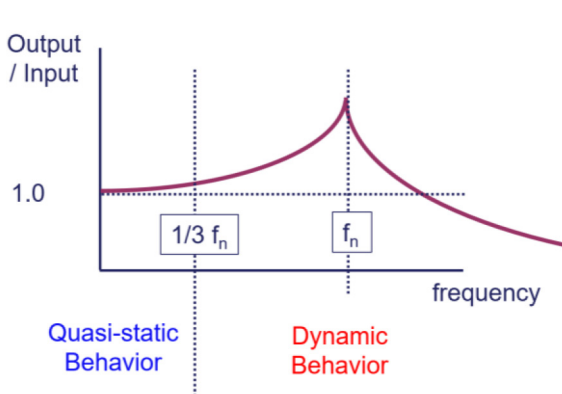


Figure 5. Static versus dynamic response

Stiff components like control arms and steering knuckles often behave in a quasi-static manner because their natural frequencies are far higher than the frequencies characteristic in road, engine, and other service loading. In this case, a static FEA solution can be used successfully to calculate stress, and dynamics is not needed in calculating stress or fatigue life.

Other parts behave dynamically as their resonant frequencies may be excited by road and engine loading. Examples of these parts include crankshafts, fuel rails, and body structures. In this case, dynamic FEA is required to adequately capture resonance and the resulting amplified stresses. Dynamic FEA can be more realistic than static analysis, but it requires knowledge of additional characteristics such as mass distribution and damping.

Often assumptions are made about damping in order to perform dynamic FEA, but this can lead to errors in predicted stress and life. Using experimentally-determined modal damping improves the calculation of both stress and life.

WHAT'S IMPORTANT IN FATIGUE

Fatigue is a progressive failure mode that is driven by stress cycles. If numerous enough, these stress cycles may cause structural failure over time even though the stress magnitude is not large enough to

cause immediate failure. It is the combination of stress cycle size and number of repetitions of those cycles that drive fatigue failure. The size of these stress cycles is often described by the total change of stress, called stress range. The number of cycles seen in service can be counted many ways, including rainflow cycle counting [5].

The Stress Life (SN) Curve

A stress-life (SN) fatigue curve describes for a given material how the stress range and number of cycles to failure are related. The SN curve is experimentally determined by subjecting material coupons to known cyclic loading and recording failure times. Typically 10-20 material coupons are broken in order to understand the relationship between stress and life. An example SN curve is shown in Figure 6.

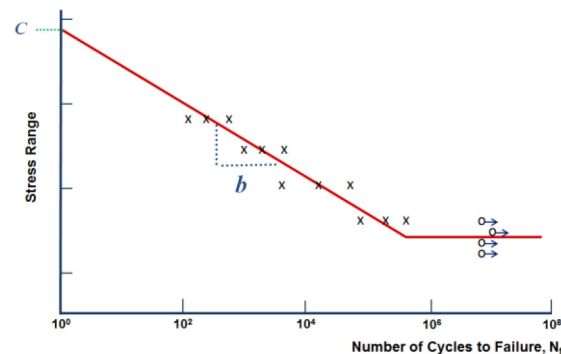


Figure 6. Stress-Life (SN) curve

Basquin's Exponential Relationship

The relationship between stress range and number of cycles to failure is often described mathematically using Basquin's formulation [6], in Equation 3:

$$S_R = C \cdot N_f^b \quad (3)$$

where S_R is the range of the applied stress cycle, N_f is the number of cycles to failure, C is the Basquin coefficient (intercept of the SN curve with the stress range axis), and b is the Basquin exponent (slope of the SN curve).

This power law can be rearranged to solve for fatigue damage, which is the inverse of number of cycles to failure. Equation 4 illustrates the nonlinear relationship between stress range and damage:

$$Damage = \left(\frac{S_R}{C} \right)^{\left(-\frac{1}{b} \right)} \quad (4)$$

Fatigue damage per cycle increases exponentially as stress range increases. This is mathematically related to b , the slope of the SN curve. The negative reciprocal of this is sometimes called the damage exponent, which is useful for illustrating how sensitive fatigue damage is to stress range.