Estimating Damping Values Using the Half Power Method

Tech Brief 150401 D

Integrated Systems Research, Inc.

April, 2015

steve.carmichael@isrtechnical.com

Abstract:

This Tech Brief discusses potential pitfalls to be aware of when employing shake table data and the half power method in estimating the damping associated with structural modes. Data provided to the engineer is oftentimes incomplete. Being cognizant of the limitations in the data, how to assess what has been provided, and extract damping values at resonances are topics covered in this paper.

Background:

Modal structural parameters (e.g., resonant frequencies, modal stiffness, mass and damping) are typically estimated with curve fitting programs using impact or shake table data. More frequently, however, the design engineer is provided with data that has not been curve fit and damping is often a desired parameter to be obtained. Damping is the critical parameter at resonance, since it is the mechanism that controls the response of the structure as it freely exchanges its kinetic and potential energy at a natural frequency. Endurance tests are usually run at the fundamental natural frequency of the component. Knowing the amount of damping in the component is critical to evaluating the alternating stresses during test and assessing the components' margin against fatigue failure.

Techniques for Estimating Damping:

There are three common means of estimating damping:

The first is a time domain approach and is used to estimate the damping associated with the components' fundamental frequency. This is often referred to as the *log decrement* approach. The rate of decay of an impulse response is the basis for the estimation. The approach is straight forward to use as long as higher modes are not modulated with the fundamental response.

A second approach is to extract damping values based on phase angle response around the resonance. This is a frequency domain approach. Without damping, the response of a structure is either in or out of phase with the source of excitation. At resonance, however, such a structure would have an unbounded response. Since all actual systems have some internal damping, there exists a phase relationship between the structure and the excitation source. The response of the component 90 degrees to the input is referred to as the imaginary response. The real response away from a resonance is either in or out of phase with the excitation. The slope of the phase angle response at resonance is controlled by the damping in the system. This phase angle data can also be used to estimate damping values.

The third means of estimating damping is referred to as the *half power* approach. The method uses the bandwidth at resonance, obtained from the response modulus, to estimate damping. The same data (bandwidth) can be obtained from phase angle information employing a modal circle.[1] This paper, however, reviews only the half power approach using the response modulus and how it relates to the structural response of a mechanical component.

Dynamic Behavior of a Structural Component:

The three (3) degree of freedom (DOF) system, shown in Figure 1, illustrates the general behavior of a mechanical system under dynamic loading and the potential pitfalls in using the half power approach in estimating damping.

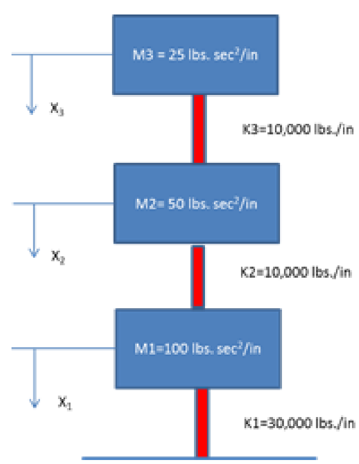

Figure 1 – 3 DOF System

[1] Reference Modal Testing: Theory and Practice, D.J. Ewins, Research Studies Press section 4.3.3

Figures 2 and 3 provides the total real and imaginary responses of the 3 DOF system for a unit excitation input at DOF X_3 and response at DOF X_1 (e.g., $H(\omega)_{31}$).

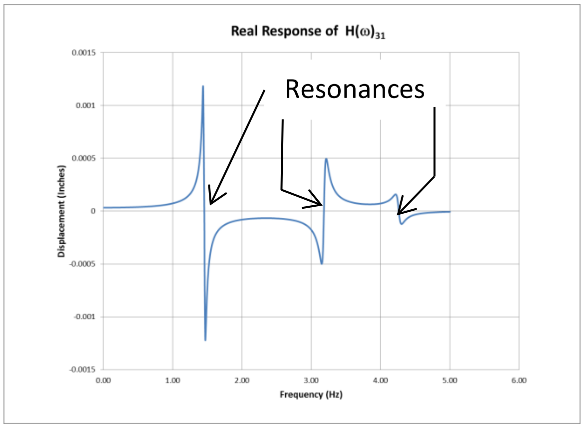

Figure 2 – Real Response $H(\omega)_{31}$

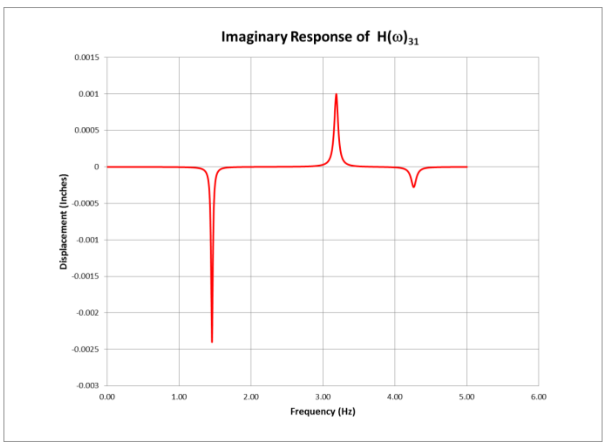

Figure 3 – Imaginary Response $H(\omega)_{31}$

Typically, shake table data will be provided as magnitude and phase. The resonant frequencies are normally identified by observing where the phase angle response passes through 90 degrees. The relationship between the phase angle and the damping factor γ is provided in equation 1.0.

$$tan\varphi = \frac{2\gamma \frac{\omega}{\omega_n}}{1 - \left(\frac{\omega}{\omega_n}\right)^2}$$

Equation 1.0

Figure 4 provides the phase plots for two levels of damping.

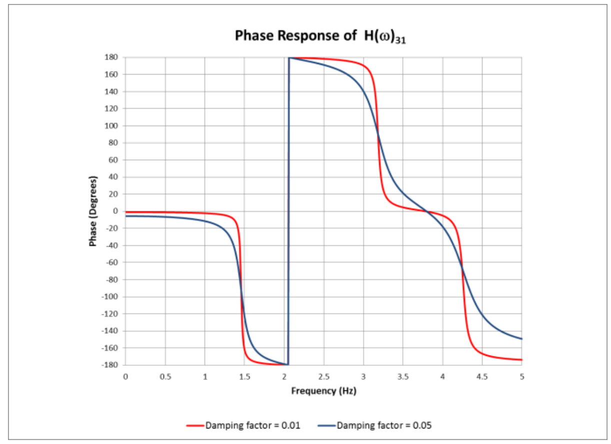

Figure 4 – Phase Response $H(\omega)_{31}$

The magnitude of the response for a damping factor $\gamma = 0.01$ is shown in Figure 5.

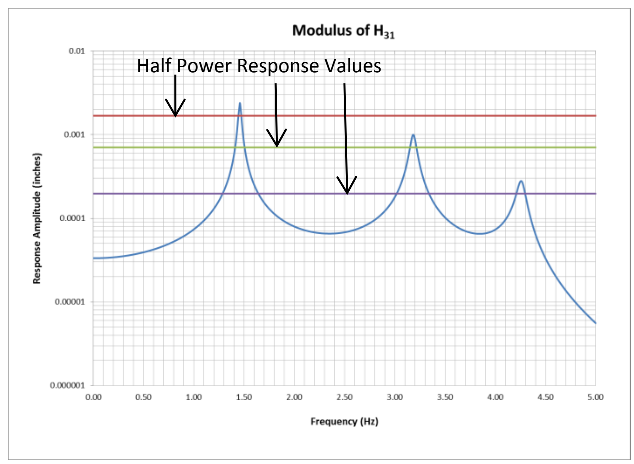

Figure 5 –Modulus Response of $H(\omega)_{31}$

The system response is the combination of 3 modes which are superimposed to obtain the total response. Figure 6 provides a plot of the real contribution of the three modes and Figure 7 the imaginary responses.

When using the half power approach with modulus plots, the engineer needs to be aware that the resonant response at a particular mode is influenced by the mass lines of lower modes and the stiffness lines of higher modes. Due to this influence the bandwidth estimated by this approach can have significant error and care needs to be taken to identify when this situation may arise.

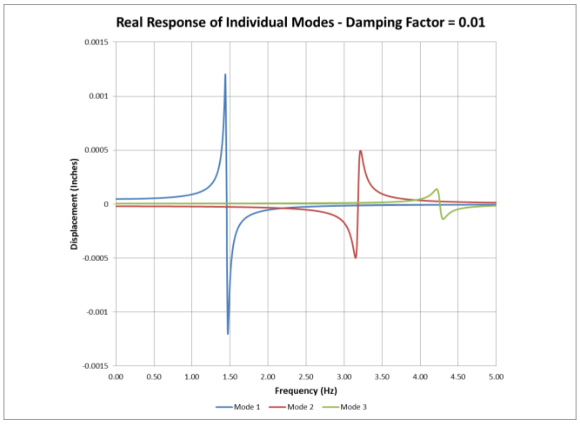

Figure 6 – Real Response of Individual Modes

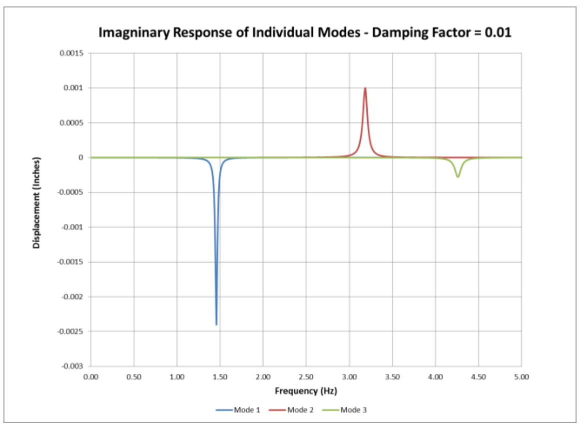

Figure 7 – Imaginary Response of Individual Modes

Half Power Damping Estimations:

The magnitude or modulus of the response is the root sum squared of both of the real and imaginary terms. The half power approach for estimating damping is based on finding the bandwidth for each mode. The bandwidth is the normalized $\Delta\omega$ across the resonant response at the amplitude of $0.707R_{max}$. The bandwidth is equal to the structural damping term η_r. Using this value the damping is proportional to the strain in the system and is sometimes referred to as complex stiffness damping or hysteretic damping. Typically, the damping factor γ is used in a finite element simulation. The damping factor γ, which is often referred to as viscous damping, is proportional to the velocity and is half the value of η_r.

In this example, calculating the damping using the half power method provides excellent results for the first mode. This is shown in Figure 8.

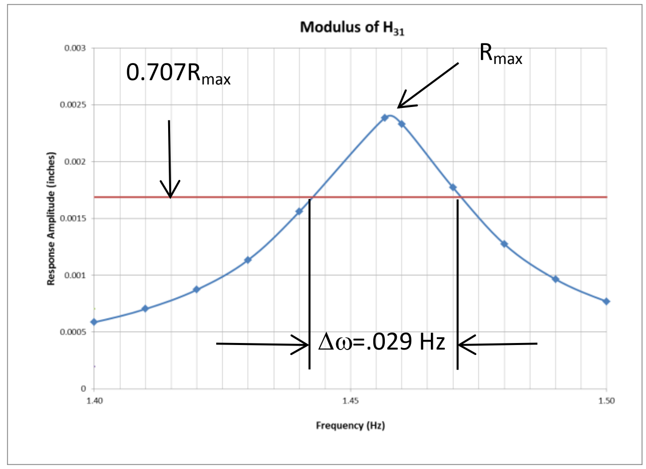

Figure 8 – Half Power Method – Mode 1

The natural frequency for the first mode is 1.456 Hz

$$\gamma = \frac{\Delta\omega}{2\omega_r} = \frac{0.029}{2x1.456} = 0.01$$

Equation 2.0

Using the same approach for the second mode, however, yields different results. The damping used for each mode is the same.

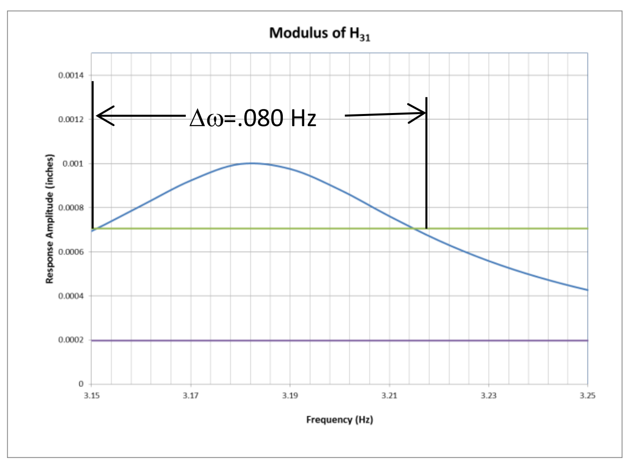

Figure 9 – Half Power Method – Mode 2

$$\gamma = \frac{\Delta\omega}{2\omega_r} = \frac{0.080}{2x3.183} = 0.013$$

Equation 3.0

For mode 2 the half power is approximately 30% higher than the actual value. If the contributions from the lower and higher modes are eliminated and only the stiffness and mass line of mode 2 is used the resulting damping factor is once again in excellent agreement with the actual amount of damping present in the mode.

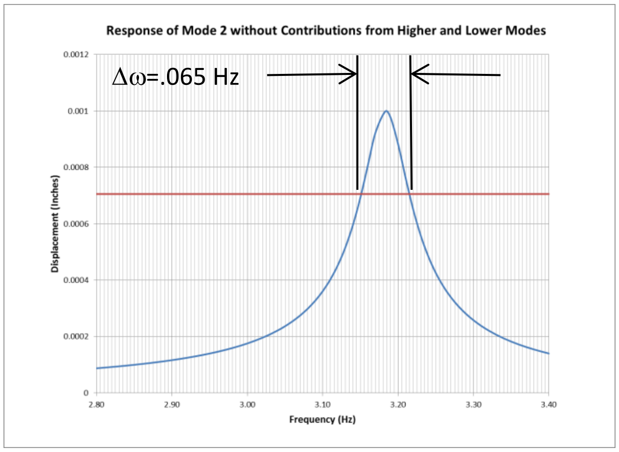

Figure 10 – Half Power Method – Mode 2 Only

$$\gamma = \frac{\Delta\omega}{2\omega_r} = \frac{0.065}{2x3.183} = 0.010$$

Equation 4.0

When the influences of the lower and higher modes are eliminated the half power method yields excellent results. This is one of the benefits of curve fitting data and extracting the modal parameters for each mode.

Considerations:

When extracting modal parameters from empirical data by curve fitting is not an option, the following considerations should be made when using the half power method to extract damping information.

- Ensure that the resolution of the data is such that the peak response at resonance has been captured. In shake table tests the resonant frequency will typically be determined by phase angle and data collected at that frequency. In lightly damped systems, a relatively small frequency offset from resonance can have a large effect on the peak response.

- Be aware of how close neighboring modes are to the one that is being evaluated. The closer the modes are the more likely they will influence the extracted damping value.

- If modes appear to be relatively close, executing an eigenvalue analysis with the mode shapes normalized to the mass matrix will provide the information for evaluating how coupled the modes may actually be. If the modes are coupled, the damping values are likely to be overestimated.

- Typically, shake table data is obtained from components that are close in design to the one being evaluated. It is always good practice to cross check the damping used in the analytical model by simulating the shake table test with the extracted damping values and comparing them to test.