CHAPTER 2

EXPERIMENTAL MODAL ANALYSIS OF FLOOR SYSTEMS

2.1 Objectives

The basis for accurately characterizing the dynamic behavior of an in-situ floor, and validating the ability of FE models to represent that behavior, relies solely on the acquisition of high quality dynamic measurements. Thus, it is important to first outline the fundamental theory of modal analysis that allows experimentalists to estimate the dynamic properties of a floor through testing. It is also important to discuss the equipment and techniques used for testing floor systems because the acquisition of high quality modal data is a product of the available equipment and its effective employment. Lastly, it would be beneficial to researchers, consultants, and building owners if there were a common language, or classification system, for floor vibration testing that allowed all parties to understand the limitations on the type, quality, and accuracy of the data obtained using the various methods for testing floors. All too often, excellent opportunities for testing floors that would contribute to the current state of knowledge of in-situ floor behavior is met with the lack of proper equipment to acquire high quality results. Equally as often, ornate FE models are presented that were "validated' using only the most basic or minimal amount of dynamic testing.

The objectives of this chapter are to address all of the above listed issues. The following sections outline the theory of modal analysis and the processing of dynamic measurements, the equipment used to acquire dynamic measurements, a description of the applied and recommended experimental modal testing techniques for floor systems, and finally a classification system for floor vibration testing is presented to aid researchers, consultants, and building owners in understanding the advantages, disadvantages, and limitations of the various methods and equipment used for investigating floor behavior. It should be noted that many of the techniques used in testing the in-situ floors in this research were based on recommendations outlined by Hanagan et al. (2003) and others (Pavic and Reynolds, 1999; Reynolds and Pavic, 2000a, 2000b) from their experiences testing in-place steel composite floors and concrete floors, respectively; however in many cases, the recommended techniques are improved or expanded upon based on the experiences of testing the in-situ floors of the presented research.

2.2 Modal Testing Theory

Important information about the behavior of the floor can be achieved through a simple measurement of the time history response. The spectral analysis of the time history response to yield its autospectrum shows the frequency content of the response. The time history response yields important indicators of peak accelerations due to various types of unmeasured excitation such as a heel-drop impulse load or walking parallel or perpendicular to the joists, a good demonstration of the floor's behavior in service. Examination of the decay from the heel-drop or a bouncing excitation can be useful in providing limited estimates of the damping in the system, however there are certain limitations to the usefulness of such "unreferenced" response measurements. Modal testing offers an expanded capability to characterize the dynamic properties of a floor by measuring both the input force and output response and forming a frequency response function (FRF) between the excitation and measurement locations. These dynamic properties (frequencies, damping, and modes shapes) estimated from experimental measurements can be used to validate analytical models that represent the dynamic behavior of the floor, usually formulated with finite element analysis. The actual process through which the FRF is computed using experimental measurements and digital signal processing requires a bit more explanation, but only in general terms as needed to explain how the process pertains to the presented research, because much more comprehensive texts are available on the subject (Ewins 2000). Before the theory behind the experimental computations is presented, a brief explanation of the theoretical analysis is warranted.

2.2.1 Analytical Modal Analysis

Ewins (2000) used three phases to describe theoretical multi-degree-of-freedom (MDOF) vibration analysis, which is based on the premise that the response of a structure is just a combination of the responses of its individual modes of vibration. The first stage is to compute the spatial properties of the model (mass, stiffness, and damping) to use in setting up the governing equations of motion. The second phase is performing a free vibration analysis to compute the modal properties of the model. The solution to the free vibration analysis yields the eigenvalues and eigenvectors of the system of equations, which correspond to the modal properties of natural frequencies and damping, and mode shapes, respectively. The last stage is a forced vibration analysis where the applied force is harmonic excitation, and the solution is a frequency response function, $H(\omega)$. The frequency response function is a complex frequency domain function that when multiplied by the frequency domain harmonic forcing function yields the steady-state frequency domain response. Alternatively, the frequency response function can be defined as the response divided by the input force,

$$H(\omega) = \frac{X(\omega)}{F(\omega)} \tag{2.1}$$

Note that ω is the circular frequency in units of radians/second and is used simply for convenience; it is interchangeable with the cyclical frequency representation, *f,* in units of cycles/second, or Hz.

The term $H(\omega)$ in Equation (2.1) is a general term often used to annotate a frequency response function. Using the same notation as Ewins (2000), a more specific designation for the quantity described in Equation (2.1) is $\alpha(\omega)$, the *displacement* response per input force, which is known as the "receptance" or "compliance." It is quite literally the ratio of the harmonic steady-state displacement response to the harmonic input force, preserving both the amplitude ratio between the two harmonic signals and the difference in phase, and thus is a complex function. Alternate forms of this are based on the response quantity used in its formulation. If the *velocity* response per input force is used, this is called the "mobility," and the notation $Y(\omega)$ is used. Similarly, if the *acceleration* response per input force is used, this is called the "inertance" or more often the "accelerance," and the notation used is $A(\omega)$. Displacement, velocity, and acceleration can all be defined by derivatives or integrals of one another, and they can all be related as follows:

$$\begin{aligned}
\alpha(\omega) &= \frac{X(\omega)}{F(\omega)} && \text{RECEPTANCE} \\
Y(\omega) &= \frac{\dot{X}(\omega)}{F(\omega)} = (j\omega)\,\alpha(\omega) && \text{MOBILITY} \\
A(\omega) &= \frac{\ddot{X}(\omega)}{F(\omega)} = (j\omega)^2\,\alpha(\omega) = -\omega^2\alpha(\omega) && \text{ACCELERANCE}
\end{aligned} \tag{2.2}$$

Note that $j=\sqrt{-1}$ and the $j\omega$ terms in the mobility and accelerance expressions in Equation (2.2) are derived from the complex representation of the harmonic forcing function, $f(t)=Fe^{j\omega t}$, and the harmonic displacement response, $x(t)=Xe^{j\omega t}$. Thus, the harmonic velocity and acceleration responses are $\dot{x}(t)=(j\omega)Xe^{j\omega t}$ and $\ddot{x}(t)=(j\omega)^2 Xe^{j\omega t}=-\omega^2 Xe^{j\omega t}$, respectively.

In a multi-degree-of-freedom system like floors, not just one frequency response function is computed, but an entire matrix called the frequency response function matrix, $[H(\omega)]$. Each term of the FRF matrix is a complex function in the frequency domain representing the force/response relationship between two degrees of freedom on a structure for a given frequency, ω. Thus, the accelerance FRF matrix is an assemblage of terms $A_{ik}(\omega)$, which relate the acceleration response of a particular location on the floor, *i*, to an input force at another location, *k*, as shown:

$$[A(\omega)]=\frac{\ddot{X}_i(\omega)}{F_k(\omega)}=\frac{a_i(\omega)}{F_k(\omega)}=\begin{array}{c}\text{Forcing Location, } k \\ \begin{bmatrix} A_{11}(\omega) & A_{12}(\omega) & \cdots & A_{1k}(\omega) \\ A_{21}(\omega) & A_{22}(\omega) & \cdots & \vdots \\ \vdots & \vdots & \ddots & \vdots \\ A_{i1}(\omega) & A_{i2}(\omega) & \cdots & A_{ik}(\omega) \end{bmatrix}\end{array}\begin{array}{l}\text{Response} \\ \text{Location, } i\end{array} \tag{2.3}$$

While the derivation of the harmonic acceleration response of a MDOF system from the uncoupled equations of motion is straight forward, it is not presented here for brevity. The fundamentally important solution is shown in Equation (2.4), which is a single term from the accelerance FRF matrix:

$$A_{ik}(\omega)=\frac{a_i(\omega)}{F_k(\omega)}=\sum_{r=1}^{R}\frac{-\omega^2(\phi_{ir})(\phi_{kr})/m_r}{(\omega_r^2-\omega^2)+j(2\beta_r\omega_r\omega)} \tag{2.4}$$

where

R = the number of modes

ω_r = the circular natural frequency of the r^{th} mode

m_r = the modal mass of the r^{th} mode

β_r = the modal damping ratio of the r^{th} mode

ϕ_{ir},ϕ_{kr} = the corresponding i or k DOF term of the the r^{th} mode shape

$[\Phi]=\begin{bmatrix}\{\phi\}_1 & \{\phi\}_2 & \cdots & \{\phi\}_R\end{bmatrix}$ = matrix of R mode shapes

Note the expression is in a form that assumes viscous damping, and although not shown here, there are more general forms that accommodate other damping models such as proportional or hysteretic damping (Ewins 2000). The denominator of Equation (2.4) holds the global properties of the structure, the resonant frequencies and damping ratios, and does not hold any information pertaining to the location of excitation or response. This information is contained in the numerator, which holds terms from the mode shape that scales the response. This fundamental analytical expression of the accelerance FRF is the basis from which all experimentally determined dynamic properties of the system are derived, as it is a mathematical expression of these quantities of interest (frequencies, damping, and mode shapes) in a formulation directly related to quantities that are measured during modal testing (acceleration response and input force).

2.2.2 Digital Signal Processing

As stated previously, the frequency response function goes by many names, but it is simply a transfer function, which is a general term for a function that describes the relationship between two simultaneously measured signals, an input signal and an output signal. For modal testing of a structure, this is generally set up with the input signal as the excitation of the structure, such as an applied force or support excitation, and the output signal as the physical response of the structure, such as displacement, velocity, or acceleration. It is the accelerance FRF between an applied force and the acceleration response of floor structures that is of interest in the presented research.

The basis of the whole process relies on the accurate measurement of signals in the time domain and Fourier analysis of those signals to transform them into the frequency domain. Fourier analysis simply states that a function in time can be written as an infinite series of sines and cosines with varying amplitudes and frequencies. Obviously anything involving infinity is fairly difficult to deal with, so several assumptions and simplifications are made along the way to make this process manageable. Rather than infinite time, only a block of time is analyzed. The signal within the time block is assumed to be periodic and is sampled a finite number of times at discrete increments, which can then be analyzed using a simplified Discrete Fourier Transform (DFT) analysis. This whole process is accomplished using a spectrum analyzer (also called a spectral analyzer, signal analyzer, or digital signal processor), a piece of equipment that receives the analog signals from the force and accelerometer transducers and immediately digitizes the signals using an analog-to-digital converter. It is this digitization that creates a discretely sampled time record of the signals, which the spectrum analyzer uses to perform a Fast Fourier Transform (FFT) on the time domain data to transform it into the frequency domain. The FFT is a very efficient algorithm for computing the DFT when the digital time record is set up in a specific way, namely the time record length must be a power of 2, such as 128, 512, 1024, 2048, 4096, or 8192. Virtually all modern day spectrum analyzers are set up this way to take advantage of the speed of the FFT (Ewins 2000). Each spectral value of the FFT is a complex number, as it represents the corresponding Fourier coefficients that belong to the sine and cosine terms of the Fourier series at the given frequency. Thus, for the two types of measured signals of interest in modal testing of floor systems, the input force f*(t)* and the acceleration response $\ddot{x}(t)$, the spectrum analyzer computes $\mathrm{FFT}_f(\omega)$ and $\mathrm{FFT}_x(\omega)$, respectively. It is at this point, with the simultaneously measured force input signals and acceleration response signals digitized in the time domain and their respective FFTs computed, where the frequency domain functions used in modal testing to describe the dynamic properties of the floor are captured.

From the FFTs of the digitized time history records, the first frequency domain function computed is the cross-spectrum of the acceleration response signal and the input force signal, $G_{xf}(\omega)$, which is a measure of the correlation between the two signals:

$$G_{xf}(\omega) = FFT_f^*(\omega) * FFT_x(\omega) \tag{2.5}$$

where

$FFT_x(\omega) =$ the FFT of the acceleration response

$FFT_f(\omega) =$ the FFT of the input force

$FFT_f^*(\omega) =$ the complex conjugate of $FFT_f(\omega)$

Because the values of the FFTs are complex numbers, the cross-spectra are complex as well, and the phase information represents the relative shift between the two signals.

The next frequency domain function computed is the autospectrum of each signal, $G_{ff}(\omega)$ for the input force and $G_{xx}(\omega)$ for the acceleration response,

$$\begin{aligned} G_{ff}(\omega) &= FFT_f^*(\omega) * FFT_f(\omega) \\ G_{xx}(\omega) &= FFT_x^*(\omega) * FFT_x(\omega) \end{aligned} \tag{2.6}$$

where $FFT_x^*(\omega) =$ the complex conjugate of $FFT_x(\omega)$

The autospectrum, which is the cross-spectrum of a signal with itself, can be seen as just a measure of the relative strength of the signal at each of the spectral frequencies at which it is computed. Note that this is a real valued function, as there is no phase shift between a function and itself (also because the product of a complex number and its complex conjugate is a real number). The autospectrum can be displayed in several formats, such as units^2/Hz (commonly called the Power Spectral Density, or PSD), but the one used in the presented research is in root-mean-squared (RMS) units, which is actually the square root of the values that are computed by $G_{ff}(\omega)$ and $G_{xx}(\omega)$. This simply puts the units of the autospectrum back into the intuitive units of the original measured signal (lbs RMS or g RMS).

Making a few assumptions on whether the uncorrelated content is on the input or on the output, the computation of the frequency response function (FRF) in its general form is:

$$H(\omega) = \frac{G_{xf}(\omega)}{G_{ff}(\omega)} \tag{2.7}$$

Equation (2.7) states that the FRF is computed by dividing the cross-spectrum of the response and excitation signal (a complex valued function that holds the phase information between the signals as well as the general correlation) by the autospectrum of the excitation signal (a real valued function that simply scales the correlation in cross-spectrum). This makes the FRF a complex valued function, complete with magnitude and phase information (or real and imaginary parts) at each spectral frequency in the units of response per unit of input excitation. For floor vibrations (including the presented research), the FRF measured is the accelerance, the acceleration response per unit of input force, and is displayed in units of in/s^2/lb for convenience of presenting the measured values, which generally have a magnitude ranging from 0 to 2 in the presented research. Other units, such as g/lb or m/s^2/N are acceptable, but they lead to much smaller values that can be difficult to work with (1 in/s^2/lb = 2.59x10^{-3} g/lb = 5.71x10^{-3} m/s^2/N).

It should be noted that while the cross-spectra and autospectra that formulate the FRFs are computed between two signals each time a measurement is taken, it is the averaging of repeated measurements that allows a level of confidence that the computed spectral values are well correlated (or not well correlated, an indicator of either high levels of noise or very little response or excitation). Thus, the coherence function is computed with each average, providing a quality check on the correlation of all of the measured data. This computation is:

$$\gamma^2(\omega) = \frac{\left|G_{xf}(\omega)\right|^2}{G_{xx}(\omega)\,G_{ff}(\omega)} \tag{2.8}$$

This function shares a very similar form to the correlation coefficient between two random variables, as it should, because the coherence function gives a measure of the linear dependence between the input excitation signal and the acceleration response (Formenti 1999). The coherence function is a real valued function because the autospectrum terms in the denominator are real valued, and the magnitude of the complex valued cross-spectrum in the numerator is also real valued. The values of the coherence function lie between 0 and 1. A coherence of 0 indicates absolutely no correlation, and a coherence of 1 indicates the two signals are perfectly correlated. It is this end of the range of coherence values, very close to 1, that the experimentalist strives for. Diligent care was taken in the presented research to ensure the highest quality measurements were obtained whenever possible.

2.2.3 Parameter Estimation

In general, modal testing can either be classified as gathering a set of *fixed-input, roving-response* measurements (one column of the accelerance FRF matrix shown in Equation (2.3), and restated for convenience in Equation (2.9)), or conversely a *fixed-response, roving-input* set of measurements (one row of the FRF matrix).

$$\begin{array}{c} \text{Forcing Location, } k \\ \left[A(\omega)\right] = \dfrac{\ddot{X}_i(\omega)}{F_k(\omega)} = \dfrac{a_i(\omega)}{F_k(\omega)} = \begin{bmatrix} A_{11}(\omega) & A_{12}(\omega) & \cdots & A_{1k}(\omega) \\ A_{21}(\omega) & A_{22}(\omega) & \cdots & \vdots \\ \vdots & \vdots & \ddots & \vdots \\ A_{i1}(\omega) & A_{i2}(\omega) & \cdots & A_{ik}(\omega) \end{bmatrix} \begin{array}{l} \text{Response} \\ \text{Location, } i \end{array} \end{array} \tag{2.9}$$

If done properly, each yields the same information for the system. The former is typically accomplished using a shaker excitation in a fixed location while accelerometers are swept across the area of interest at measurement points. The latter is usually accomplished with an accelerometer placed in a fixed position while an impulse excitation is provided across the area of interest. In this scenario, an instrumented impact hammer is traditionally used for the impulse excitation in the fixed-response, roving-input test protocol. One should note the two testing techniques are not limited to the described set-ups, but it is much less common to use an impulse excitation in the fixed-input, roving-response setup. An alternative to shaker excitation in the fixed-input, roving-response set-up is using an instrumented heel-drop on a force plate at a fixed location, which is an attractive alternative provided the same high quality data can be obtained.

Analysis of a set of measurements, the experimentally measured accelerance FRFs (one or multiple columns or rows of the accelerance FRF matrix), is the final step in the process of experimental modal analysis. These measurements are used to extract the dynamic properties of frequencies, damping, and mode shapes. This process is called parameter estimation. The different methods for conducting parameter estimation, and the theory behind the mathematics for each of the methods, is also quite vast and is better left to texts devoted to the subject. Again, a comprehensive source for parameter estimation is also found in the text by Ewins (2000), and a paper by Avitabile (2005) gives a brief overview and history of the development and implementation of the various methods of parameter estimation. Only a few comments on parameter estimation are noted in this section, as the methods used in the presented research will be addressed in the sections dedicated to presenting the experimental results or testing techniques.

Each term in Equation (2.9), $A_{ik}(\omega)$, represents the accelerance FRF of one location on the floor with respect to an input force at another location. An example accelerance FRF trace is shown in Figure 2.1. In the example shown, the accelerometer is located at the same point on the floor as the input force, thus it is known as a driving point measurement. Plotted separately are the accelerance FRF's magnitude and phase, although an alternative representation of the FRF would be plots of the real and imaginary parts. The process of parameter estimation is based on relating these FRF measurements to the mathematical formulation of the accelerance in Equation (2.4). In the most basic of terms, the peaks in the FRF indicate the presence of at least one mode, and the sharpness of the peaks indicates the level of damping of that mode. The relative magnitude and phase between the measurement shown in Figure 2.1 to accelerance FRFs of other locations around the floor characterize the mode shapes. While various methods are used for parameter estimation, they all essentially formulate a mathematical expression simulating the *measured accelerance* in terms of estimated parameters (frequencies, damping, and mode shape terms) that would closely approximate the *analytical expression of the accelerance* in Equation (2.4). This is why the term *parameter estimation* is synonymous with the term *curve fitting*. Specific methods used for curve fitting and estimating the dynamic properties of the floor are presented elsewhere in this dissertation. With a cursory review of the theoretical basis of modal analysis, the fundamentals of the digital signal processing, and parameter estimation, it is important to describe the equipment used in modal testing that enables estimates of the floor's dynamic properties.

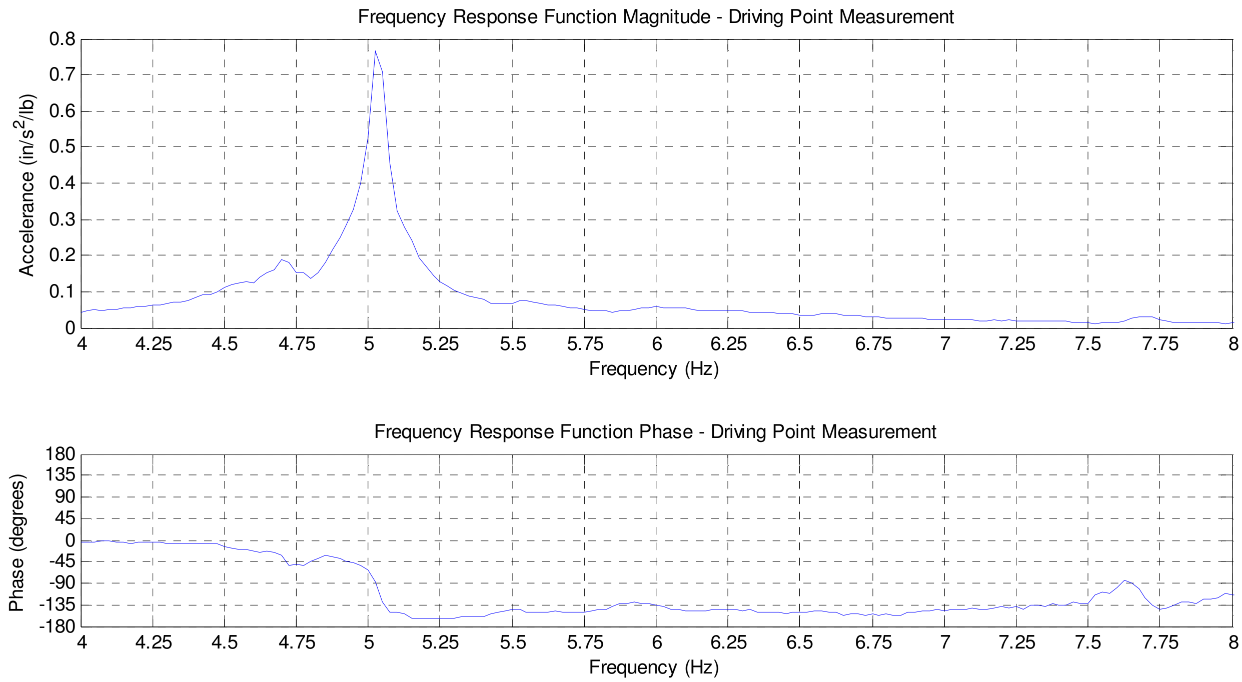


Figure 2.1: Example Accelerance Frequency Response Function Plot

2.3 Dynamic Testing Equipment

The tested floors were loaded dynamically in the fixed-input, roving response setup using an electrodynamic shaker inputting various excitation functions. A force plate beneath the shaker was used to measure the input force to the floor system and an array of accelerometers was used to measure the acceleration response. Figure 2.2 shows a schematic of the test setup.

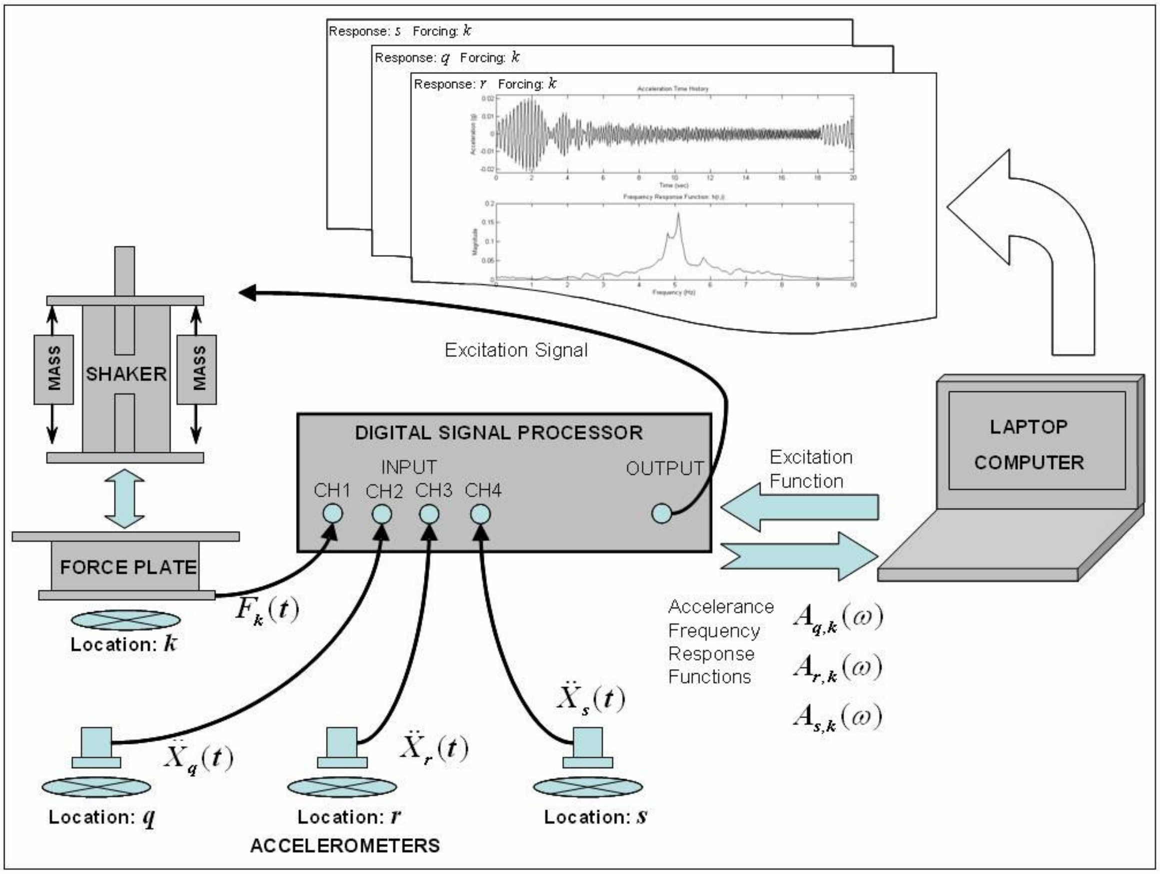


Figure 2.2: Modal Testing Schematic

To briefly summarize, the laptop computer interfaces with the digital signal processor (a multi-channel spectrum analyzer) to generate the excitation signal that is sent to the shaker. The excitation signal sent to the shaker is amplified to drive the shaker's armature and reaction mass, which in turn generates a force through the base of the shaker to the force plate and ultimately the floor system. The force plate measures the excitation force applied to the floor and sends the information back to the digital signal processor as a voltage signal. As the floor responds to the excitation force, the accelerations at various points on the floor are measured with accelerometers, which also send their signals back to the digital signal processor. With the simultaneously measured force and acceleration response signals, the digital signal processor can compute all of the various autospectra, cross-spectra, and accelerance FRFs, all of which are sent back to the laptop computer to be stored for future analysis. Outside of the traditional modal testing setup described above, unreferenced measurements from heel drop excitation were also recorded for each floor. Detailed descriptions of the testing equipment used are presented in the following sections.

2.3.1 Electrodynamic Shaker

The shaker used is an APS Electro-Seis Model 400 shaker produced by APS Dynamics, Inc. (Figure 2.3(a)), which has a frequency range of 0 to 200 Hz and a rated peak sinusoidal force of 100 lbs from 2.2 Hz to 12 Hz (APS Dynamics 400). The shaker is comprised of a stationary main core with a moving armature and suspended reaction mass blocks that have a weight of 67.4 lbs (armature and reaction mass) per the manufacturer's specifications. The entire shaker assembly static weight is 236 lbs. The shaker induces a load by taking a time-varying voltage signal and driving the reaction mass up and down to generate the applied dynamic force. A laptop computer interfacing with the digital signal processor was used to generate the excitation function as a voltage signal to send to the APS Dual-Mode Model 144 Amplifier (Figure 2.3(b)), which would send it to the shaker at an amplified level (APS Dynamics 144).

(a) APS Model 400 Shaker on Force Plate (b) APS Dual-Mode 144 Amplifier

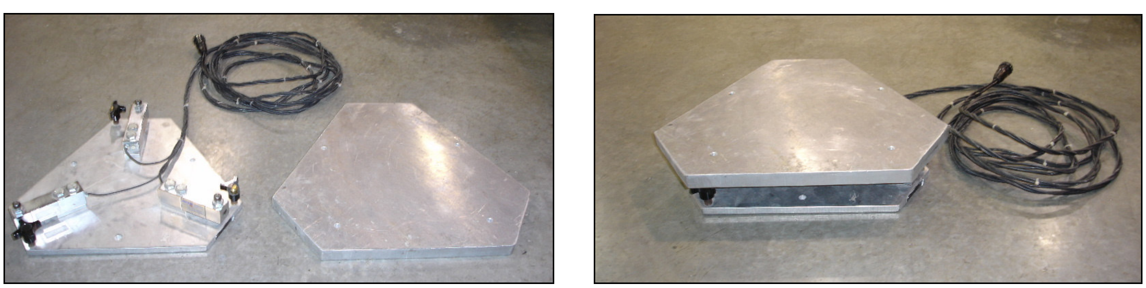
Figure 2.3: Electrodynamic Shaker and Amplifier

Although the rated peak sinusoidal force of the shaker is 100 lbs, the typical peak magnitude of the sinusoidal driving force applied during testing was much lower than that to be commensurate with expected magnitudes of applied force to simulate walking excitation (sinusoidal signals less than 25 lbs). A constant compromise was necessary, however, because it was also important to be able to introduce a large enough force into the system to incite a measurable response and to help maximize the signal-to-noise ratio by having a strong response signal.

By far, the electrodynamic shaker and amplifier are the most costly pieces of equipment used for floor testing. However, the advantages of shaker excitation over the various forms of impulse excitation, like a force hammer or instrumented heel-drop, far outweigh the high cost. More specifically, shaker excitation allows better signal-to-noise ratio throughout the time sample, it can produce controlled excitation within a specified frequency range of interest, and its variety of forcing inputs such as steady-state sinusoidal, chirp, and random is unparalleled (Mayes and Gomez 2005). With the Model 400 shaker weighing in at 236 lbs, it was fairly hefty for the individuals moving it to different locations around the floor, however its localized mass was considered negligible to the system as the tested floors had a total weight nearly four orders of magnitude larger.

2.3.2 Measurement of Input Force - Force Plate

A force plate was placed between the shaker and the floor to measure the time history of input force. The force plate used was the only non-commercial piece of equipment, and it was specifically fabricated for floor vibration testing. The force plate consists of three Nikkei Model NSB-500 shear beam load cells, rated at 500 lbs each, mounted in a triangular orientation to support a 1-in. thick triangular aluminum plate (Figures 2.4(a) and (b)). A summing amplifier was used to collect the voltage output of the three individual load cells, integrate and amplify the signal, and pass it along to the digital signal processor as a single voltage signal representing the total input force on the force plate (Howard et al. 1998). This force input signal served as the reference signal from which all accelerance FRF measurements were based.



(a) Force Plate Shear Beam Load Cells (b) Force Plate with Top Plate

Figure 2.4: Force Plate

The force plate was used to measure the force as opposed to monitoring the shaker voltage signal or placing an accelerometer on the armature of the shaker and using an acceleration computation. Because the shaker itself is a mass-spring-damper system, it has inherent dynamic properties that affect the output force for a given input voltage (Hanagan et al. 2003). An accelerometer attached to the moving armature of the shaker measures an *absolute* acceleration, but it is the *relative* acceleration between the floor and the reaction mass that induces the dynamic load. A force plate directly measuring the input force at the shaker-floor interface avoids these issues.

The force plate signal output is a measured voltage, thus a calibration factor was determined to convert the voltage into physical force units. This voltage-to-force calibration factor, C, has the units of lbs/volt and was multiplied by the voltage time history to convert it into physical force units. It was also used in the conversion of the accelerance FRFs from the measured units of volts/volt to physical units of in/s^2/lb. The following is a brief derivation of how this calibration factor *C* was determined:

$$
\begin{aligned}
&F_{(lbs)} = F_{v_{(volts)}} \cdot C_{(lbs/volt)} = mass \cdot acceleration = \frac{W_{(lbs)}}{g} \cdot a_{(g)} \\
&C_{(lbs/volt)} = \frac{W_{(lbs)}}{\cancel{g}} \cdot \frac{a_{(\cancel{g})}}{F_{v_{(volts)}}} = W_{(lbs)} \cdot \frac{a}{F_{v_{(volts)}}} \\
&S_{(1/volts)} = \frac{a}{F_{v_{(volts)}}} \\
&C_{(lbs/volt)} = W_{(lbs)} \cdot S_{(1/volts)} = 67.4_{(lbs)} \cdot S_{(1/volts)}
\end{aligned}
\tag{2.10}
$$

where

$C =$ voltage-to-force calibration value $(lbs/volt)$

$S =$ ratio (or spectral value) of armature acceleration to force plate output voltage $(1/volts)$

$W =$ 67.4 lbs, the weight of the shaker armature and reaction mass (lbs)

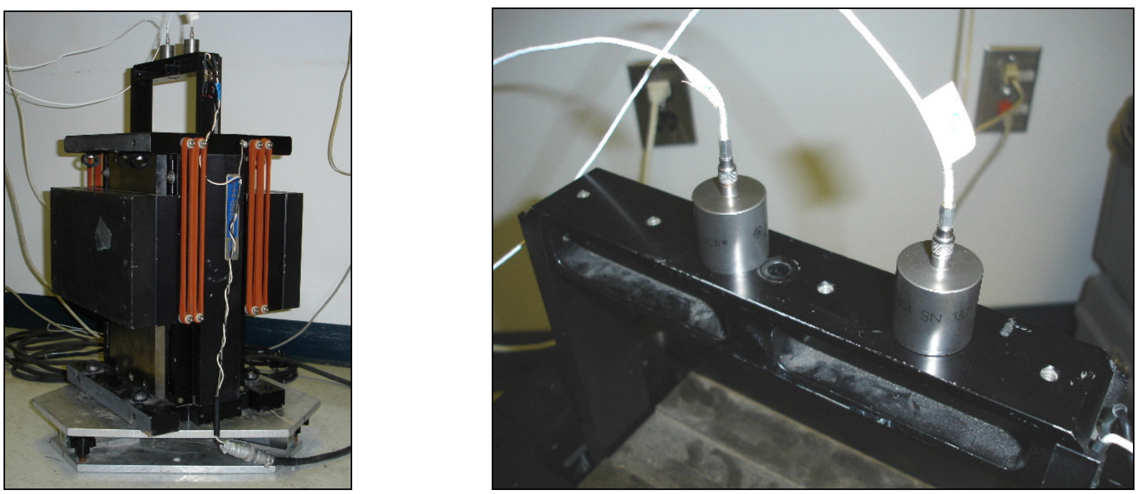
$F_v =$ measured force plate output voltage $(volts)$

$F =$ computed force plate output in physical force units (lbs)

$a =$ measured armature acceleration in units of g or $volts$, since $1\ g = 1\ volt$ (g)

$g =$ acceleration of gravity (in/s^2)

Derivation (2.10) states that the applied force is simply equal to the mass of the moving part of the shaker (the armature and attached reaction blocks) multiplied by the acceleration of the moving part of the shaker. The mass (or weight) of the armature assembly is known as 67.4 lbs. By mounting an accelerometer to the armature assembly, the acceleration time history of the mass was recorded. This acceleration time history was multiplied by the armature mass to serve as the baseline force time history. Note that the shaker and force plate must be on a rigid surface, such as a slab on grade or adjacent to a column, to measure (as closely as possible) the true acceleration of the armature and not the relative acceleration between the armature and a moving structure (such as a floor). The applied force of the moving mass was measured by the force plate output voltage, and thus a calibration factor, *C*, was determined by dividing the computed force (mass times acceleration) by the measured force voltage. From this method of calibration testing, the calibration value *C* used for the presented research was 240 lbs/volt. Figure 2.5 shows a calibration test performed on a slab on grade with the shaker on the force plate and accelerometers mounted to the armature of the shaker.



(a) Shaker/Force Plate on Rigid Surface (b) Accelerometers Mounted on Shaker Armature

Figure 2.5: Force Plate Calibration

2.3.3 Measurement of Acceleration Response – Accelerometers

Acceleration response measurements were taken using PCB Model 393C seismic accelerometers produced by PCB Piezotronics (Figure 2.6). The 393C accelerometer has a frequency range of 0.025–800 Hz and a sensitivity of 1.0 volt/g, which made it easy to judge real-time accelerations as a percentage of gravity. Depending on the number of digital signal processing units utilized, arrangements of three to seven accelerometers were used to simultaneously capture multiple response measurements. Each accelerometer weighs 31.2 ounces, and with measured peak accelerations less than 0.10g, the effect of gravity prevented the need for any mechanical fastening of the accelerometers to the floor structure, although Hanagan et al. (2003) recommends fixing accelerometers to the structure using beeswax or clay.

Figure 2.6: PCB Model 393C Seismic Accelerometer

While testing the in-situ floors, accelerometers were placed on very thin and stiff rubber bearing pads to minimize the possible rattling effect due to small debris or imperfections in the unfinished concrete surface (Figure 2.6). The stiffness and small thickness of the pad were such that measured accelerations were not affected. This was verified from tests run on a laboratory test floor. In these tests, the laboratory test floor was driven sinusoidally at acceleration levels ranging from 1% of gravity to 10% of gravity with the accelerometer either resting on the bare steel floor surface, resting on a stiff rubber bearing pad, or fastened to the floor using removable putty adhesive. Nearly identical acceleration values were recorded for each case. The large flat base of the accelerometer also alleviated the need for any additional leveling devices to ensure the measurements were taken perpendicularly to the floor surface.

2.3.4 Cables

One aspect to modal testing large in-situ structures that is often overlooked in the literature is the cable requirements for connecting the components of the test setup. The higher the channel count (i.e. the more channels used in the modal test) and the larger the test area, the more cables are required. Hunt and Brillhart (2005) claim that accelerometer cables are often the weakest link in the modal test setup, especially microdot, which can be easily broken if they are bent or kinked into sharp angles. The PCB 393C accelerometers required microdot connections, and the digital signal processors had BNC connector inputs, thus a microdot-to-BNC cable was required. Microdot cable is the most expensive cable type, and as stated above, likely the most fragile (Hunt and Brillhart 2005). Because of the long cable runs and austere conditions of the tested floor systems that were under construction, heavier and more rugged coaxial cables with BNC connectors were fabricated for the long runs to the accelerometers. Short 10-ft lengths of microdot cable (with microdot/BNC connections) were hooked up to the accelerometers and left coiled to protect the fragile cables from wear and kinks. Figures 2.6 and 2.7 show the microdot cable in its coiled position. Heavier and more durable coaxial cable was connected to the microdot for the long cable runs. These cables were assembled from combinations of 50-ft, 75-ft, and 100-ft lengths, typically running for 150-ft to 200-ft total length. Whenever possible, the number of splices was kept to a minimum to reduce the locations where a malfunction could occur.

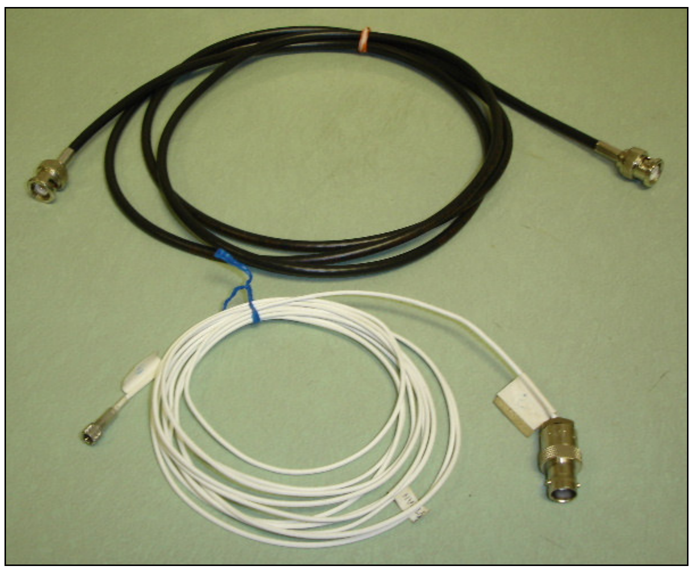


Figure 2.7: Accelerometer Cables – Coaxial (above) and Microdot (below)

The majority of the time required to test a floor system is the time required for taking actual dynamic measurements, however a close second is cable management, especially when long cable runs are involved and a high channel count is used. In one of the floor systems tested, a total of 1700 ft of cable was moved around the floor for each measurement (seven accelerometers with 200-ft cables and two 150-ft cables for the shaker and force plate). In performing a modal sweep across an area of the floor, the accelerometers are constantly moved from one response point to another for the next measurement. Extreme care was taken to ensure the cables were laid out in a fashion so that they would not tangle each time the accelerometers were moved.

2.3.5 Multi-Channel Spectrum Analyzer

The digital signal processing (DSP) equipment consisted of Model 20-42 SigLab units produced by Spectral Dynamics, Inc. (Figure 2.8). These multi-channel DSP units are fully functioning spectral analyzers and have the ability to record and compute the full suite of dynamic measurements, including (but not limited to) the time history response, autospectrum, cross-spectrum, frequency response function, and coherence. These devices have four input and two output channels per unit, with the capability of being connected in series to provide up to 16 input channels. A standard laptop computer running SigLab v3.28, a front-end program driven by MATLAB, was used to control the DSP unit to generate the output excitation signal for the shaker and process the input measurements from the force plate and accelerometers. The SigLab units have a frequency bandwidth of 5 Hz to 20 kHz, although for floor vibration testing only the 10 Hz and 20 Hz bandwidth selections were used. The units are capable of generating a variety of output excitation signals for the shaker, including sine, square, sawtooth, triangle, impulse, random, and chirp. Only the steady-state sine and chirp functions were used in the presented research.

(a) Four-Channel DSP Unit (b) Two DSP Units in Series and Laptop

Figure 2.8: Model 20-42 SigLab Digital Signal Processor

For most test setups in the presented research, two SigLab DSP units were connected in series to provide eight input channels (Figure 2.8(b)). The advantage of a higher channel count was that more response measurements could be taken simultaneously, reducing the time to test a given area of the floor. The only disadvantage was that there were more measurements to keep track of and an increased difficulty in keeping the cables untangled, however the advantages far outweighed the disadvantages. During modal testing using the shaker, input Channel 1 was always designated as the reference channel for the FRF measurements and was connected to the force plate. The remaining input channels were connected to the array of accelerometers, and thus the accelerance FRFs were computed.

Despite having the capability to compute the FRF, a multi-channel spectrum analyzer does not have to be used in this capacity and can easily serve as simply a multi-channel response-only measurement tool. While not as sophisticated as true modal test measurements (i.e. referenced), simple unreferenced response-only measurements can be useful in evaluating floor systems and their dynamic properties. In some testing situations, response-only measurements may be the only measurements available because capturing the input force is not practical or possible; such is the case for walking excitation. In this situation, an array of accelerometers simultaneously measuring the response of an area of interest from an unreferenced walking excitation can still yield important information about the in-service behavior of the floor.

If a multi-channel spectrum analyzer is available but there is no way to measure an input force (i.e. no force plate or shaker with mounted accelerometer), all hope is not lost if there is still an interest in quantifying the general shape(s) of the floor when excited dynamically. Although not used in the presented research, there is a measurement type called an Operating Deflection Shape (ODS) FRF, which is simply an FRF taken with the reference signal being an acceleration measurement rather than a force measurement (Richardson 1997). While not nearly as fully descriptive as a force-referenced FRF, the result is a relative magnitude and phase difference between two responses on a floor. This could potentially be used in floor testing by conducting unreferenced heel drops at the middle of a bay next to the reference accelerometer, while the roving accelerometers are placed in the middle of adjacent bays (or other locations). The ODS FRF should reflect the relative magnitude of the responses in reference to the response at the excitation location, as well as the difference in phase. It should be stressed that this would only yield information about the frequencies and relative shape, and no information about the acceleration response per input force, which is key in validating FE models.

The available record lengths of the DSP units used in the research ranged from 64 samples to 8192 samples over the time block. A selection of DSP settings and the resulting time and frequency domain resolutions are presented in Table 2.1. The frequency domain resolution and time domain resolution of the measurements were products of the selected frequency bandwidth and record length. A fine frequency domain resolution comes at a cost of a longer time record. The entries of Table 2.1 in bold print represent settings used at various times for the floors tested in the presented research.

Table 2.1: Digital Signal Processor Settings

Bandwidth (Hz)	**Record Length** (samples)	**Frequency Resolution** Δf (Hz)	**Record Length** (sec)	**Time Resolution** Δt (sec)
10	512	0.05	20	0.0390625
20	512	0.10	10	0.01953125
50	512	0.25	4	0.0078125
100	512	0.50	2	0.00390625
10	**1024**	**0.025**	**40**	**0.0390625**
20	**1024**	**0.05**	**20**	**0.01953125**
50	1024	0.125	8	0.0078125
100	**1024**	**0.25**	**4**	**0.00390625**
10	2048	0.0125	80	0.0390625
20	**2048**	**0.025**	**40**	**0.01953125**
50	2048	0.0625	16	0.0078125
100	2048	0.125	8	0.00390625
10	4096	0.00625	160	0.0390625
20	4096	0.0125	80	0.01953125
50	4096	0.03125	32	0.0078125
100	4096	0.0625	16	0.00390625
10	8192	0.003125	320	0.0390625
20	8192	0.00625	160	0.01953125
50	8192	0.015625	64	0.0078125
100	8192	0.03125	32	0.00390625

2.3.6 Single-Channel Spectrum Analyzer

An extensive database of problem floors has been accumulated over the years for which only subjective analysis exists. In many of the floors evaluated, the only measured data are acceleration response time histories from unreferenced heel drop, bouncing, and/or walking excitations. In such cases, as well as the tested floors in the presented research, the response-only measurements were taken using a single-channel spectrum analyzer, the Ono Sokki CF-1200 Handheld FFT Analyzer, and a PCB 393C seismic accelerometer (Figure 2.9).

Figure 2.9: Ono Sokki CF-1200 Handheld FFT Analyzer

The CF-1200 has an available frequency bandwidth of 100 Hz to 20 kHz and a maximum time record length of 1024 samples. For all measurements in the presented research, the handheld analyzer was set to its finest resolution, a 100 Hz bandwidth with record length of 1024 samples, resulting in a four second acceleration time history block, a time domain resolution of 0.003906 sec., and a frequency domain resolution of 0.25 Hz (i.e. a frequency accuracy of ± 0.125 Hz). Although unparalleled in its portability and ease of use, the only weakness of the handheld analyzer for use in capturing unreferenced response measurements is its coarse frequency resolution.

2.4 Experimental Testing Methods

Almost every aspect of experimental dynamic testing of in-situ floor systems is a compromise. The compromises are based on a variety of influences, most notably the time and equipment available for testing. Limitations on the types of information achievable based on the available equipment are obvious, however the time allowed for testing is the critical factor because time has the greatest influence on the quality, quantity, and level of detail of the dynamic measurements. In many cases, the desired test areas are large, with perhaps hundreds of test points. A fine frequency resolution is often critical, particularly when damping values are very low, and an accurate representation of the peak magnitudes of the accelerance FRF is required. Fine frequency resolution comes at the cost of longer record lengths. A common trait of in-situ structures is the presence of extraneous noise, which may require a higher number of averages during testing to improve the quality of FRF measurements. Large test areas, numerous test points, long record lengths, and high number of averages all take time. Thus, with a limited amount of time to test a floor system, these are all aspects of testing that lend themselves to compromise. In most cases, compromise is made by reducing the area to be tested, using a coarser grid of test points, and finding a balance between frequency resolution, time record length, and the number of averages that will provide a set of measurements of an acceptable quality.

The best way to address the inevitable compromise is by clearly defining the overall objectives of the dynamic testing of the floor system and match those objectives with the time available for testing. If the objective of the testing is to provide a reasonable estimate of frequencies and to capture behavior under service conditions from individuals walking, bouncing, or performing heel drops, then perhaps simple unreferenced (response-only) measurements are needed. On the other end of the spectrum, the objective may be to gather a set of high quality measurements for a fine mesh of test points over an area, perhaps for developing a detailed model for use in extensive response simulation. This would require extensive modal testing with an electrodynamic shaker, and plenty of time. In most cases, such as the presented research, the objectives lie somewhere in between. The objectives were to capture enough measurements to adequately estimate the modal properties for use in validating an FE model of the floor. This translates into identifying the resonant frequencies, estimating the level of damping in the floor, and determining the mode shapes for a reasonable comparison with the ones generated by an FE model.

While all measurements satisfy the overall objective of quantifying the dynamic behavior of the floor to some degree, different types of measurements and methods of testing have different goals. In the traditional modal testing setup using a shaker, the chirp signal (swept sine) or other types of broadband excitation (instrumented heel drop) is used to measure the accelerance FRF over a certain frequency range of interest. Steady state sinusoidal excitation is used primarily to verify the accelerance FRF values at specific spectral lines (frequencies) and also serve as the initial excitation for decay measurements, useful for damping estimates. Unreferenced response-only measurements have their place in measuring in-service accelerations or the autospectrum of response, which in some cases can capture low frequencies of a structure that are not possible with a shaker due to its own limiting dynamics. Any of the above listed measurement types can be used to measure the response, referenced or unreferenced, at long distances from the point of excitation. The following sections describe the acquisition and use of the different types of measurements for testing floors, including best practice techniques recommended by others and expanded upon from the experiences of testing the in-situ floors in the presented research.

2.4.1 Chirp Signals

The chirp signal, also known as a swept sine, is a sinusoidal function driving at a changing frequency over time to provide force input to the floor system over a range of frequencies. This is a quick and controlled method to get the frequency response, whereas other broadband methods such as impulse excitation can create measurement difficulties by exciting a wider range of frequencies. The chirp signals can sweep from lower frequencies to higher frequencies, or vice versa. Theoretically the resulting accelerance FRF should be identical for either direction of the sweep; however the most common arrangement is to sweep from a lower to higher frequency because lower frequencies generally take longer to die out. This maximizes the chances that all the response will be captured within the time record. The goal of a chirp signal measurement is to provide a controlled excitation within a specific frequency band so that the accelerance FRF can be clearly defined within this range of frequencies without the effect of out-of-range frequencies. The controlled level of excitation and limited range of frequency make it much easier to capture well correlated accelerance FRFs with high coherence, indicating quality measurements. Two types of chirp excitation were used to derive the accelerance FRFs in the presented research, a continuous chirp signal and a burst chirp signal. The type of chirp signal, the sweep frequency range, and the chirp duration were varied for each of the three tested floors in the presented research and will be discussed later.

Continuous Chirp - A continuous chirp signal is where the signal sweeps between two frequencies over a period of time and continuously repeats itself without pause. Hanagan et al. (2003) points out that if the chirp sweep cycle matches the data acquisition time, there is a decrease in signal processing errors. A typical time history and autospectrum of a 4-12 Hz continuous chirp excitation signal is shown in Figure 2.10. As suggested, the duration of the chirp exactly matches the time block. The average peak magnitude of force was around 25 lbs, with an occasional surge when the continuous chirp cycle would reinitialize to 4 Hz. As the chirp cycle finished its sweep to 12 Hz, it would immediately start the cycle over at 4 Hz, as shown in Figure 2.10 at the 18-second point.

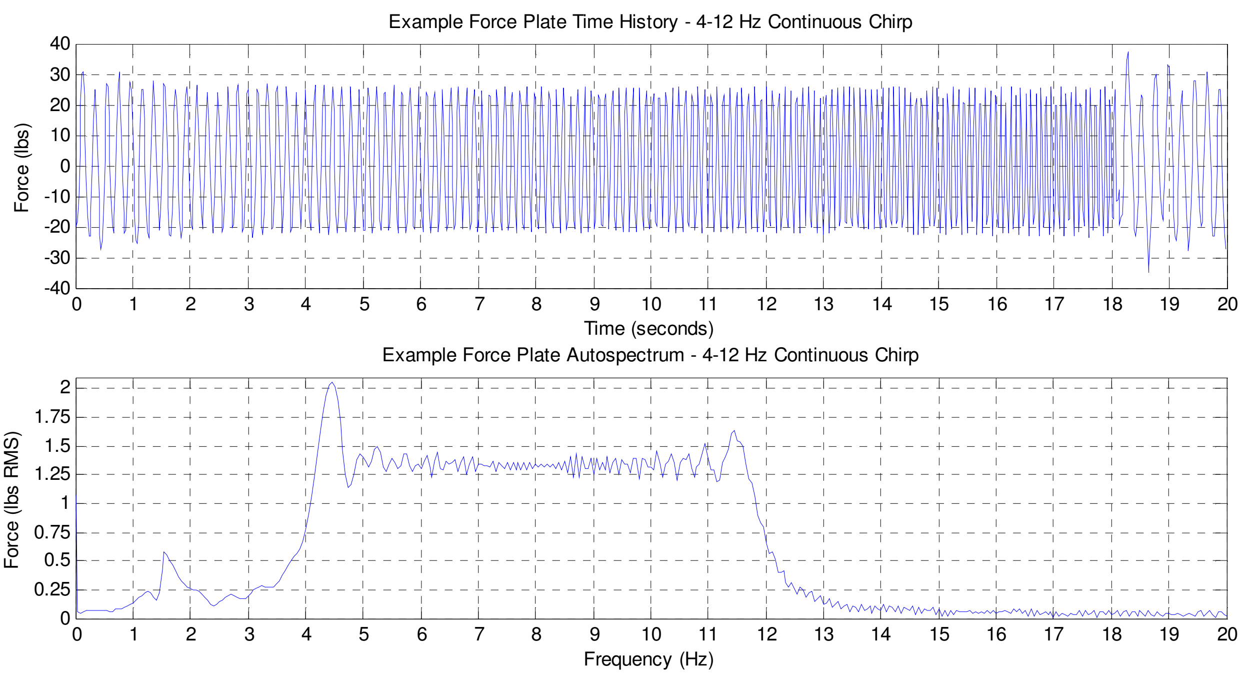


Figure 2.10: Typical 4-12 Hz Continuous Chirp Force Input Signal

The start of a measurement was manually controlled and an attempt was made to initialize at the beginning of the measurement to coincide with the chirp cycle starting over at 4 Hz. Despite the best efforts to start the measurement right at the transition, most measurements started around two seconds after reinitializing, highlighting one disadvantage of the continuous chirp measurement. Although various measurement trigger methods are available in the spectrum analyzer to correct this, they were not employed in the above measurement. Figure 2.10 also shows the autospectrum of the chirp signal time history, as noted by the roughly constant level of energy introduced into the system within the 4-12 Hz frequency range of the chirp. The choice of the chirp frequency range was based on previous unreferenced response-only measurements that indicated all frequencies of the floor were greater than 4 Hz. The 12 Hz upper end of the sweep was used because there was no significant response around that frequency and natural frequencies above 10-12 Hz are not typically of interest for walking induced vibrations as they are unlikely to be excited by one of the higher harmonics of walking.

Figure 2.11 is the time history and the autospectrum of the acceleration response of the floor at the driving point (response at location of excitation) and is typical of the floor response at other locations. The time history shows the varying response as the floor is excited by the sweeping frequencies, and the autospectrum of the acceleration response clearly shows the vast majority of the response to a chirp signal is within the targeted 4-12 Hz frequency range of interest and very little at other frequencies. The autospectrum in Figure 2.11 also highlights the frequency of the tested bay from the dominant peak response at 5.05 Hz.

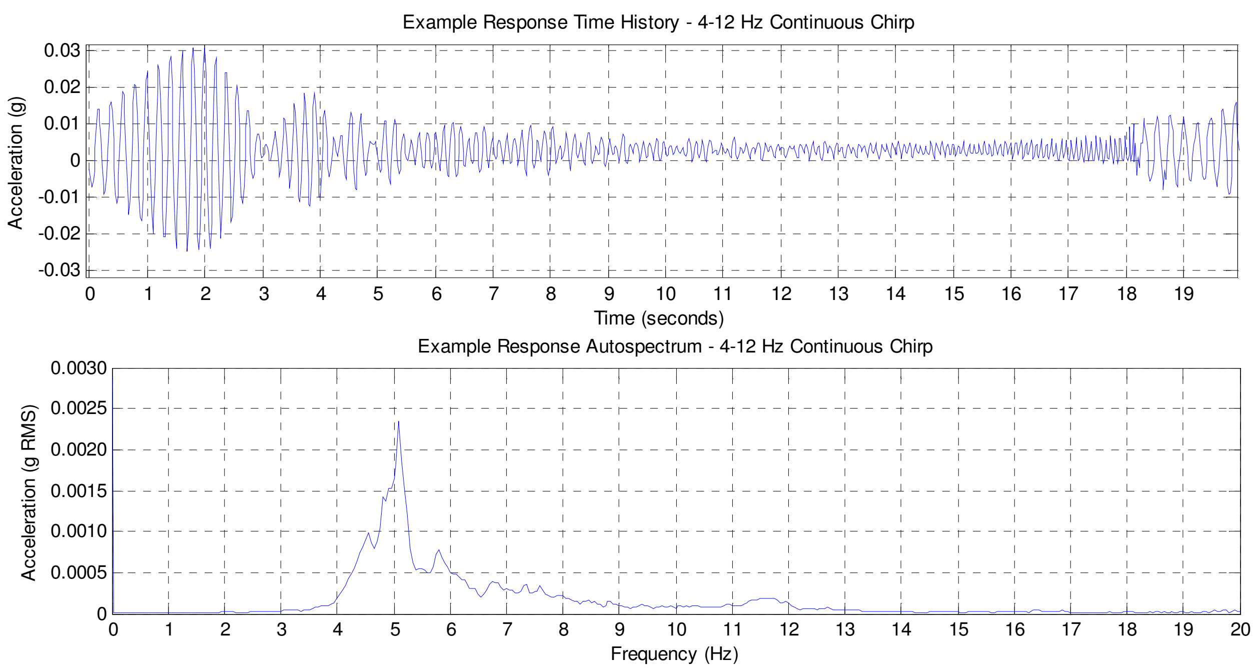


Figure 2.11: Typical 4-12 Hz Chirp Signal Acceleration Response

With the input force and acceleration response of the system concentrated within the 4-12 Hz chirp sweep, it was expected to have well correlated data within this range and not much elsewhere. Because there is negligible force input and acceleration response outside the range of the chirp signal frequencies, poor correlation for accelerance FRF measurements is expected in these areas, as indicated by the poor coherence outside 4-12 Hz in Figure 2.12, and good coherence within 4-12 Hz as shown in Figures 2.12 and 2.13.

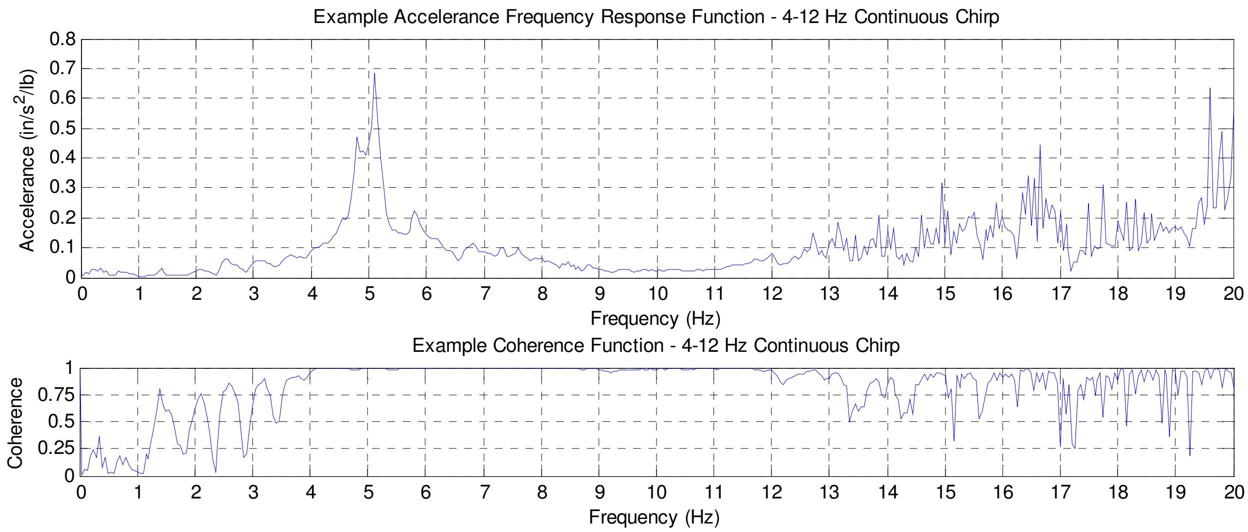


Figure 2.12: Typical 4-12 Hz Continuous Chirp FRF Magnitude and Coherence (0-20 Hz)

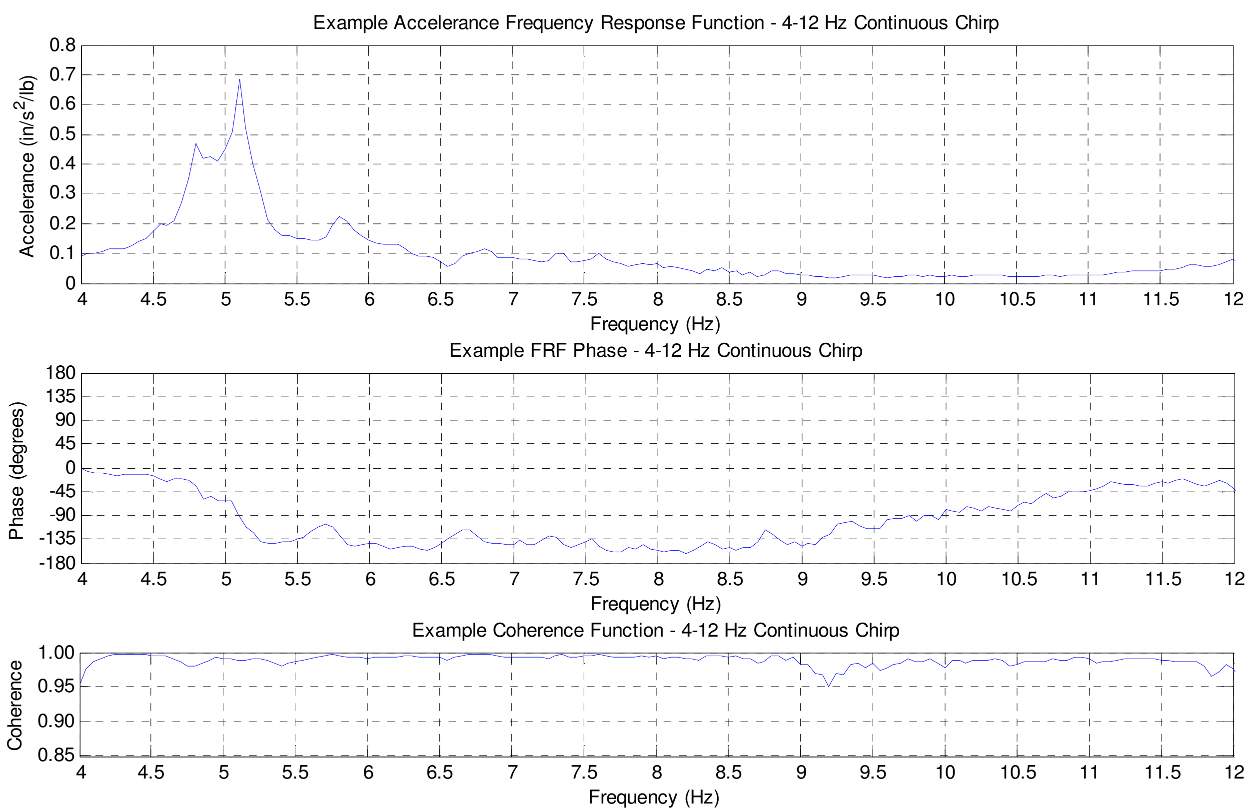


Figure 2.13: Typical 4-12 Hz Continuous Chirp FRF Magnitude, Phase, and Coherence

With poorly correlated data outside the 4-12 Hz range, future frequency domain plots (autospectra, accelerance FRF magnitude and phase, and coherence) resulting from chirp signal excitation will display just the usable frequency range of interest, unless otherwise noted. The accelerance FRF magnitude, phase, and coherence for the example driving point measurement are presented within the chirp frequency range in Figure 2.13.

It should be noted that the nature of obtaining accelerance FRFs is based on averaging measurements; more specifically, the cross-spectra and autospectra are updated with each average using the most recently recorded input force and acceleration response measurements. When frequency domain plots are presented (autospectra, accelerance FRFs), they reflect the averaged functions. When time domain measurements (force and acceleration time histories) are presented, they reflect the last of the averaged measurements that were recorded. The number of averages needed for a measurement is a product of the compromise noted earlier, but the minimum number should be whatever ensures quality (well correlated) measurements as indicated by good coherence. From preliminary measurements with the equipment, it was determined only three averages of each chirp cycle were needed to ensure the quality of the measurement within the frequency range of interest, and a higher number of averages did not improve the coherence, only lengthened the time required to take the measurement.

While the continuous chirp excitation signal seemed to produce accelerance FRF measurements of acceptable quality, it is not recommended for testing floors. As recommended by Hanagan et al. (2003), the duration of the chirp sweep was 20 seconds, equal to the time record length. Even though the continuous nature of the excitation and response is intuitively suspect in the computation of the FRF, the assumptions made in the digital signal processing make it possible to achieve measurements of acceptable quality. Specifically, the input force and acceleration response are assumed to be periodic within the time window, which occurs if the chirp duration exactly matches the time record length because consecutive time records should be identical. Secondly, well correlated measurements are assumed to be a product capturing the entire force input and the entire acceleration response. In the continuous chirp signal, the response at the beginning of the time record is due to the previous input force (not measured), and the force at the end of the time block produces response that occurs after the end of the time block, which intuitively suggests the measurement should be poor. However, the periodicity of the force and response time blocks help because the FRF is based on *frequency domain* representations of the signals, which are not concerned with the actual order of the input and response signals in the *time domain*. Again, despite the capability to produce acceptable quality FRF measurements, the continuous chirp signal is not recommended for testing floors. A better alternative for measuring accelerance FRFs of floor systems is a burst chirp excitation signal, which takes advantage of all the chirp signal concepts and has proven to provide superior quality measurements.

Burst Chirp Signal **–** For a burst chirp signal, the floor system initially starts at rest and is excited as the chirp signal sweeps between two frequencies over a period of time. The duration of the chirp is shorter than the overall length of the time record, allowing the floor to come to rest before the end of the record. In this manner, the entire force and response time histories are captured within each time record, maximizing the chances of computing high quality and well correlated FRFs within the chirp frequency range. While adjustments of the chirp duration and sweep frequency range must be made to accommodate the frequency range of interest and selected record lengths, the burst chirp signal is a highly recommended method of excitation for floor vibration testing due to its flexibility and superior quality measurements. A typical time history and autospectrum of a 4-12 Hz burst chirp excitation signal is shown in Figure 2.14.

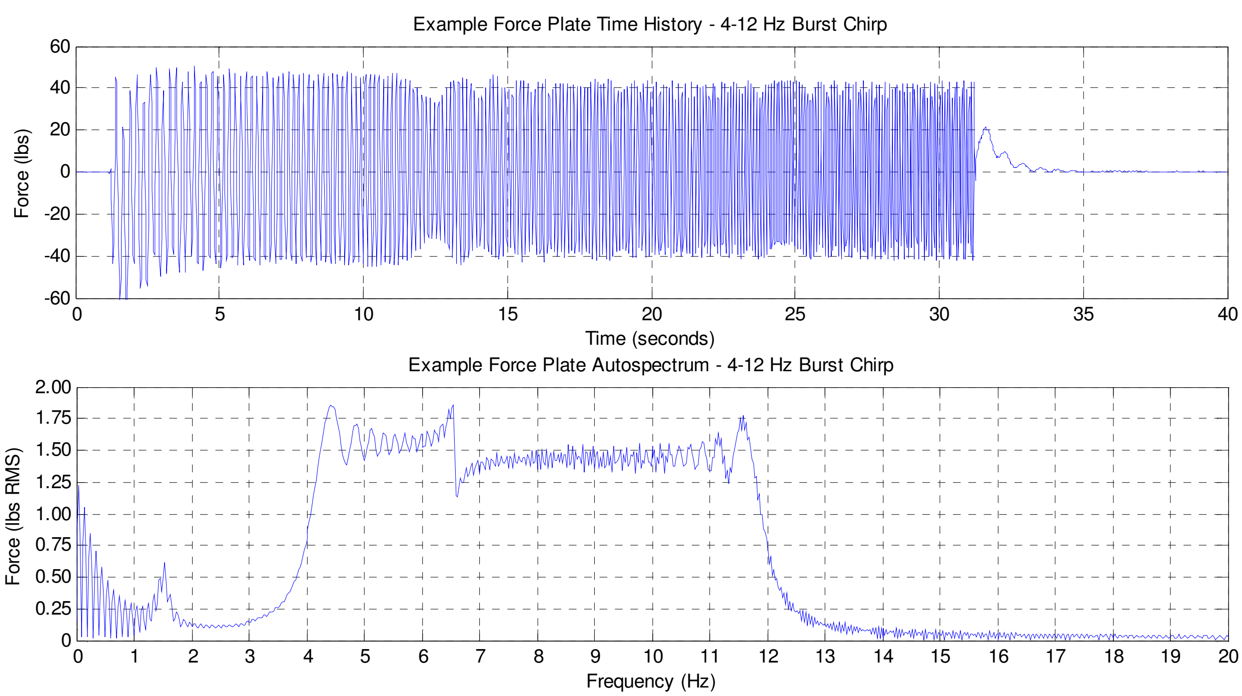


Figure 2.14: Typical 4-12 Hz Burst Chirp Force Input Signal

The small peak in the force input autospectrum around 1.5 Hz corresponds to the natural frequency of the shaker's reaction mass and suspension bands. This force input is visible at the 31-second point of the time history in Figure 2.14, where the chirp signal ends and the shaker's reaction mass comes to rest over a few seconds. This has no effect on the measurement. The measurement was set up with a 40-second time record, and the duration of the chirp sweep was 30 seconds, with a 15-second idle period before the next burst chirp signal, allowing the floor to come to rest before the next measurement. Note that the 30 seconds of chirp signal and 15 seconds of off time are longer than the 40-second time record length. This technique served two purposes. First, it enabled the use of the triggering mechanisms in the SigLab DSP units. The triggers are cues for starting a measurement and are based on the signal from an input or output channel. For the burst chirp measurement, the trigger for every average was set to start a measurement when the output channel providing the signal to the shaker started the chirp. Additionally, the SigLab DSP unit has the capability for positive or negative delays with the use of the trigger. In the chirp signal of Figure 2.14, a negative 2.5% delay was used. This informed the DSP unit that the start of a time block to be recorded was 2.5% of the total time record length prior to the trigger mark (1 second for a 40-second record).

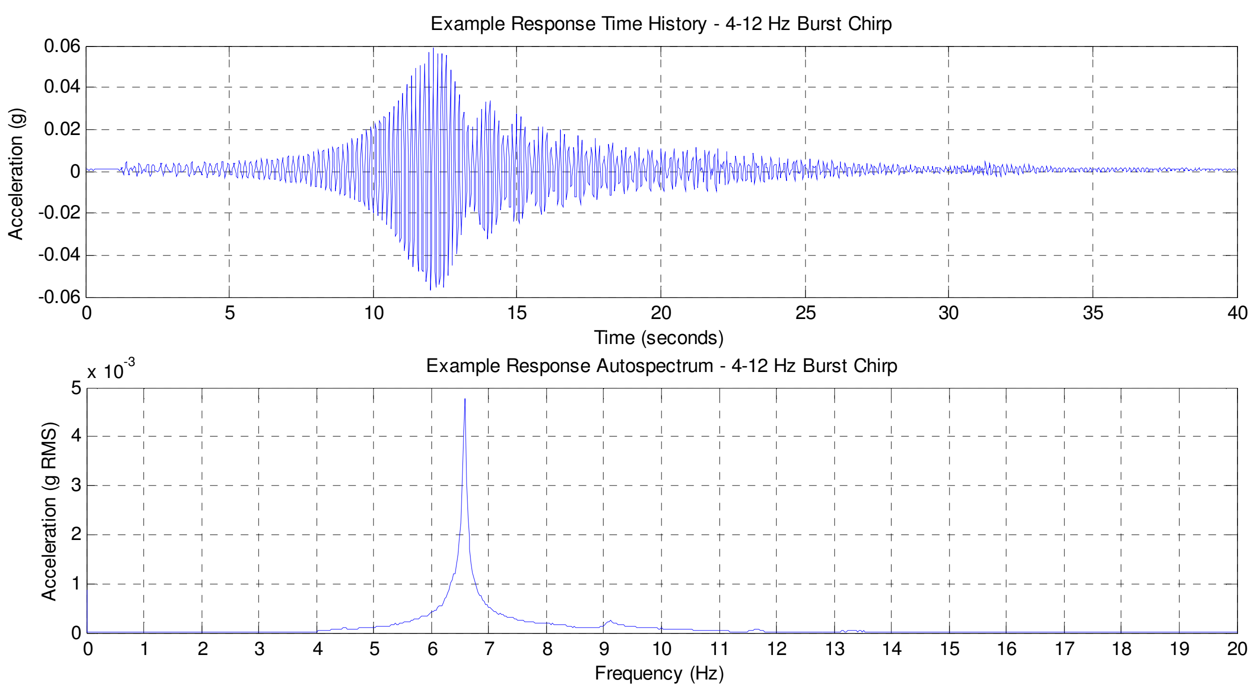


Figure 2.15: Typical 4-12 Hz Burst Chirp Acceleration Response

Using this burst chirp technique, as shown for the input force of Figure 2.14 and the corresponding acceleration response of Figure 2.15, the measurement has one full second of no input or response, 30 seconds of a burst chirp signal sweeping from 4 to 12 Hz, followed by no input force and decaying response for the remainder of the 40-second record. Note that both the input force and acceleration response are completely captured within the time record block. Nothing is measured during the extra five seconds the shaker is off, which gives the floor additional time to come to rest before the next record. The driving point accelerance FRF magnitude and coherence for the full 20 Hz bandwidth is shown in Figure 2.16. Note the impeccable coherence within the 4-12 Hz frequency range of the burst chirp signal and a clearly dominant peak at the bay's resonant frequency of 6.575 Hz.

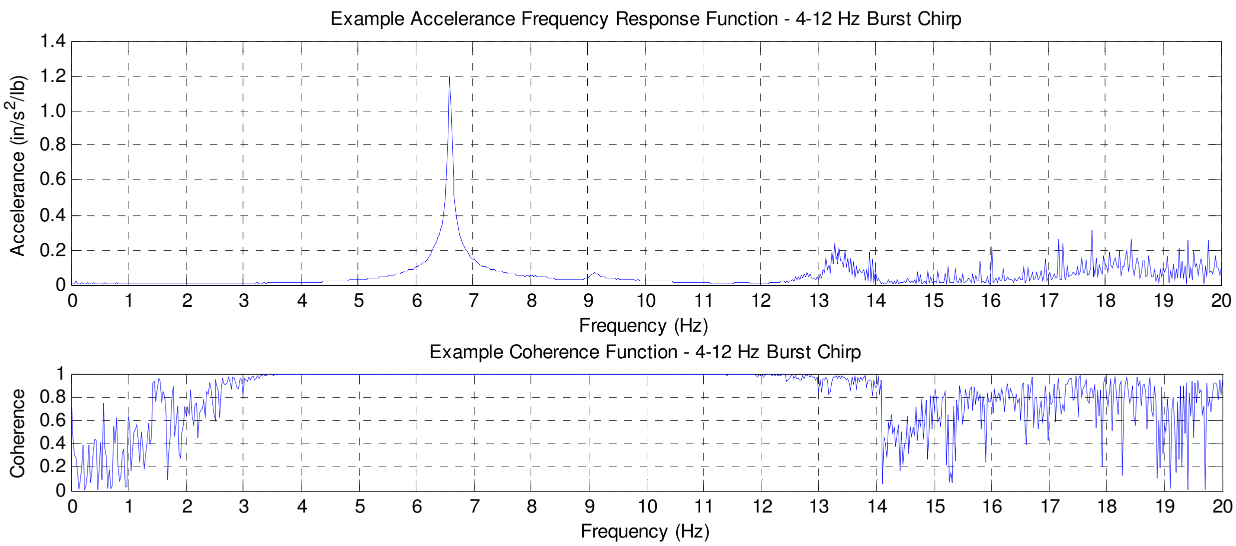


Figure 2.16: Typical 4-12 Hz Burst Chirp Magnitude and Coherence (0-20 Hz)

The accelerance FRF magnitude, phase, and coherence for the example driving point measurement within the 4-12 Hz range are shown in Figure 2.17. Again, the superior coherence achieved with only three averages of the burst chirp signal is apparent, particularly considering the vertical axis scale. For comparison of quality, the coherence from the previous continuous chirp is shown with the burst chirp at a similar scale in Figure 2.18.

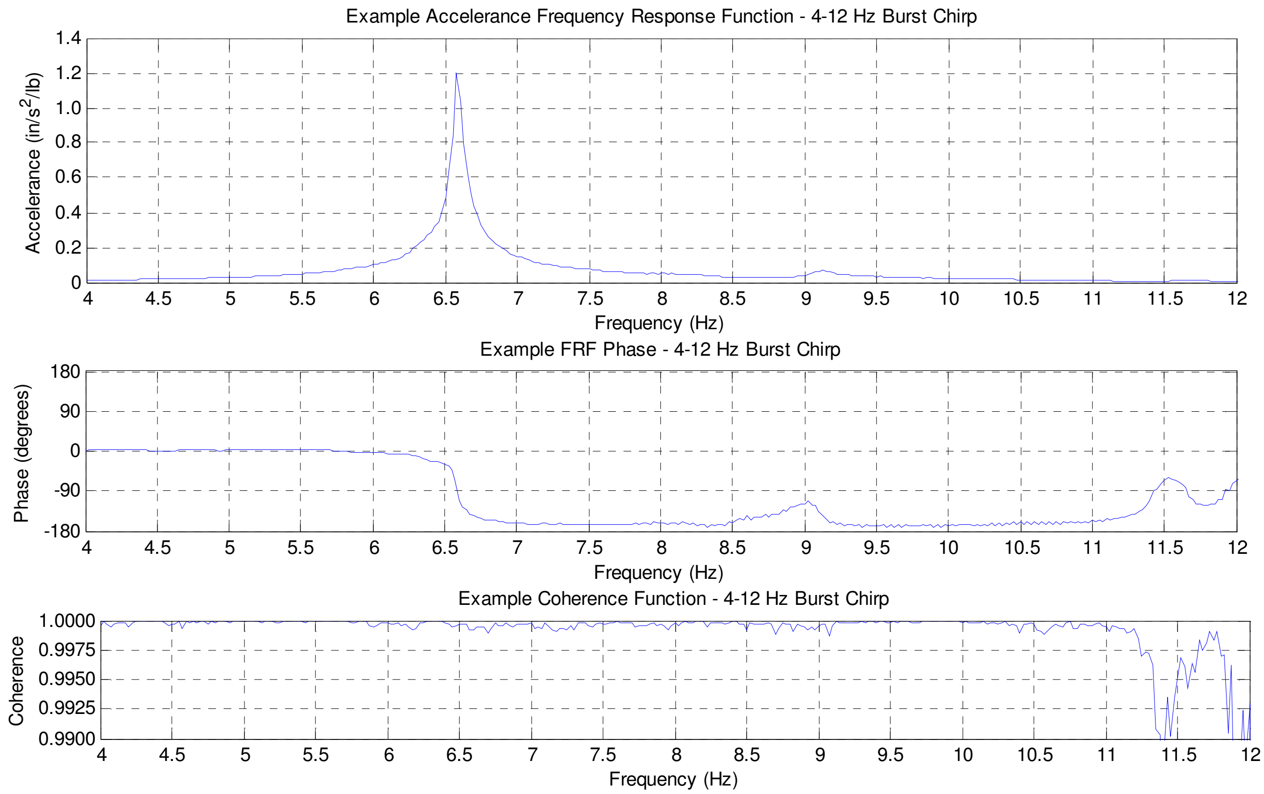


Figure 2.17: Typical 4-12 Hz Burst Chirp Magnitude, Phase, and Coherence

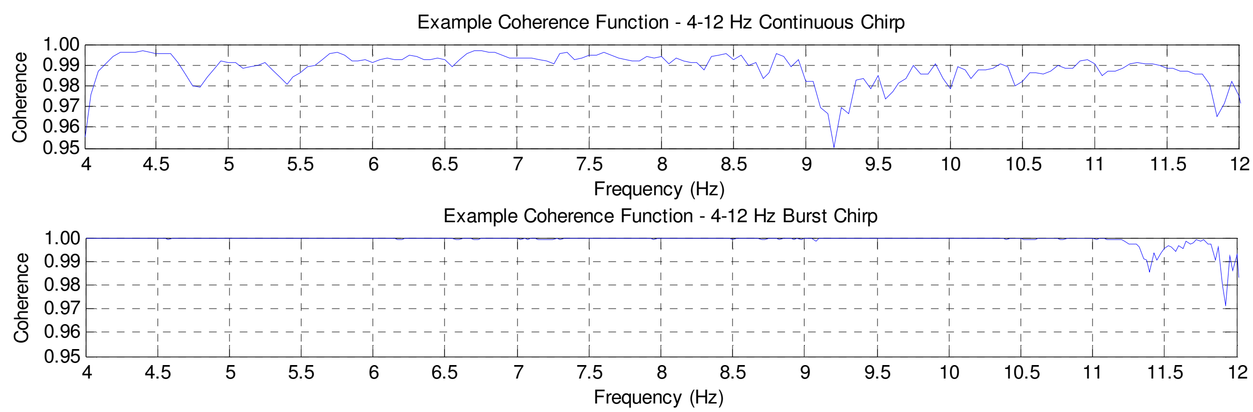


Figure 2.18: Coherence Comparison of Continuous Chirp & Burst Chirp Excitation Signals

One last item worth mentioning is the use of an accelerometer mounted to the armature of the shaker as a measure of the input force. In the autospectrum of this example's force input shown in Figure 2.14, note the dip or hitch of the measurement at 6.575 Hz, which corresponds to the resonant frequency of the tested floor for this measurement. This is also noticeable in the force plate time history of this example as a dip in input force around the 12-second point. This occurs when driving the floor because it takes very little force to achieve very large amplitude response at the resonant frequency. This dip in force is not captured when accelerance FRFs are derived using an accelerometer mounted to the armature of the shaker and not a force plate. The actual force applied to the floor is due to the *relative* acceleration between the armature and the floor, which at resonance are 90 degrees out-of-phase from one another, and not the absolute acceleration of just the armature. Because the absolute acceleration is measured on the armature, this dip in the reference signal (i.e. the armature acceleration) at resonance is not captured. For example, Figure 2.19 shows the autospectra of a measurement taken on a flexible structure (i.e. a floor) subjected to a 5-10 Hz burst chirp signal. For comparison, both the armature acceleration and the force plate voltage were measured simultaneously and used as reference channels for a driving point accelerance FRF measurement.

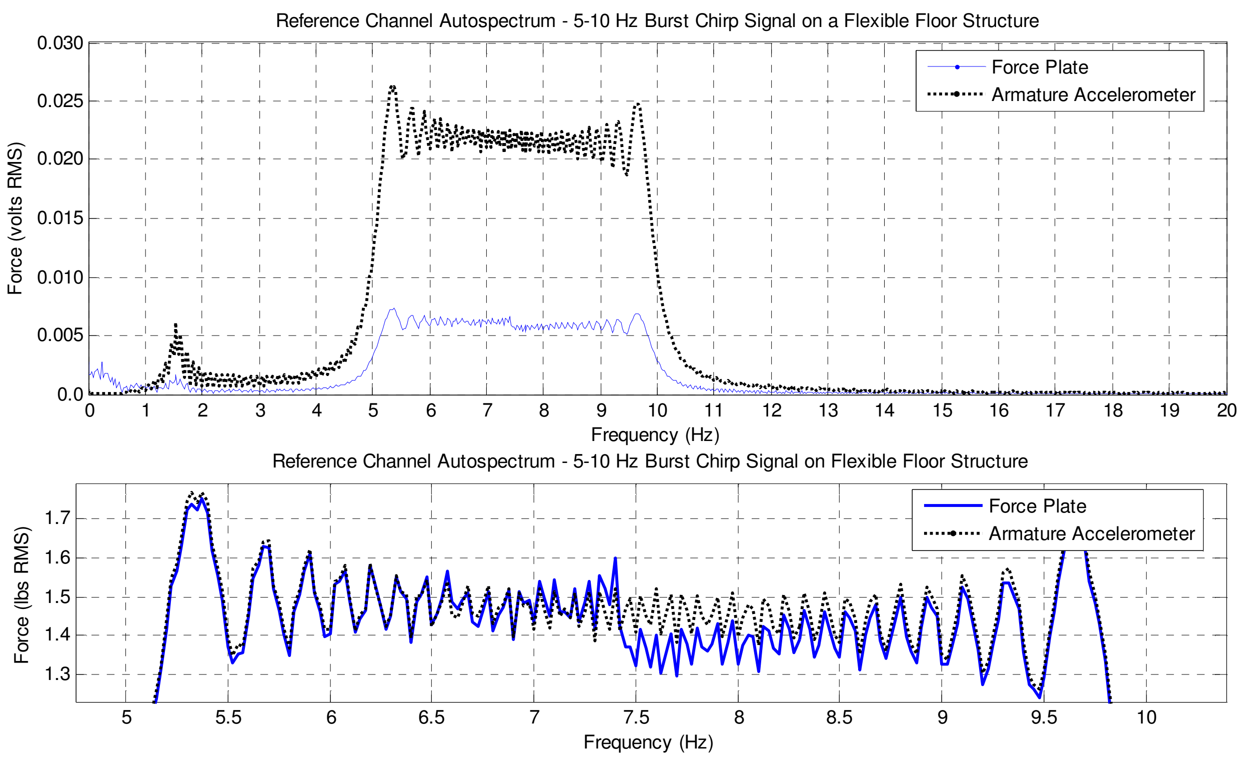


Figure 2.19: Armature Acceleration and Force Plate Autospectra for 5-10 Hz Burst Chirp

The first set of input force autospectra in Figure 2.19 is in raw voltage units. In the second set of autospectra, the two signals are converted into pound units for direct comparison by multiplying the armature acceleration by 67.4 lbs (the weight of armature) and multiplying the force plate signal by its calibration value, 240 lbs/volt. There is a noticeable deviation at and after 7.45 Hz, the resonant frequency of the tested floor in this example. Although the difference is slight in the provided example, it becomes worse as the tested floor accelerations are greater. The overestimation of force by the armature acceleration leads to an underestimation of the accelerance value. The deviations only occur where accelerations are high, such as at the resonant frequencies, but these are the most critical areas of the accelerance FRF for analysis, which is why the use of an armature accelerometer for estimating force is discouraged.

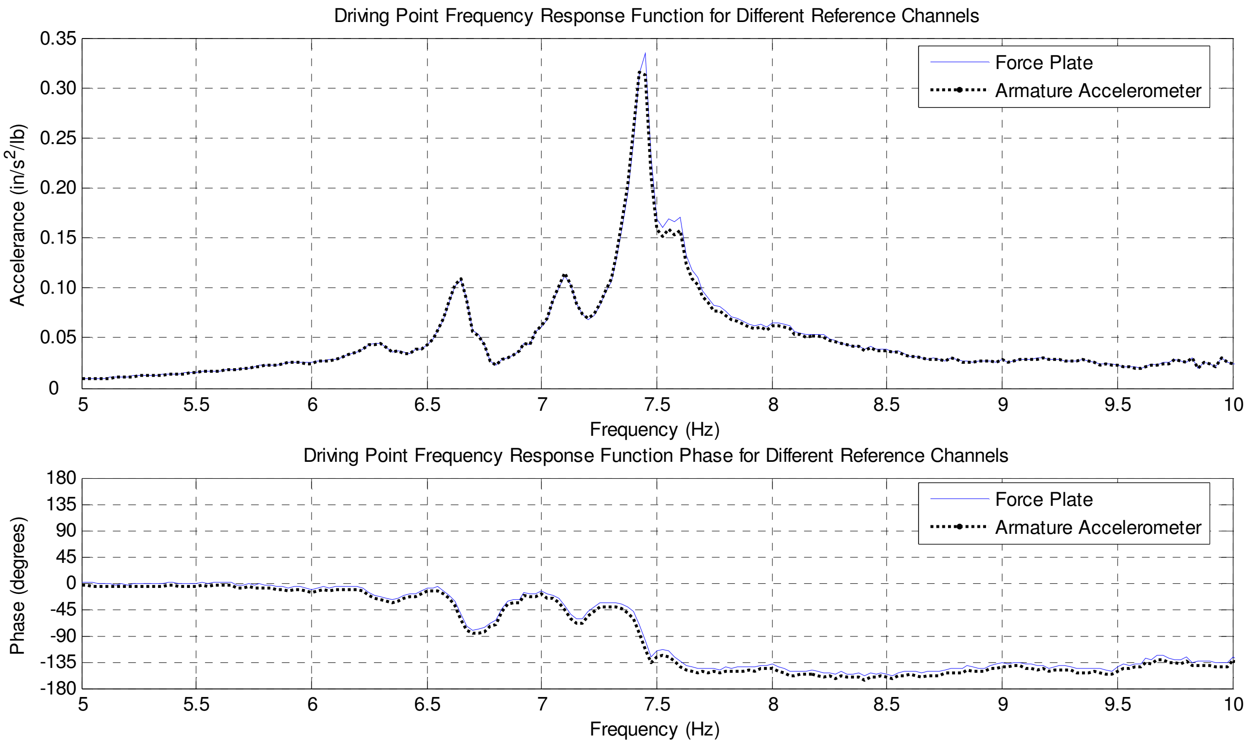


Figure 2.20: Driving Point Accelerance FRF Magnitude and Phase from Different References

The differences in measured accelerance FRF magnitude and phase are shown in Figure 2.20. While the two FRFs are nearly identical, they deviate at the most critical areas, the peaks. At the 7.45 Hz resonant frequency, the peak magnitude is underestimated by 6.6% by the armature accelerometer accelerance FRF, which may or may not be considered negligible depending on the desired accuracy and the application of the measurement. It should be noted that the provided example had relatively low peak accelerance values compared to the tested floors of the presented research, which had peak values often exceeding 1.0 in/s^2/lb, three times the magnitude of the example accelerance FRF. If at all possible, the use of a force plate is recommended as the best practice, and the use of an armature accelerometer is discouraged unless no other means for measuring the force is possible. Even then, the presented data should include a qualifying statement reflecting the possibility of underestimated accelerance FRF magnitudes around the resonant peaks.

2.4.2 Instrumented Heel Drop

Like the chirp signal, another form of broadband excitation that may be useful for efficiently deriving the accelerance FRFs during modal testing is the instrumented heel drop, where a heel drop impulse excitation is performed on a force plate. Blakeborough and Williams (2003) evaluated the use of an instrumented heel drop test for performing modal analysis on floor systems. They concluded the instrumented heel drop was a more effective modal testing technique than an instrumented impact hammer because it gave better results at lower frequencies, while sufficiently exciting the structures with frequencies in the range of 2 to 15 Hz, which is the range of interest for floor vibration problems due to walking excitation. Hanagan et al. (2003) also concluded the instrumented heel drop yielded high quality data and served as a good alternative to a shaker when cost and portability are an issue. The instrumented heel drop was not used extensively on the floors in the presented research; however this technique for testing floors is described because it could be used efficiently in the absence of a shaker device.

A heel drop is an impact force caused by a person assuming a natural stance, maintaining straight knees, shifting their weight to the balls of the feet, rising approximately 2.5 in. on their toes, and then suddenly relaxing to allow their full weight to freefall and strike the floor with their heels. A demonstration of this technique on a force plate is illustrated in Figure 2.21.

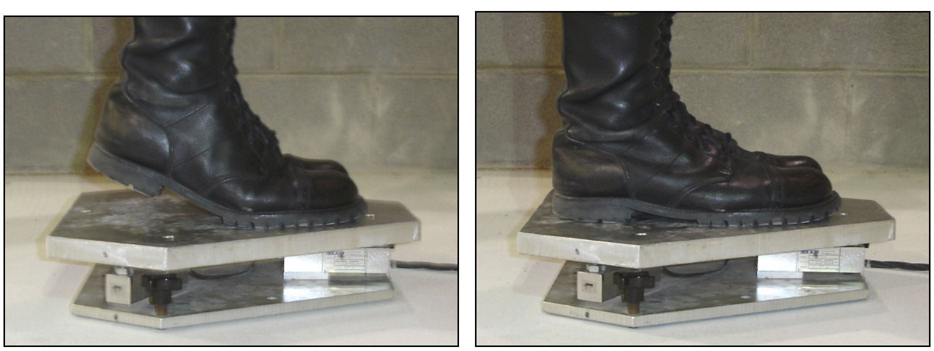


Figure 2.21: Instrumented Heel Drop

An obvious advantage of this type of impulse excitation is how easy it is to perform and that it does not require any equipment other than what is needed to measure the response. Figure 2.22 includes the time histories and autospectra for instrumented heel drops conducted on a rigid surface (slab-on-grade) and a flexible floor structure. The measurements were taken using three heel drop averages from a 245-lb individual wearing rubber heeled boots (Figure 2.21). It is difficult to achieve exactly the same heel drop multiple times, even when they are conducted consecutively by the same individual; thus averages were taken, which had the effect of smoothing out the autospectra to the form shown in Figure 2.22.

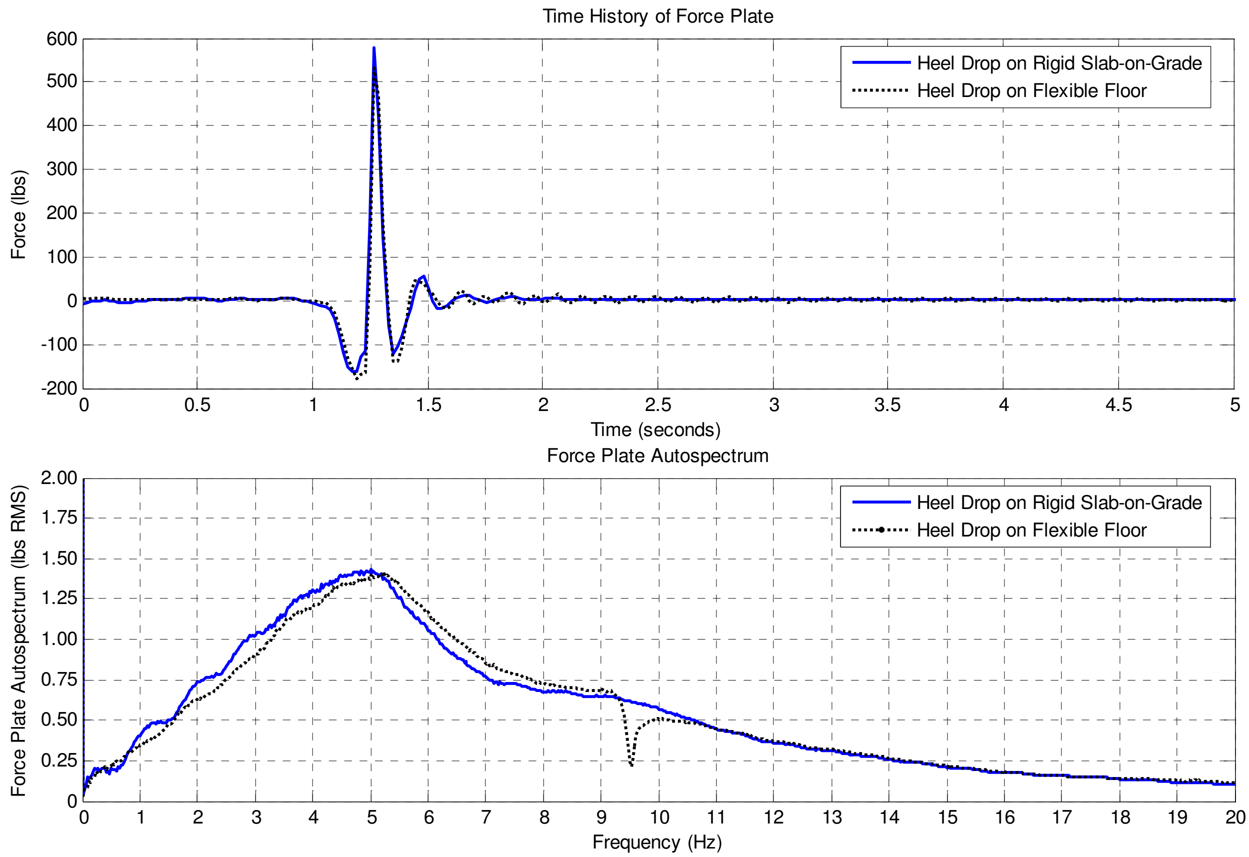


Figure 2.22: Instrumented Heel Drop Time Histories and Autospectra

The impulse loads of the time histories in Figure 2.22 line up because the same trigger settings were used for both the rigid and flexible surface measurements (trigger on input of the force plate with a -2.5% delay). Note that both the time histories and the autospectra are very similar, with the exception of a dip in the input force autospectrum for the flexible floor measurement at 9.55 Hz. Again, like the autospectrum of the force input for a burst chirp measurement, this force drop off is typical at the resonant frequencies of the structure.

Figure 2.23 is a comparison of the force input autospectra for instrumented heel drop excitation and the example 4-12 Hz burst chirp signal discussed in the previous section, which was a different flexible floor system but is still valid for general comparison (Figure 2.14, dominant frequency at 6.575 Hz). One advantage of the instrumented heel drop over the shaker is the ability to excite the lower frequencies that may not be reachable with a shaker due to its internal dynamics. Although concentrated more in the 2-7 Hz range, the instrumented heel drop does seem to provide a reasonable amount of energy over the general frequency range of interest for floor vibration serviceability.

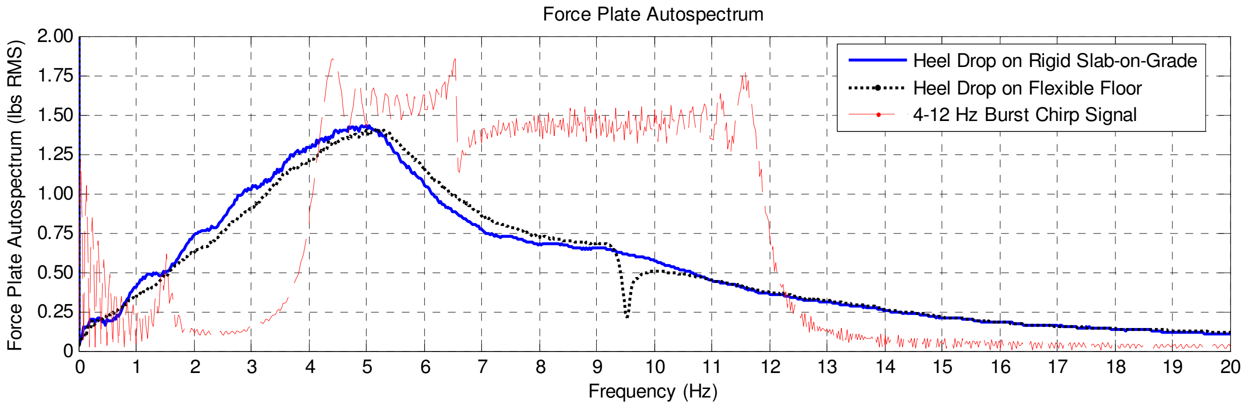


Figure 2.23: Comparison of Chirp & Instrumented Heel Drop Force Input Autospectra

As mentioned previously by Hanagan et al. (2003) and Blakeborough and Williams (2003), the instrumented heel drop could serve as an inexpensive and portable method for conducting rough modal testing, however it should be noted that the quality, consistency, and control offered by modal testing using an electrodynamic shaker cannot be replaced by the instrumented heel drop. Unlike the instrumented heel drop, the electrodynamic shaker has the ability to input concentrated excitation at resonance. It is also worth mentioning that in practice the instrumented heel drop requires the individual to remain on the force plate, likely located in the middle of a bay, for the duration of the measurement. The presence of the individual on the floor will add some amount of damping to the system, particularly at a location of large response. For floor structures with very low levels of damping to begin with, the effects on damping by the individual may be considerable, although this was not studied in this research effort.

2.4.3 Driving Point Accelerance Frequency Response Functions

A driving point measurement is taken by an accelerometer located next to the force plate at the point of excitation and is the key measurement for defining mode shapes. Whether measuring a row or a column of the accelerance FRF matrix of Equation (2.3), this measurement represents a diagonal term, where the input force location, k, is the same as the acceleration response location, i, and thus $A_{ik}(\omega) = A_{ii}(\omega) = A_{kk}(\omega)$. More importantly, it is the datum measurement used by modal analysis software from which responses at all other locations on the floor are scaled. For testing floor systems, the driving point measurement has additional significance. Because most of the current design guidance for floor vibration serviceability is based on the response at the middle of a bay, this is typically where the shaker is located during modal testing for a direct comparison of experimental response to the response predicted by other hand calculation methods. Typically, this mid-bay location also corresponds with the anti-node of response over a tested area, or the point with the greatest level of response. This makes the driving point measurement ideal for parameter estimation because it will typically have the highest signal-to-noise ratio, the best coherence, and the most consistency between repeated measurements of any location on the floor.

By looking at the accelerance FRFs for the driving point measurements of an excited bay, the "dominant frequency" can be visually identified as the frequency with the largest magnitude peak. Most driving point FRFs for the numerous bays tested had several peaks, usually closely spaced with some larger than others. Therefore, "dominant frequency" is used rather than "fundamental frequency" because this term is typically reserved to describe the lowest frequency of a structure. The intent of the investigation was not to determine the lowest frequency of the floor, but the frequency of excitation that would create the greatest response at the center of a bay within a certain bandwidth. This is a logical approach because it is the center of the bay that will experience the greatest response due to one of the harmonics of walking exciting the dominant frequency.

Figure 2.24 shows 35 driving point accelerance measurements overlaid on one another. These 35 measurements were taken over roughly a three hour period during a modal test of a floor system. An accelerometer was left with the shaker throughout testing to record the driving point accelerance FRFs shown as well as the accelerance FRFs for the other locations of the floor where the roving accelerometers were placed (not shown). While there are a couple of things to note, the first is the consistency and quality of the measurements, even though 35 traces are plotted together. Secondly, each peak on the accelerance FRF indicates the presence of at least one mode, of which there are several, but clearly the dominant frequency is 7.20 Hz.

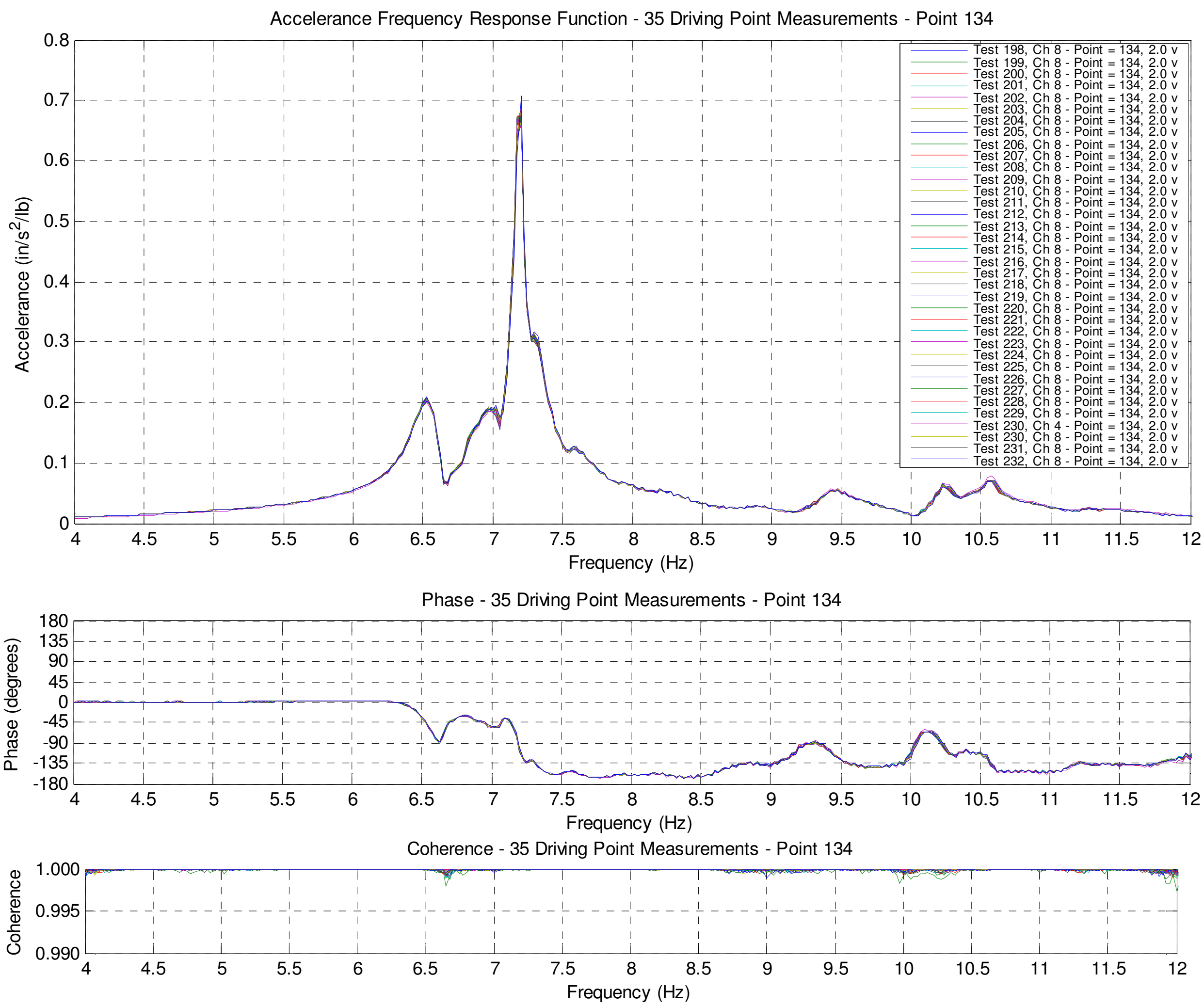


Figure 2.24: 35 Driving Point Accelerance FRFs (Magnitude, Phase, and Coherence)

Most modal analysis software only needs one driving point measurement for scaling mode shapes and performing parameter estimation; however there is a practical reason for always reserving one channel of a multi-channel spectrum analyzer to record the driving point measurement, which is to build an additional level of quality assurance into the modal test. Having an accelerometer with the shaker for every measurement to record the driving point accelerance is a way to identify equipment malfunctions, monitor any change in the floors response over the long periods of time that it may take to sweep the modal coverage area, and simply for redundancy of this key measurement. Although the driving point measurements of Figure 2.24 are very consistent, this may not always be the case, particularly if the sweep over the modal test area is large and takes a long period of time (possibly over multiple days of testing). The consistency of a driving point measurement is an assumption, and given the means of a free channel on the multi-channel DSP unit, this assumption should constantly be validated. If the driving point measurements do change over time, having the redundant measurements provides an opportunity to identify how they have changed, and possible ways to adjust for it.

2.4.4 Sinusoidal Driving Functions

In the interest of time, certain assumptions must be made by the experimentalist; however it is also the responsibility of the experimentalist to validate those assumptions. Forming the accelerance FRF over a specific frequency range by using a burst chirp excitation is an assumption that claims each FRF spectral value computed has the same magnitude and phase as a measurement where the structure is driven sinusoidally at that spectral frequency and allowed to achieve steady state conditions. Fortunately, with an electrodynamic shaker, this assumption is easily validated. This manual verification of the accelerance measurements derived from the burst chirp signal is a good technique to ensure the quality and consistency of the modal test, and to build confidence that the measured response of the structure corresponds to measured forces, an important part of validating the computed response from FE models.

In a steady state sinusoidal excitation, there is virtually no input force or response at any other frequency besides the driving frequency. Consequently, the accelerance FRF computed for a steady state sinusoidal measurement is completely invalid except for the single spectral value at the driving frequency because the response and force computations at other frequencies are based purely on measured noise. The magnitude of the accelerance for a driving frequency f can be computed by dividing the amplitude of the acceleration response by the amplitude of the input force as shown in Equation (2.11), which is just a single driving frequency representation of the accelerance between two points previously expressed by Equation (2.3).

$$A(\omega) = A(f) = \frac{a_{peak}(f)}{F_{peak}(f)} \quad \text{or} \quad \frac{a_{rms}(f)}{F_{rms}(f)} \qquad (2.11)$$

The second expression in Equation (2.11) is valid because the relationship between peak and root-mean-square (RMS) values for a sinusoidal function is simply the constant $\sqrt{2}$. Equations (2.12) are a short derivation of this relationship in terms of $a(t)$, a steady state sinusoidal acceleration function with amplitude a_{peak}:

$$a(t) = a_{peak} \sin \omega t \qquad \text{and} \qquad T = \frac{2\pi}{\omega}$$

$$a_{rms} = \sqrt{\frac{1}{T}\int_0^T a(t)^2 dt} = \frac{a_{peak}}{\sqrt{2}} \qquad \text{and} \qquad a_{peak} = a_{rms}\sqrt{2} \tag{2.12}$$

This relationship is convenient because the autospectrum values of input force and acceleration response are usually expressed in terms of RMS voltage by the spectrum analyzers and can easily be extracted from the data at a specified frequency for the spectral value computation using Equation (2.11).

Using the 7.20 Hz dominant frequency identified in the previous example of driving point accelerance FRFs (Figure 2.24), an example steady state sinusoidal measurement is shown in Figures 2.25 and 2.26. For this example measurement, the floor was excited with a 7.20 Hz sinusoidal forcing function with a magnitude of approximately 20 lbs. The resulting driving point acceleration response had a magnitude of approximately 0.035g (3.5%g). Note there are two channels measuring the driving point response at Point 134 (Channels 4 and 8) for this example measurement, and the other accelerometers are located at the middle of other bays on the floor, which is why some are in-phase and others are out-of-phase with the driving point response. The autospectra for both the input force and acceleration response are also included in the figures.

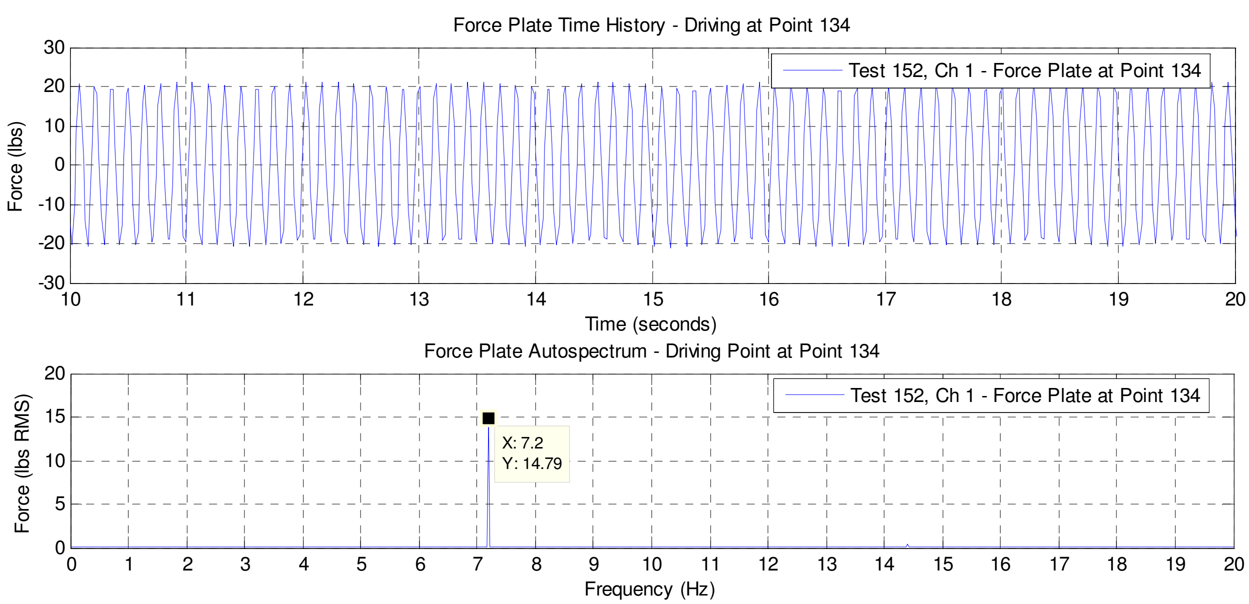


Figure 2.25: Example Sinusoidal Excitation Force Time History & Autospectrum

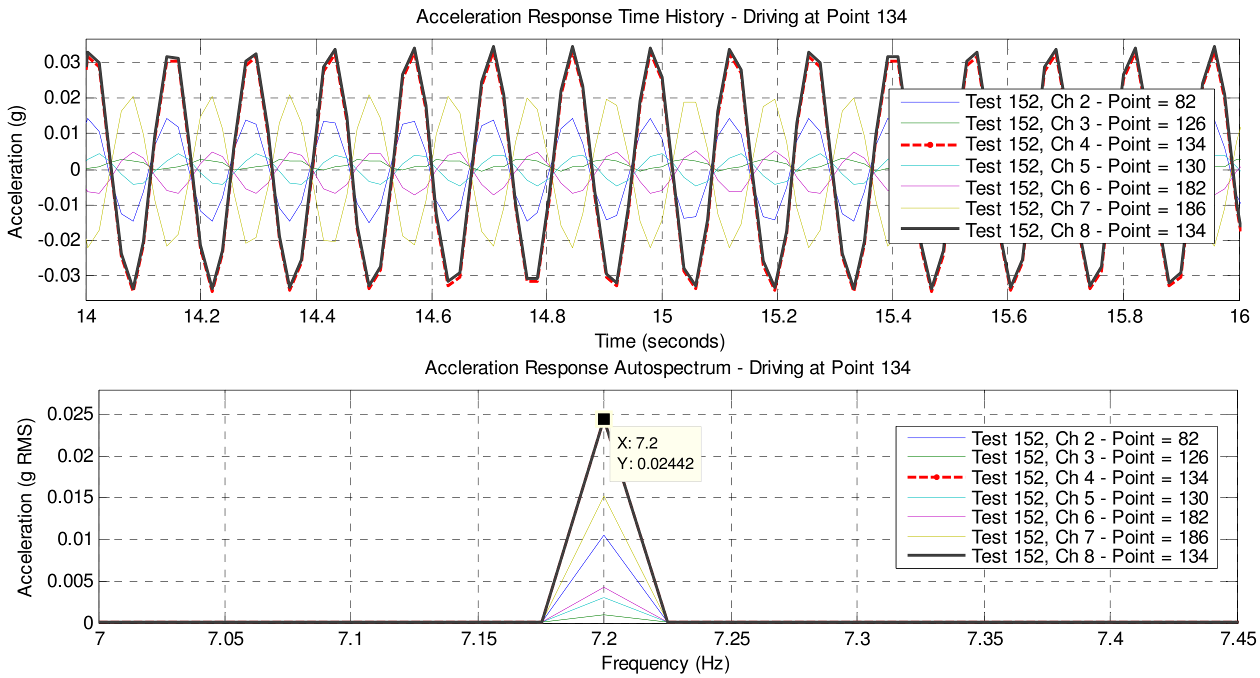


Figure 2.26: Example Sinusoidal Acceleration Response Time History & Autospectrum

The RMS acceleration and force values at the 7.20 Hz driving frequency are highlighted in the autospectra of Figure 2.25 and Figure 2.26. Equation (2.11) can be applied to compute an accelerance of 0.6375 in/s^2/lb of input force, that is

$$A(f = 7.20Hz) = \frac{a_{rms}(f)}{F_{rms}(f)} = \frac{(0.02442 \text{ g RMS}) * (386\ in/s^2/g)}{(14.7871 \text{ lbs RMS})} = 0.6375 \ {}^{(in/s^2)}\!\big/_{lb}$$

The average peak magnitude of the 35 driving point accelerance FRFs in Figure 2.24 was 0.67 in/s^2/lb at 7.20 Hz, which is in excellent agreement with the manually derived value of 0.64 in/s^2/lb (< 5% difference). Using the RMS values from the autospectra rather than estimated peak values from the time history functions gives a better representation of the accelerance for several reasons. First, there may be some DC offset in the accelerometer or force plate signals, thus requiring an average of the positive and negative peaks to account for the offset. The computation of the autospectrum automatically removes the average of the signal, which for a sinusoidal function is either zero or the amount of the DC offset. Secondly, the time history shown is the last of the measured time block(s), whereas the autospectrum reflects the updated average of all time blocks included in the measurement.

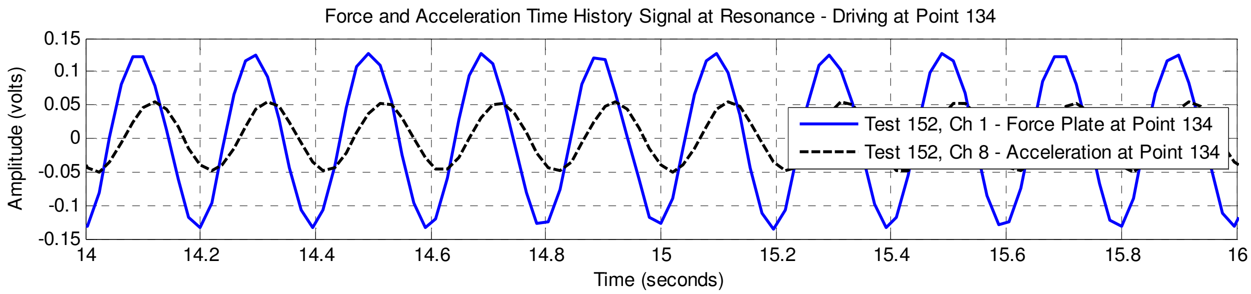


Figure 2.27: Comparison of Force and Acceleration Time Histories

Because the autospectra do not carry any phase information, a different approach is needed if there is an interest in the phase between the excitation and response signals for a sinusoidal measurement. For the example provided, the difference in phase is quite noticeable (especially because it is being driven at resonance) as shown in Figure 2.27.

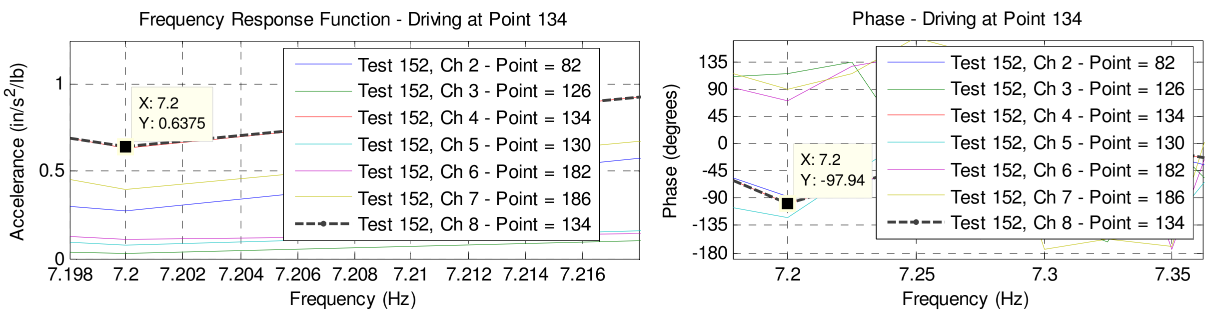


Figure 2.28: Accelerance FRF Magnitude and Phase for a Sinusoidal Measurement

One method to estimate the phase would be the visual evaluation of the phase shift of the two signals; however there is a much easier and more accurate way to measure this quantity. If one or multiple averages are taken of the sinusoidal measurement, the computation of Equation (2.11) is automatically performed by the spectrum analyzer and given as the magnitude of the accelerance FRF at that frequency. Additionally, the phase is computed in this manner. If more than one average is used, then a coherence function is computed, providing a tangible representation of the measurement quality. Figure 2.28 shows the computed accelerance FRF magnitude and phase values for the provided example, demonstrating that by taking an average during a steady state sinusoidal measurement, the hand calculations for accelerance and phase can be avoided by relying on the computation by the spectrum analyzer. Additionally, because more than one average was taken in the example, a coherence of 0.99995 was computed at the 7.20 Hz driving frequency, providing additional confidence in the measurement quality.

The most useful application of the sinusoidal measurement is not simply to verify the accelerance values computed by an accelerance FRF derived from burst chirp excitation, but the verification of the location of the peak frequency. As a manual check to ensure the dominant frequency (i.e. the frequency of peak accelerance) was correctly located using burst chirp excitation, the shaker can be run sinusoidally at frequency increments above and below the identified dominant frequency. This must be accomplished only at frequency increments that correspond to computed spectral lines, which are based on the chosen frequency resolution, otherwise the estimated values will be invalid (underestimated, actually). In some instances, the actual peak accelerance from steady state sinusoidal excitation may be at a frequency that is one or two spectral lines different from the one identified from the chirp derived accelerance FRF (0.025 Hz to 0.05 Hz for the tested floors). Although not strictly required for every modal test, this is another example of checking the assumptions often taken for granted during modal testing so that the experimentalist has confidence in the quality of the modal measurements.

2.4.5 Unreferenced Measurements

The easiest dynamic measurement to take when testing a floor system is the unreferenced response-only measurement because it requires the least amount of equipment and technique. Handheld single-channel spectrum analyzers are much less expensive and more portable than their multi-channel counterparts that have the ability to conduct the referenced modal testing measurements previously discussed. Examining the acceleration levels from various types of unreferenced excitation can indicate a lot about the behavior of the floor under service conditions. As previously shown in the autospectrum of a heel drop in Figure 2.22, despite whether the input force of the heel drop is measured or not, it can still provide adequate excitation to a floor system within the frequency range of interest for floor vibration serviceability. Thus, analysis of the frequency content of the response from an unmeasured heel drop excitation can give insight into the active frequencies of the floor. Arguably, the only drawback to unreferenced measurements is when too much is interpreted from the limited information they provide. The only information a single-channel unreferenced measurement can provide is an accurate measure of the acceleration over a period of time, and the frequency content of that acceleration time history. Given certain types of excitation, dynamic properties of the floor can be estimated to a limited degree from the unreferenced measurement; however it should be understood that only modal testing provides the accuracy typically needed for any detailed characterization of the behavior of a floor system. As long as this limitation is understood, then the unreferenced measurement is an effective tool that has its place in dynamic testing of floor systems.

Unreferenced heel drop measurements are most effective for indicating the lower frequency content of the floor system. Previously, Figure 2.23 showed the autospectrum of an instrumented heel drop compared to the autospectrum of a chirp signal. The heel drop has a noticeable advantage at lower frequencies because the shaker is limited by its internal dynamics and is limited in its ability to excite these lower frequencies. This is why it is good practice to conduct several heel drop measurements prior to the start of modal testing to identify the active frequencies of the floor system, which are crucial for choosing the parameters for the chirp signal. Figure 2.29 shows an example acceleration time history for a heel drop measurement using a single-channel Ono Sokki CF-1200 Handheld FFT Analyzer.

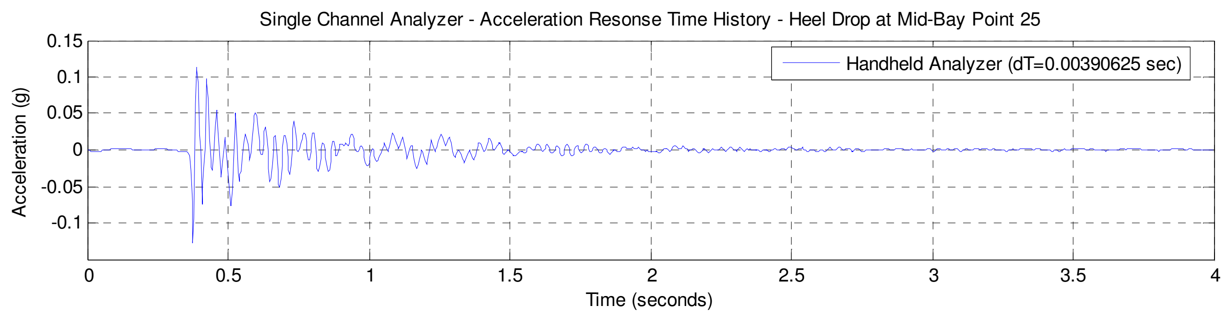


Figure 2.29: Sample Acceleration Time History to Unreferenced Heel Drop Excitation

Note that the unreferenced time history measurement has excellent time domain resolution, which is a product of 1024 samples taken over the four second time record. Unfortunately, a byproduct of the shorter time history is a loss in frequency domain resolution, which for measurements using the Handheld FFT Analyzer was 0.25 Hz (+/- 0.125 Hz) as shown in Figure 2.30. This frequency domain resolution is in stark contrast with the autospectrum of the response at the same location on the floor due to a 4-12 Hz chirp signal (Figure 2.30). The chirp signal autospectrum was taken using the multi-channel SigLab spectrum analyzer set at a frequency domain resolution of 0.05 Hz.

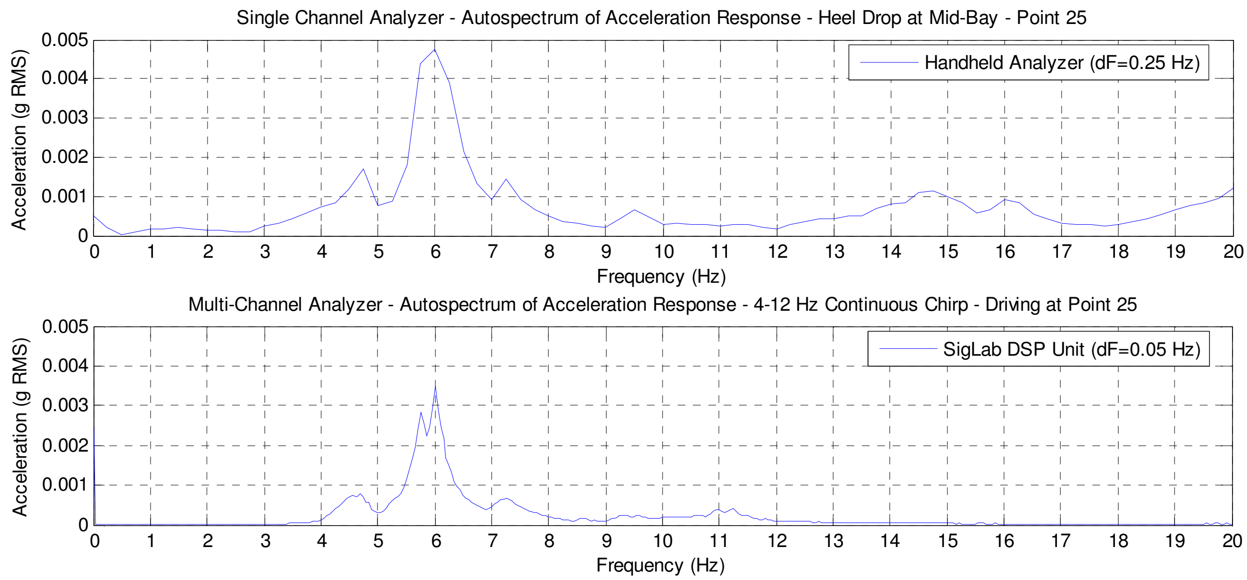


Figure 2.30: Autospectra of Unreferenced Heel Drop and Chirp Measurements

The first three significant frequencies of the heel drop autospectrum of Figure 2.30 are 4.75 Hz, 6.00 Hz, and 7.25 Hz. The respective peaks in the autospectrum of the chirp signal (with much finer frequency domain resolution), are 4.70 Hz, 6.00 Hz, and 7.25 Hz; however the peak at 5.75 Hz is completely missed by the poor resolution of the heel drop autospectrum. If only the unreferenced autospectra were available, the wide peak at 6.00 Hz may be misunderstood as a large amount of damping at this frequency, rather than the presence of the two closely spaced peaks shown in chirp autospectrum. It should be stressed that this was not a weakness of the unreferenced measurement, but a weakness of the equipment that could not provide adequate frequency domain resolution for low frequencies and low levels of damping. It is also worth mentioning that for all but the strongest of frequencies, the peaks of the autospectrum may not be the exact peak frequencies of the accelerance FRF, should it be measured. The magnitude of the accelerance FRF can be interpreted as the autospectrum of the response divided by the autospectrum of the force. Unless the autospectrum of the force is a constant value across all frequencies, which it is not even for a chirp signal, the magnitude of the peaks in the accelerance FRF may not be the same as those in the response autospectrum. Although not shown, the peaks of the accelerance FRF computed for the chirp signal shown in Figure 2.30 are 4.75 Hz, 5.80 Hz, 6.00 Hz, and 7.25 Hz, two of which differ from the autospectrum by one spectral line The difference may be slight in this instance, but the nuance should not be overlooked when interpreting data from unreferenced measurements. It should also be noted that for heel drop measurements, the shape of the autospectrum is definitely not constant across the frequency range, as shown in Figure 2.23. Because of this, extra caution should be exercised when attempting any comment on the relative strengths of modes from the relative size of the peaks of the autospectrum. These discrepancies highlight the need for using a referenced modal testing setup for floor systems if it is desired to accurately characterize their dynamic behavior.

Several different types of unreferenced measurements are available. Some examples include measurements from ambient conditions and various human excitations, including heel drops, walking parallel and perpendicular to the floor framing, and bouncing at harmonics of the bay's dominant frequency. One technique is to attempt to excite the floor by bouncing at a harmonic of the resonant frequency in an attempt to achieve steady-state resonant response. The bouncing excitation can then be suddenly stopped and the resulting response decay recorded for the purpose of estimating damping in the system. While this method provides a crude estimate of damping, it is typically higher than actual values due to the damping that the human body introduces into the system. Additionally, closely spaced modes can create a beating effect in the decay that makes the estimate difficult. More accurate methods exist for estimating damping of the system, such as the half-power bandwidth method using the accelerance FRF, or multi-degree-of-freedom curve fitting methods of a set of accelerance FRFs, which are discussed later.

As previously mentioned, unreferenced measurements have their advantages, but high quality referenced modal data is a better indicator of the dynamic behavior of a structure, particularly if it is to be used to validate FE models. Much of the previous research and FE modeling of steel composite floors have been based on frequency content of unreferenced measurements. As demonstrated in the unreferenced heel drop example, the frequency content and/or damping of a system can be misinterpreted or overlooked by poor frequency domain resolution or other misguided assumptions about the autospectrum of response.

2.4.6 Damping Estimates

Although it is commonly known that the actual mechanism of energy dissipation of an in-situ floor system (or any large structure for that matter) is much more complicated than simple viscous damping used for mathematical convenience, the equivalent modal viscous damping ratio, β_r, is still the most common value used for expressing the damping behavior of the floor.

Unlike mass and stiffness, damping cannot be determined by the materials and layout of the structure other than in the most general estimated terms (Clough and Penzien 1993). It is a value that must be estimated from measurements, and several methods are available for this process. Some of these methods (as used in the presented research) are presented in this section along with their advantages and disadvantages.

The half-power method, also known as the bandwidth method or half-power bandwidth method, is the most straightforward of the methods for estimating damping and is based on the properties of the frequency response function of a system at resonance. The half-power bandwidth of a resonant peak is defined as the width of the frequency response function at $1/\sqrt{2}$ of the peak amplitude at resonant frequency f_r. Equation (2.13) shows the computation of the damping ratio, where f_a and f_b are the frequencies on either side of the resonant frequency that mark the half power width.

$$\beta_r = \frac{f_b - f_a}{2f_r} \tag{2.13}$$

A derivation of Equation (2.13) can be found in the text by Chopra (2001). The advantage of this method is the ability to compute damping directly from the peaks of the measured accelerance FRFs. This simple computation can be done in the field or in post processing. The driving point measurements are typically the best measurements to use for the computation, as the peaks of the accelerance FRFs are generally much more defined than measurements at other locations of the floor. Theoretically, the estimated damping should be the same regardless of the chosen measurement; however the peaks in measurements that are far away from the driving point can sometimes be poorly defined if at a boundary or other location not significantly excited by the frequency. As a cautionary note, it was previously discussed that the autospectrum of response can share a similar shape as the accelerance FRF, but it is not the same. The half-power method is derived from a forced-vibration frequency response curve, and not the autospectrum, which is simply a response-only measurement. Although the computed damping values may seem reasonable, the half-power method is not suitable for unreferenced response autospectra. A potential disadvantage of the half-power method is that the frequency response must be accurate at and around the resonant peak (Clough and Penzien 1993). At the low levels of damping typically found in bare floor systems (< 2%), the peaks of the accelerance FRFs are quite sharp. Not only does this highlight the need for good quality control of the accelerance measurements, it also demonstrates how the choice of frequency domain resolution can affect the ability to adequately estimate the damping of the system using the half-power method. Another disadvantage of the method, as with all the methods for estimating damping, is the difficulty of properly estimating damping in system with closely spaced modes (peaks). If two peaks of the accelerance FRF are close, the half-power estimate may be skewed by the presence of the other peak, which has the effect of widening the accelerance FRF and increasing the computed estimate of damping.

To verify the measured accelerance FRFs adequately capture the resonant peak(s), it may be worthwhile to perform sinusoidal measurements to manually compute the accelerance values at frequencies around the peak. If steady state sinusoidal measurements are taken at enough frequency increments on either side of the resonant frequency (i.e. out to frequencies with accelerance values less than $1/\sqrt{2}$ of the peak accelerance), the accelerance FRF may be defined enough to perform a half-power damping calculation on the manually derived values. The advantage of this stepped sine sweep approach is that it provides an additional level of verification of both the accelerance FRF values and the estimated damping. The disadvantage of this approach is the time required to perform the measurements, and if it is determined that the accelerance FRF measurements from chirp excitation are of a high enough quality, it may be unnecessary or redundant.

Driving the floor sinusoidally is also useful as an initial part of capturing the time history of the response decay, another limited method for estimating damping. While the time history of decay of the response from any type of excitation demonstrates the floor's ability to dissipate energy, impact excitation using a heel drop or instrumented impact hammer tends to excite a wide range of frequencies, thus the response decay to these types of excitations may include response from several excited modes. If there is only one very dominant mode of vibration, however, the resulting decay trace may be very smooth. Unfortunately, there is typically more than one participating mode for this type of excitation on in-situ floor systems. Additionally, an individual performing the heel drops adds damping to the system, as does an individual with instrumented impact hammer. With closely spaced modes, or several strong modes, there will not be a clean decay trace if multiple modes are participating in the response. One method of attempting to minimize the effect of other modes is to drive the floor sinusoidally at the resonant frequency of the mode of interest, and then shut off the sinusoidal excitation to record the time history of response decay. To estimate the damping of the mode, the resulting decay trace can be analyzed by either the logarithmic decrement method or by curve fitting the decay with an analytical expression. The equation for the logarithmic decrement is not shown here, but it is a method that uses the magnitude of the response peaks over several decay cycles to estimate the damping in the system. A brief derivation of the analytical expression of decay that is used to curve fit the response is:

The displacement free decay of a SDOF oscillator in terms of intial conditions is:

$$u(t) = e^{-\beta\omega_n t}\left[u(0)\cos(\omega_d t+\phi)+\left[\frac{\dot{u}(0)+u(0)\beta\omega_n}{\omega_d}\right]\sin(\omega_d t+\phi)\right]$$

where

$\omega_d = \omega_n\sqrt{1-\beta^2}$ = the damped natural frequency

For $u(0)=0$ (i.e. initial velocity only): (2.14)

$$u(t) = e^{-\beta\omega_n t}\left[\left[\frac{\dot{u}(0)}{\omega_d}\right]\sin(\omega_d t+\phi)\right]$$

$\therefore$ The acceleration free decay of a SDOF oscillator from an initial velocity is:

$$\ddot{u}(t) = \left[\frac{\dot{u}(0)}{\omega_d}\right]e^{-\beta\omega_n t}\left[(\beta^2\omega_n^2-\omega_d^2)\sin(\omega_d t+\phi)+(2\beta\omega_n\omega_d)\cos(\omega_d t+\phi)\right]$$

The natural frequency, ω_n, should equal to (or be very near) the driving frequency prior to decay. The initial velocity term, $\dot{u}(0)$, is arbitrary and just scales the magnitude of the decay curve. The only two terms in the expression that define the curve fit decay are the natural frequency ω_n and the damping ratio β (the damped natural frequency is defined using those terms). Thus, the analytical decay trace can be defined exclusively by these two estimated values and shifted along the time axis until it visually overlays the measured response decay. It is worth mentioning that the logarithmic decrement method is frequency independent, whereas the analytical expression in (2.14) accounts for a decay frequency that may be slightly different than the driving frequency. This may occur because the applied frequency is usually at a spectral frequency increment and the actual frequency of the floor may fall between spectral lines.

The theory of why the decay from resonance should work is based on the orthogonality of mode shapes and the principle that if a structure is initially displaced in the exact shape of a mode shape, then the resulting free decay response will only be in that mode of vibration. Because it is impossible to place an in-situ floor in an initial displaced shape by static methods, driving the floor at resonance theoretically produces a standing wave of the floor in the shape of that mode of vibration and stopping the excitation should result in free decay in that mode. Theoretically, this is a great method for estimating the damping of a mode. In practice, however, there are a lot of assumptions in the preceding statement and *this method of estimating damping only has limited application and is likely not as efficient as other methods*. Ensuring that the exact resonant frequency is achieved is difficult, thus it is assumed that the floor is being driven only very near resonance. More importantly, shutting off the excitation at exactly the right instant so that the floor does not undergo a step function is not possible. As a result of the step function, however small it may be, the floor is excited in other frequencies than just the targeted resonant frequency. The resulting response decay trace may not be a single mode function and the resulting damping estimates of the analytical decay expression (or the logarithmic decrement method) may be skewed. These same limitations apply when the response decay is achieved from an impulse load like a heel drop, because the impulse load excites multiple frequencies.

To demonstrate the limitations of this method, a few examples are presented of actual measured accelerance FRFs for an in-situ floor system and the corresponding decay analysis that was performed as described above. The first example is where the decay curve fit is an effective method for estimating damping, at least for modes that are clearly dominant and separated from other modes. An accelerance FRF for a driving point measurement is presented in Figure 2.31.

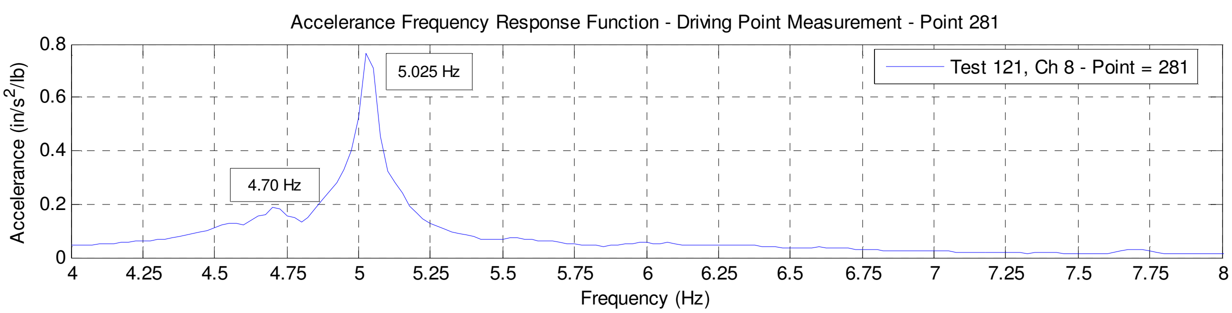


Figure 2.31: Accelerance FRF – Driving at Point 281

The dominant frequency is clearly 5.025 Hz, and the floor was excited to steady state conditions at this frequency before the shaker was shut off to allow free decay. The resulting decay trace is shown in Figure 2.32 and the corresponding analytical decay curve fit is shown in Figure 2.33 for the entire time record as well as a representative time segment. The curve fit estimated a damping ratio of 0.70%, which was in excellent agreement with the corresponding half-power method estimate of 0.65% from the 5.025 Hz peak in Figure 2.31.

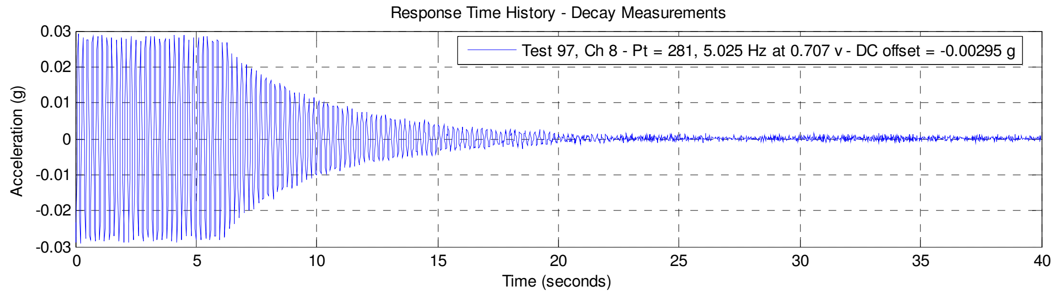


Figure 2.32: Point 281 – Response Decay from 5.025 Hz

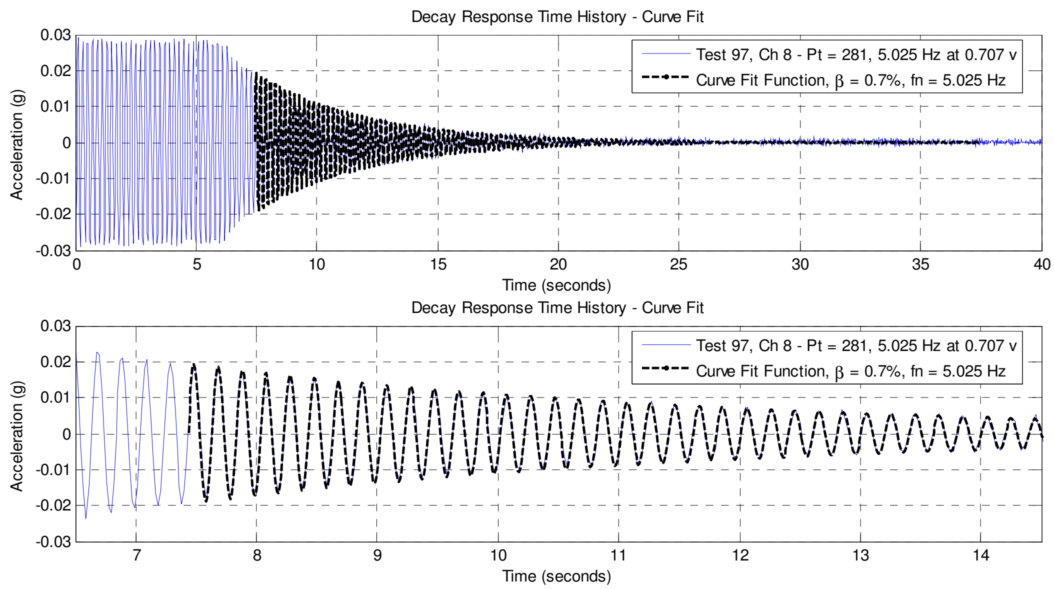


Figure 2.33: Point 281 – Decay Curve Fit, $\beta = 0.70\%$ at 5.025 Hz

For contrast, Figure 2.34 shows the decay from steady state at 4.70 Hz, the other peak frequency of the accelerance FRF of Figure 2.31. This is a clear demonstration of the limitations of the decay curve fit method, as the decay response is obviously not a single frequency response. In fact, the clear beating phenomenon indicates a significant contribution from another frequency, which in this case is the 5.025 Hz dominant frequency of the bay. If two sinusoids with slightly different frequencies are added together, the resulting beating phenomenon is a product of the signals going in and out of phase with one another, alternately adding and canceling, respectively. The period of the resulting beat is equal to the difference in the two frequencies. In Figure 2.34, the period of the beat is approximately 3.2 seconds, indicating a difference in frequency of 1/3.2 ≈ 0.31 Hz. The difference between the actual peaks of the accelerance FRF (5.025 Hz and 4.70 Hz) is 0.325 Hz.

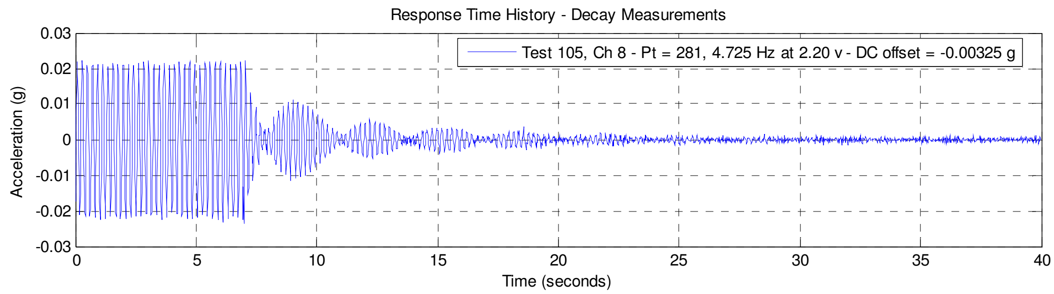


Figure 2.34: Point 281 – Response Decay from 4.725 Hz

The beat of Figure 2.34 is very obvious because of the significant contribution of the other frequency, and the effects of smaller contributions may not always be as clear. For example, Figure 2.35 shows the accelerance FRF for another bay on the floor with several closely spaced frequencies of similar magnitude. The resulting decay from driving at its dominant frequency of 4.875 Hz is shown in Figure 2.36.

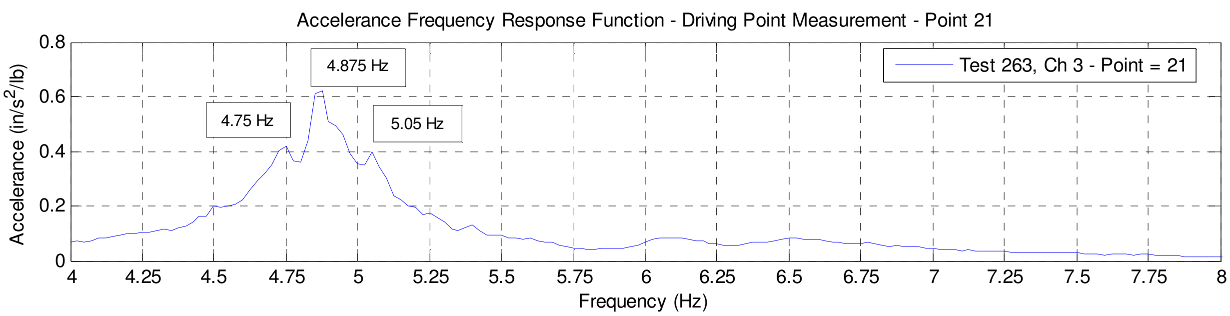


Figure 2.35: Accelerance FRF – Driving at Point 21

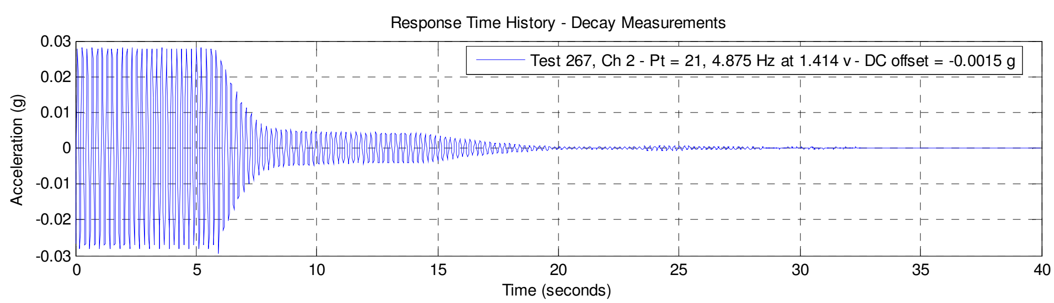


Figure 2.36: Point 21 – Response Decay from 4.875 Hz

While the shape of the decay trace is obviously quite unexpected, it may not be clear that this is the beating phenomenon previously described and a result of the contribution of the other modes. In fact, using only the driving frequencies and one additional frequency in decay expressions (not shown), it did not take much effort to reproduce the same general unexpected shapes of the previous two decay traces as shown in Figure 2.37.

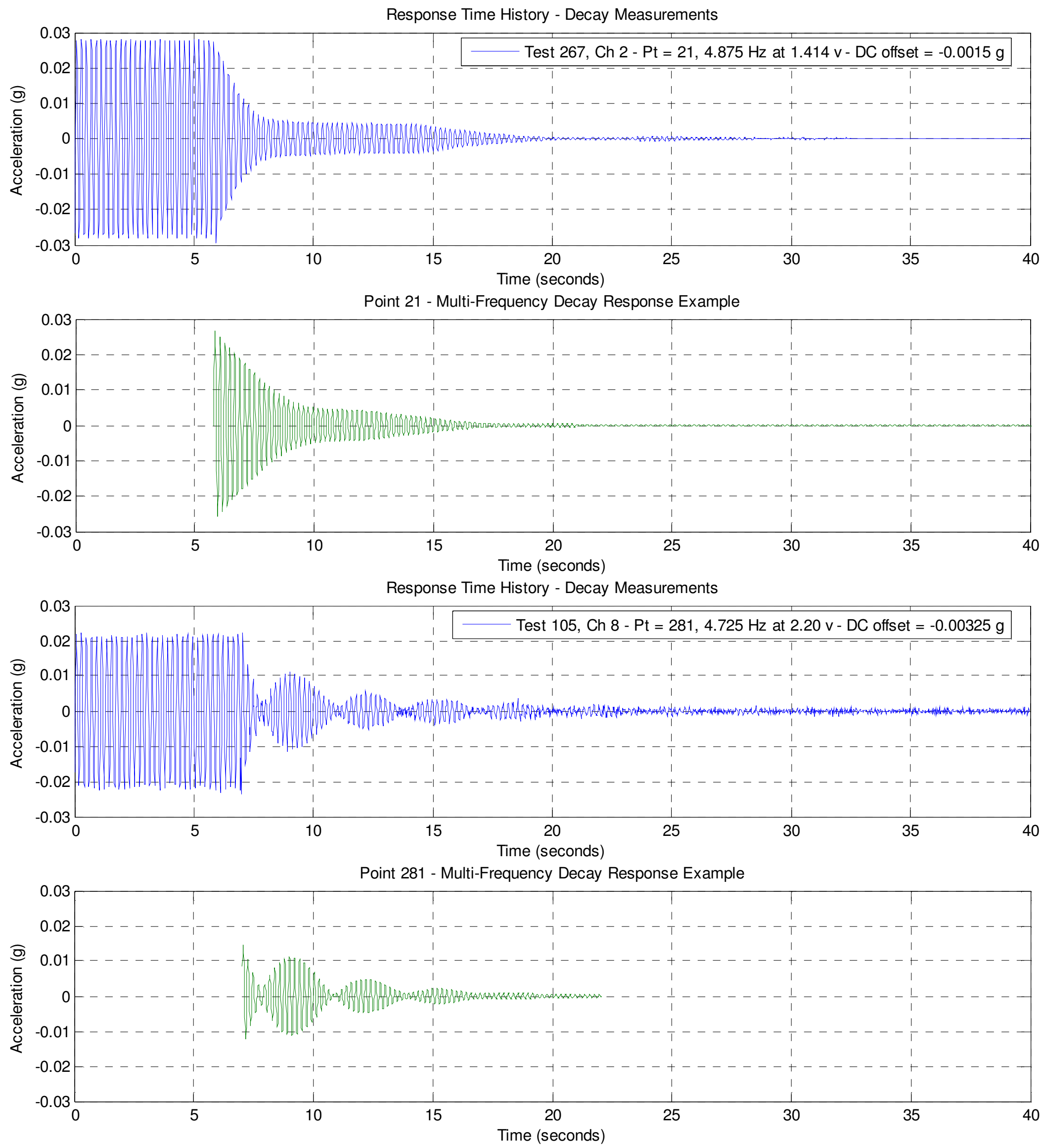


Figure 2.37: Example Analytical Decay Curves with Multiple Frequencies

Obviously, trying to reproduce a series of non-intuitively-shaped decay traces using analytical expressions is not an efficient use of time, especially considering there are other methods of estimating damping for systems with closely spaced modes, namely MDOF curve fitting of the acceleration FRFs using modal analysis software. The previous examples were presented as a cautionary note and to demonstrate the limitations of the decay curve fitting method.

Although it was demonstrated the analytical decay curve fitting technique is a valid method for estimating damping for systems with well-defined dominant modes, a final example is presented to show another situation where the method could be misinterpreted. Figure 2.38 is an accelerance FRF for another bay on the floor, with a clearly dominant peak at 5.075 Hz and a much smaller, but closely spaced, peak at 4.90 Hz. A half-power method estimate of the peak may be skewed due to the contribution of the closely spaced mode, so a response decay from resonance was performed and is shown in Figure 2.39.

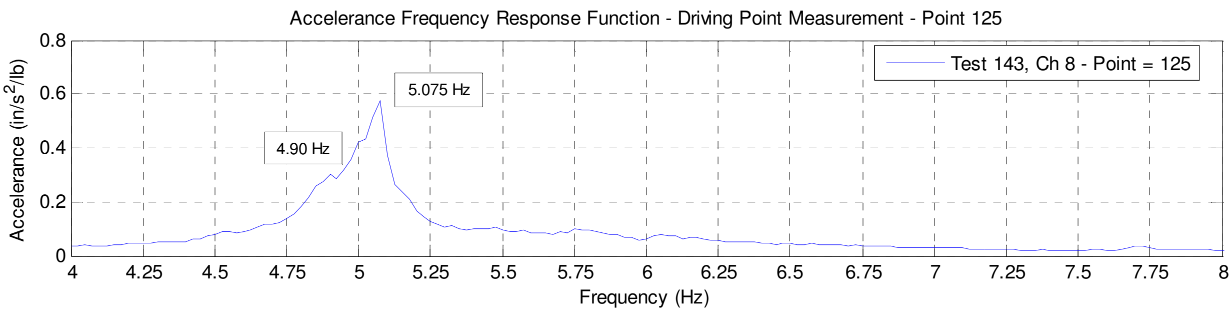


Figure 2.38: Accelerance FRF – Driving at Point 125

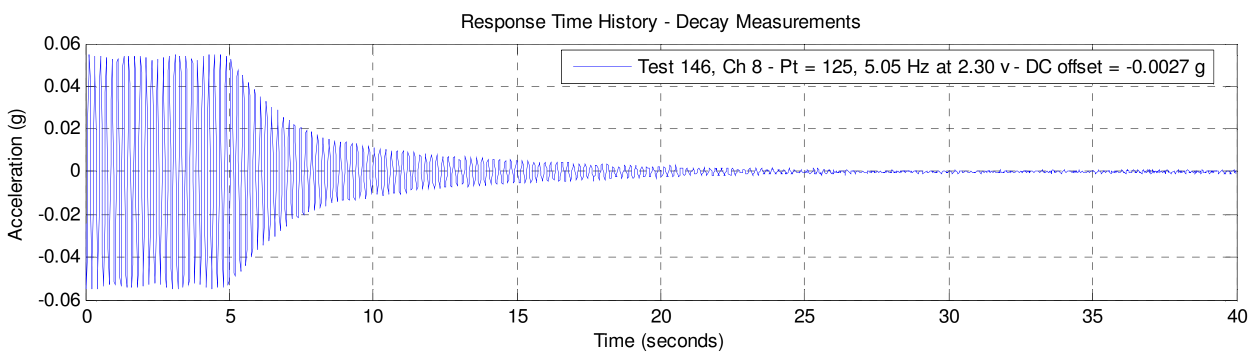


Figure 2.39: Point 125 – Response Decay from 5.05 Hz

At first glance, the decay trace in Figure 2.39 may seem accommodating to an analytical curve fit, however attempts would prove difficult as there seem to be very different decay properties for the first three seconds of decay and the remaining portion of the decay. Again, the beating phenomenon (contributions from multiple frequencies) is the cause, and without an understanding of this the decay of Figure 2.39 could easily be mistaken for amplitude dependent damping, which does often occur (Clough and Penzien 1993). In Figure 2.40, two portions of the curve seemed to be fit well by two different damping values, an initial level of 1.2% over the first three seconds of decay, and a final level of 0.5% starting at an acceleration amplitude of 0.02g. However, this is an incorrect assessment, and although not shown, the actual shape of the decay can be expressed as a combination of the driving frequency of 5.05 Hz and a small contribution from the closely spaced peak frequency 4.90 Hz.

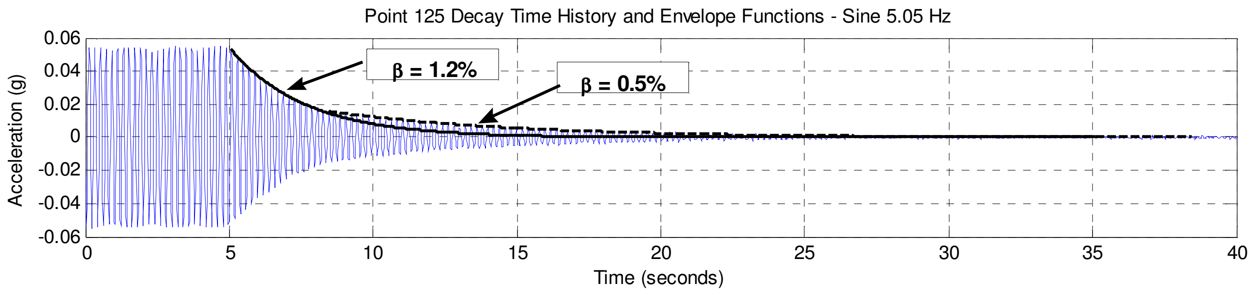


Figure 2.40: Point 125 – Misinterpreted Curve Fit Decay

To summarize the different methods for estimating damping, the half-power method using the driving point accelerance FRFs is the quickest, easiest, and most reliable in the field. Additional verification that the peak of the accelerance FRF is correctly captured may be accomplished through steady-state sinusoidal measurements. Using any method of analysis on the decay response should be approached with caution because it is very difficult to capture a pure single mode response. This applies for using the logarithmic decrement method or curve fitting with a single frequency analytical expression. It should also be noted that this also applies to decay response to impulse loads, such as instrumented or unreferenced heel drops, as well as decay from steady state sinusoidal excitation. Overall, the most reliable method for estimating the damping of a system is the MDOF curve fitting of a set of accelerance FRFs using modal analysis software, and a manual check using the half-power method is a good quality check to ensure that realistic values are computed.

2.4.7 Floor Test Area

As previously stated, the test area of a floor may be quite large, and the time available for testing is typically limited. If the tested floor is within a building under construction, testing may be limited to evenings or weekends due to the extraneous noise in the building during construction activities. The rapidly changing conditions of a building under construction may also pose problems, particularly if the work being done affects the boundaries of the tested floor. If the tested floor is within an occupied office building, similar problems arise due to working hours and the presence of cubicle partitions or other office fit out that can make testing difficult. Thus, the “ideal area to be tested” and the “actual area to be tested” are typically quite different. Within the actual area to be tested, the number and location of test points must be determined on a case-by-case basis, resulting in a compromise based on test objectives, access to test areas, available equipment, and available time to test.

For a modal sweep of a floor area (i.e. measuring the accelerance FRFs over an area to extract the mode shapes), a grid of quarter points within a bay is recommended. For a rectangular bay, the quarter point grid has 25 points, including the middle of the bay where the shaker is typically located and all boundaries of the rectangular area including the columns. Although this may seem like a coarse test grid, it has the ability to capture both single and double curvature of the rectangular floor panel. For a perfectly symmetric rectangular panel, the midpoint is a node for the higher modes with double curvature; however the different boundaries of a real floor system place the node slightly off center. Although the mode shapes of double curvature within a bay are almost always beyond the frequencies of interest for floor vibrations, a shaker located at the middle of the bay is still capable of exciting these higher modes provided the chirp signal covers the frequency range.

One method for reducing the number of measurements on a floor is to not test certain areas based on the assumption of little or no response, like some types of boundaries. Columns, exterior walls, interior full height partitions above and below the tested floor, and shear walls are all locations where the response will likely be small, but may or may not be negligible. Not taking extensive measurements at these locations is acceptable, provided the assumptions are validated by a few measurements.

When testing time is limited but the behavior over a larger area is of interest, a reduced testing protocol consisting of just the centerline measurements or the three interior quarter point measurements over a strip of bays may be used. While not as detailed as a full area set of measurements, this reduced set of measurements can still provide enough resolution to animate the mode shapes and describe the general behavior of the floor along the length or width of the building. The shapes from a centerline set of measurements are essentially 2-D cross-sections of the floor sliced down the centerline. Floor systems are plate structures in two-way bending, so even though a cross-section through the middle of the plate maximizes the chances of capturing the overall behavior of the floor, it may still lead to misinterpretations. Measuring the interior three quarter points along a strip rather than just the centerline takes advantage of a reduced set of measurements, but also provides the ability to capture the two-way behavior.

A minimal reduced set of measurements consists of just measuring the mid-bay accelerance FRFs of all other bays while forcing at the middle of one bay. What this method provides is a point-to-point comparison of the magnitude and phase for the middle of two bays at frequencies of interest. The difference in accelerance magnitude at a dominant frequency gives an indication of the relative acceleration response between the two points in that mode, and the phase will give an indication whether the response is in-phase, out-of-phase, or somewhere in between with the excited bay. Because the lowest frequency modes of a floor system are generally a combination of single curvature in each bay, this comparison can give an idea of the rough shape of the mode by indicating which bays are concave up or concave down with respect to the excited bay. It is recommended that for characterizing the general behavior of the floor over a large area, this is the absolute minimum set of measurements.

2.4.8 Measurement Quality and Other Behavioral Investigations

The text by Ewins (2000) provides general recommendations for checking and maintaining the quality of measured data, and many of those techniques were applied when testing the in-situ floors of the presented research. He recommends five components of a modal test that should be monitored throughout: signal quality, signal fidelity, measurement repeatability, measurement reliability, and measured data consistency, including reciprocity. These components of the test protocol apply to any type of structure to be tested, however they are described in the following section as they apply to the modal testing of in-situ floor systems. Besides describing methods to ensure the quality of the measurements through proper techniques and use of the equipment, other methods are described for checking the behavior of the floor for nonlinearities that would excessively degrade the quality of the measurement. While most of the recommended techniques in this section should be accomplished at all times during testing, others (linearity and reciprocity) are reserved for more detailed investigation of floor behavior or to serve as additional levels of quality assurance if time permits.

Signal quality is a product of the testing technique and use of the equipment, and a good way to monitor the signal quality of modal measurements is the coherence function. Poor signal quality may be the result of extraneous noise, which translates into uncorrelated content and poor coherence. For in-situ floors, the greatest source of extraneous noise is building occupancy; thus it is important to try and conduct modal testing on floors at times when extraneous noise is minimized, such as nights or weekends. Poor signal strength can also cause degraded signal quality because the noise is a much larger part of the measured signal. Poor signal strength often occurs at areas with little response, such as boundaries or long distances from the point of excitation (more than three bays away from the location of the shaker). Measurements of these areas with degraded coherence may not be invalid, they may just have verified a location of little response. Care must be taken, however, to make the distinction between a valid measurement with poor coherence due to minimal response, and just a bad measurement. Increasing the level of excitation to increase the response may force the floor to vibrate at amplitudes well outside the range that is generated by walking excitation, so it becomes a constant balancing act to determine a level of excitation and response useful for the test objectives. For a given choice of excitation level, the only way to accommodate signal quality is to ensure the dynamic range of the measurement channel is saturated as much as possible with the signal. This requires constant monitoring of all accelerometers for each modal measurement and new accelerometer location. As the amplitude of response changes for each sequential measurement, the range of the channel should be adjusted accordingly. Additionally, the coherence should be constantly monitored for each channel and each measurement. *Signal fidelity* refers to the truth of a measurement (Ewins 2000). For example, consider two accelerometers located at the same point on the floor (such as a redundant driving point measurement). If the two accelerometers are recording very different responses, then there is obviously something wrong with one of the channels. By constantly monitoring the response of all channels for each measurement, malfunctions or misrepresentative measurements can be identified during the test, rather than afterwards when it may be too late.

The *repeatability of measurements* was discussed in a previous section and refers to the ability of the same type of excitation to produce the same response each time. It is recommended to always leave one accelerometer with the shaker and force plate while conducting a modal sweep over a test area to record the driving point accelerance FRF. This serves as an additional level of quality assurance because the driving point measurement can be monitored throughout the test. A modal sweep of a large floor area may take a long time and include many sets of measurements; constantly measuring the driving point response can give an indication if the floor is changing over time, which may have a significant effect on the analysis of the measurements (Ewins 2000). Additionally, the *reliability of the measurement* can be checked by applying a different type of excitation signal to the floor to ensure the same accelerance values are obtained. For floor testing, the reliability is checked with manual validation of accelerance FRF measurements using sinusoidal steady state testing, or could even be accomplished by performing instrumented heel drop measurements.

The *measured data consistency* described by Ewins (2000) refers to the assumption that the tested structure will reflect the same frequencies, damping, and mode shapes in the measured FRFs, even though many measurements may be taken sequentially over a long period of time due to the large number of test points. A check of this consistency is the theoretical reciprocity of accelerance FRF measurements. The idea of reciprocity comes from a property of the accelerance FRF matrix, in which the same dynamic properties can be defined from either a row or a column of the matrix. Dynamic reciprocity between two points on the floor shares the same relationship as its static counterpart, although instead of relating the static deflection between two points, a dynamic response is related. This property states $A_{ik}(\omega) = A_{ki}(\omega)$, which means the accelerance FRF measured from forcing at point *k* on the floor and measuring the response at point *i* should theoretically be the same as the accelerance FRF measured from forcing a point *i* on the floor and measuring the response at point *k*. From experiences testing in-situ floor systems, this property holds well for test locations that are close together and does not hold well for long distances away from the point of excitation (greater than three bays away). It should be noted, however, that this is more likely a result of poor signal quality at those long distances than an indication of nonlinear behavior of the floor. Although a check for reciprocity is typically not conducted as a stand-alone investigation, reciprocity tests can be run simultaneously while gathering other sets of measurements. For example, if a reduced testing scheme is used and just the mid-bay accelerance FRFs are measured for direct comparison with one another, the measurement set can be used for reciprocal comparison when the shaker is moved to all the other bays for a similar set of measurements.

As a final recommendation in regard to measurement consistency, if a set of measurements is acquired with the purpose of extracting mode shapes, it is recommended to gather the entire set of accelerance FRFs at one time, preferably within the same day. Gathering the set of measurements in the least amount of time minimizes the effects of any changing conditions of the floor, which was occasionally observed testing the floors in the presented research. Shutting down the test equipment for the day and starting again the next, which theoretically should not have any effect on the consistency of the measurements, may inadvertently introduce some small amount of inconsistency in the measurements that may not be acceptable. Taking the measurements in one day, in the least amount of time, minimizes any of these unknowns.

One of the fundamental assumptions in modal analysis is that the structure is linear, which means it will exhibit the same dynamic properties and response at all levels of excitation. If the level of excitation doubles, the response should double, and the other dynamic properties (frequencies, damping, and mode shapes) should stay the same. This is another example of an assumption that should be validated at a cursory level when testing in-situ floors, if not validated extensively. Extensive linearity tests can take a lot of time, especially if entire sets of data are acquired multiple times at different levels of excitation for comparison. Thus, the extent of linearity checks should be determined as part of the overall objectives of the test beforehand. The most straightforward method for checking linearity of floor systems is to run a burst chirp signal at different levels of excitation to measure the accelerance FRFs and compare the peak frequencies and the magnitudes/phases at those peak frequencies. Large differences in either the frequencies or the magnitudes of the peaks will give an indication of any gross nonlinearity in the floor system. Another method of checking for nonlinearities is using the coherence function. It should be noted that the coherence function is simply the measure of linear dependence between the input force and the acceleration response as a function of frequency, and has nothing directly to do with level of applied force. For example, two sets of measurements taken with two different levels of excitation, each using multiple averages, may both have excellent coherence because the input force and acceleration response was well correlated for each given force level. Formenti (1999, 2000) suggested taking averages with *varying* force levels between averages to determine any force nonlinearities. His example for this technique was an FRF measured using three averages with a force hammer. The first two averages were taken with the same force level, but for the third average a much larger or smaller level of force was applied. Any degradation of the coherence between the second and third averages would be an indication of force nonlinearity. Extending this technique to in-situ floor testing, a floor could be tested in this manner using a strong/weak instrumented heel drop. For shaker excitation, the approach is a bit more complicated, but possible if the amplifier could be manually adjusted *between* averages to vary the chirp signal input force from the shaker. It should be noted that this type of testing for nonlinearities was not conducted in this research and is only included here for reference.

2.4.9 Summary of Experimental Testing Methods

The previous sections provided detailed descriptions of the various experimental testing methods and measurement types available for dynamic testing of in-situ floor systems. The availability of testing equipment significantly limits the ability to perform several of the described techniques or capture certain types of measurements, but these limitations are second only to those created by the time available to test a floor. However, simply having the appropriate equipment and ample time to conduct testing does not guarantee high quality measurements; this is a product of the proper application of the equipment and continuous monitoring of the data throughout testing to ensure quality measurements are acquired. The following is an outline of a recommended approach to planning and conducting any level of in-situ floor vibration testing and investigation, so that the previously discussed compromises can be efficiently resolved before testing, and the recommended techniques can be applied to ensure quality measurements:

1) <u>Clearly identify the overall objectives of the investigation</u>. The objective may be to simply get a rough idea of the floor's frequencies, or the objective may be to gather a detailed set of high quality measurements for modeling and response simulation. Regardless, these objectives must be identified up front, and may even be determined by a building owner, consultant, or researcher. Note that the original and the final objectives may change based on available equipment, or more likely, the available time to test, which is the greatest limitation on the feasibility to perform the objectives.
2) <u>Establish rough test parameters based on test objectives and allotted time</u>. Various test parameters affect the ability to conduct a test in the allotted time and virtually all require some degree of compromise.
    - Determine the desired measurement types to be gathered for the tested floor (accelerance FRFs, referenced/unreferenced time histories, decay from resonance, autospectra, etc.).
    - Identify the area to be tested and plan out a test grid for that area.
    - Identify digital signal processor settings of frequency bandwidth, frequency domain resolution, time record lengths, etc.
    - Determine the types of excitation to be applied based on the desired measurement types (chirp signals, sinusoidal, instrumented/unreferenced heel drops, etc.).

The term “rough test parameters” is used because pre-test parameters must be flexible and accommodating to the unexpected, such as equipment malfunctions, differing site conditions, odd floor behavior, etc. It is wise to have several contingency plans for the unexpected, including a “core” or minimal set of measurements, and an “ideal” set of measurements that would be useful to gather if time permits.

3) <u>Establish the level of additional behavioral investigation to be accomplished</u>. Although technically this may fall under establishing test parameters, it is mentioned separately here to refer to checks on linearity and reciprocity, or perhaps investigations using decay or testing boundary behavior, which can take a significant amount of time if not planned for accordingly.
4) <u>Conduct floor vibration testing using recommended best practice techniques, and adjust test parameters to accommodate the actual conditions of the tested floor</u>. For example, to acquire high quality accelerance FRFs, it is recommended to use an electrodynamic shaker on a force plate using a burst chirp signal for excitation; however the parameters of the chirp signal (frequency range, duration) can typically only be determined from preliminary measurements on site. Other test parameters are also determined on site, like the number of averages required to achieve acceptable coherence. Very low levels of damping may require a finer frequency resolution to adequately define the peaks of the FRFs, resulting in longer time record lengths. The rough test parameters, and perhaps even the feasibility to achieve all of the overall objectives, may change based on actual behavior of the floor. Again, it is recommended to be flexible and have contingency plans.
5) <u>Constantly apply recommended techniques for checking the quality of measured data</u>. Example recommended techniques include constant monitoring of the coherence function, acquiring redundant driving point accelerance FRFs, and saturating the dynamic range with the measured signal. Ensuring quality measurements in the field is critical as it is usually impossible to get a second chance to re-test an in-situ floor under the same conditions.
6) <u>Post process measurements</u>. The last step in the process, and not expanded on here, is post processing the data from a set of measurements to meaningful results. Depending on the overall objectives and the types of measurements acquired, this may range from a report detailing levels of acceleration and comments on frequency, to a full array of parameter estimation including frequencies, damping, and visually displayed mode shapes.

As previously stated, almost every aspect of testing floors is a compromise, and the choices of techniques, test settings, and measurement types have a significant effect on the time and effort required to test as well as the quality of the acquired data. The experimental testing methods and recommendations presented in this section provide a fundamental set of best practice techniques for the dynamic testing of in-situ floor systems.

2.5 Classification of Floor Vibration Testing

The previous sections in this chapter provided some theory behind modal testing, the different types of equipment available for conducting dynamic testing, the types of information available using modal testing and other types of dynamic measurements, and a set of recommended best practice testing techniques to ensure the quality of the acquired data. The presented information ranged from broad philosophy (setting overall test objectives) down to tedious techniques (choosing specific settings or types of measurements), creating numerous items for consideration when planning and conducting any level of floor vibration test. With that in mind, it may be beneficial to classify floor testing into specific categories, thus establishing a common language for building owners, engineering consultants, experimentalists, and academics intent on troubleshooting, designing, evaluating, or researching the floor vibration performance of in-situ structures.

The idea of classifying certain levels of dynamic investigation is not new. The United Kingdom's Dynamic Testing Agency (DTA) previously classified five different Levels of Test in its *Handbook of Modal Testing* (DTA 1993), each of which describes the level of detail/accuracy required for application. They are as follows:

Level 0: Estimation of natural frequencies and damping factors; response levels measured at few points; very short test times.

Level 1: Estimation of natural frequencies and damping factors; mode shapes defined qualitatively rather than quantitatively.

Level 2: Measurements of all modal parameters suitable for tabulation and mode shape display, albeit un-normalized.

Level 3: Measurements of all modal parameters, including normalized mode shapes; full quality checks performed and model usable for model validation.

Level 4: Measurements of all modal parameters and residual effects for out-of-range modes; full quality checks performed and model usable for all response-based applications, including modification, coupling, and response predictions.

The Levels of Test described by the DTA are purposely vague, as they apply to the full range of modal testing and are not tied to any specific structure being tested nor equipment employed. Another noteworthy attribute is that they apply to modal testing, which implies the traditional setup of simultaneously measuring both the input force and corresponding response.

In floor vibration testing, it is very common to take more generic response-only dynamic measurements and much less common to have the standard input-response modal testing setup. In terms of the detail/accuracy outlined in the DTA's Levels of Test, the accuracy of any floor testing measurements is a product of the quality/capability of the available equipment and experience of the experimentalist (which is arguably of equal importance). The application of the extracted information is subject to its availability; only certain types of measurements can quantify certain information. Stated differently, *only certain <u>types of equipment</u> and testing methods can take certain types of measurements and can quantify certain information*…so in this respect, not all dynamic measurements are created equal. Thus, a common ***Classes of Floor Testing*** classification system is defined here in terms of:

1) the methods and equipment used (or required) for testing
2) the <u>extent</u> of the useful information that can be extracted from the tests

A standard classification such as the ***Classes of Floor Testing*** will benefit all of the above mentioned parties, as there will not be any false expectations in the capabilities of the reportable results, nor in the equipment, time, and cost required to obtain the desired results. Below are the proposed ***Classes of Floor Testing*** in order of increasing levels of detail, complexity of testing, and most certainly increasing cost in terms of equipment, time, and testing/analysis effort:

<u>Class-I Testing</u>: Single Channel Response Testing

Equipment: Single channel spectrum analyzer.

Extent of Useful Information: Single point acceleration response time history, autospectrum for estimating frequency content, limited estimates of damping using decay of heel drops or from bouncing excitation.

<u>Class-II Testing</u>: Multi-Channel Response Testing

Equipment: Multi-channel spectrum analyzer.

Extent of Useful Information: Multi-channel (multi-location) acceleration response time histories, autospectra of acceleration response for estimating frequency content, limited estimates of damping using decay of heel drops or from bouncing excitation. The multi-channel response gives the ability to compute an operating deflection shape frequency response function between two response locations, in which the response of one location on the floor is used as a reference channel to compute the FRF between the responses at the other locations. This gives the capability to roughly define shapes using the magnitude and phase information, although this is not a true measure of accelerance.

Class-III Testing: Impact Induced Modal Testing

Equipment: Multi-channel spectrum analyzer. Force transducer for recording impact force such as a force plate for performing instrumented heel drops or a large instrumented impact hammer.

Extent of Useful Information: Simultaneous input force and multi-channel response time histories, autospectra of input force and acceleration responses, accelerance FRF measurements and all the associated benefits of that data, such as the ability to perform curve fitting operations for parameter estimation. The minimum level of testing for detailed estimates of damping, accelerance values, and scaled mode shapes. In general, this is the minimum set-up required for full modal analysis, although not the most reliable method for achieving high quality data due to the potential inconsistency of the applied impact force.

Class-IV Testing: Shaker Induced Limited Modal Testing

Equipment: Multi-channel spectrum analyzer. Electrodynamic shaker. Force transducer for recording shaker force, such as a force plate.

Extent of Useful Information: Simultaneous input force and multi-channel response time histories, autospectra of input force and acceleration responses, accelerance FRF measurements of generally higher quality than achieved through impact testing. The higher quality measurements provide more reliability for parameter estimation (frequencies, damping, general mode shapes). While this set-up is capable of full modal analysis, it only involves minimal or limited area testing for characterizing specific information. Examples of limited modal testing would be centerline or mid-bay accelerance FRFs, enough to perform MDOF parameter estimation for multiple response/reference locations and to display basic mode shape information due to the limited measurements.

Class-V Testing: Shaker Induced Full Modal Testing

Equipment: Multi-channel spectrum analyzer. Electrodynamic shaker. Force transducer for recording shaker force, such as a force plate.

Extent of Useful Information: Every capability of Class-IV testing, however one step further to include full modal analysis to experimentally quantify the shapes over a full area.

Note: For each Class, it is assumed that the response measurements are taken with accelerometers, either one or many, as is typical for floor testing. These accelerometers and the required cabling are not reported under the equipment heading. It should also be noted that any Class automatically includes the capability to perform all lower Classes of testing by the nature of the equipment. The only exception to this is Class-III testing cannot be performed without a force transducer (i.e. force plate), which may not be available if shaker testing (Class-IV and Class-V) is accomplished by attaching an accelerometer to the shaker armature as a means for computing input force. This method of force measurement is discouraged for floor system testing.

For the tested in-situ floors in the presented research, the Class of Testing fell somewhere between Class-IV and Class-V, in which modal sweeps were made over various areas of the floors to define mode shapes, but there were also many behavioral investigations that did not necessarily involve the full area (linearity, reciprocity, decay investigations). Hopefully, this classification system can establish a common language and understanding of the capabilities (and limitations) of the different types and techniques for floor testing. A potential application of the classification system would be to help consultants explain the different options of testing to their clients, allowing for a better understanding of the effort (and cost) of the proposed Class of Testing.

As a final comment, it should be noted that none of the described theory, digital signal processing, equipment, or techniques presented in this chapter were specific to the steel composite office floors of the presented research. In fact, the techniques also extend themselves to testing other types of large in-situ structures such as footbridges or stadia, which share similar behavior of low frequencies and damping and have the potential vulnerability for excessive vibration due to resonant response.