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Dynamic Characteristics of Structures Extracted From *In-situ* Testing

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ABSTRACT

The paper discusses two primary areas of interest in a structural evaluation. First, *in situ* measurements are used to confirm the predicted structural stiffnesses and resonance frequencies. Second, the evaluation characterizes the manner in which vibrations are propagated through the structure. Methodologies are presented for carrying out these measurements, and typical data are given.

Keywords: structural dynamics, *in-situ* testing, vibration propagation, vibration attenuation

1. INTRODUCTION

A modern chip manufacturing facility is typically comprised of a two- to five-story concrete structure, commonly denoted as a “fab”, with a waffle or grillage process floor supported on columns and shearwalls. Microvibration performance of these floors is of paramount importance for chip manufacturing which currently uses linewidth technology down to 0.25 or 0.18 micron.

In this paper, a mathematical model is defined which includes various components affecting the vibration performance of a facility. In understanding the complexity of this problem, one appreciates the limitations offered by a purely analytical approach. Focus is then turned to experimental data obtained from several of these facilities using forced excitation. The test methods used by the authors are reviewed. Data are presented which illustrate the effects of structural isolation breaks (SIBs), floor to floor propagation, building-to-building propagation, and other transfer functions. Data are also provided which quantify some of the fundamental dynamic parameters (i.e., damping, resonances, mobility, etc.) of the fab structures.

There are two primary areas of interest in an experimental structural evaluation using forced excitation. First, a study of this sort confirms the predicted structural stiffnesses and resonance frequencies by means of *in situ* measurements. This is accomplished using the building's measured response to dynamic loading. Second, the study can characterize the manner in which vibrations are propagated through the structure. Specifically, it quantifies how vibrations are amplified or attenuated by the structure being evaluated. This is done by applying dynamic loading over a range of frequencies and measuring the response at the drive point and at remote locations. The response curve at the location where the load is applied—the “drive point”—will provide a means to estimate the resonance frequency of the bay containing the drive point, and can be used to calculate the drive point stiffness. Comparing the response at a remote location with that at the drive point will provide a measure of the structure's attenuation between the two locations.

It is important to note that the purpose of a structural evaluation is to determine *in situ* properties. This paper will present data from several such evaluations, but the reader is cautioned that the specific results of those studies should not be taken as generally applicable to all facilities. Even structures that are nearly identical can yield different results because of variations in soil properties.

2. IDEALIZED STRUCTURAL MODEL

Figure 1 shows idealized models of three typical fab configurations. Figure 1(a) shows a facility with a two-level subfab, a four-level support wing, and a structural isolation break (SIB) between the two. The facility represented by Figure 1(b) has only one level of subfab, with a two-level support wing, and a SIB between the support wing and the fab. Figure 1(c) is identical to Figure 1(b) except for the absence of the SIB. Locations of potential vibration sources and receivers are indicated with "S" and "R", respectively.

The floors are represented by plates, which have stiffness, mass, and damping, and can propagate vibrations primarily as shear and bending waves within the frequency range of interest (up to 100 or 200 Hz).

Columns are typically assumed to be relatively massless when compared to the floor plates in a fab. They are of dynamic importance because: (1) they act as support points for the plates; (2) they act as springs in complex spring-mass systems (in which the mass is supplied by the floor and whatever it supports); and (3) they are the primary means by which dynamic forces travel from the suspended floors to the soil.

The soil beneath the facility is perhaps the most difficult part of the model to represent. Beneath a foundation, the soil acts as a spring and dashpot, resisting the static and dynamic forces from the columns. It also acts as a propagating medium, carrying vibrations from one footing to another.

A SIB causes dynamic energy generated on one side (say, on an elevated floor at fab level in the support area) to be propagated via the soil. Without a SIB, some energy can be transmitted directly from the support area floor slab to the fab floor slab.

3. INSTRUMENTATION USED FOR STRUCTURAL MEASUREMENTS

A structural evaluation can be carried out with a relatively small suite of instrumentation (though it is heavy and bulky). The minimum suite consists of:

- a shaker system, including waveform generator;
- motion sensors; and either
- a multiple-channel analyzer, or
- multiple single-channel analyzers.

A shaker is an electrodynamic device which will generate a dynamic force in proportion to an input electrical signal. There are two methods to measure the force generated by the shaker: (1) a force sensor between the shaker and the floor; and (2) an accelerometer attached to the shaker armature (the moving part). The accelerometer measures the acceleration of the moving mass; the force applied to the floor is equal to the weight of the moving mass times its acceleration in g's.

The motion sensors measuring the floor response should have adequate sensitivity and noise floor to measure the low-amplitude ambient vibrations typically found in a fab facility. At locations close to the shaker, the vibration response from a shaker input may be significantly in excess of ambient, but at remote locations, the vibrations will drop to the same order of magnitude as ambient.

The spectrum analyzer is perhaps the most important tool in the suite, and it is generally best to use one with a minimum of two channels, which can simultaneously sample and process signals representing force from the shaker and the response of the floor. In situations where long cable runs would be impractical or impossible, two single-channel analyzers can be used, but one loses some of the interpretive cross-channel quantities (such as phase and coherence) available with the two-channel analyzer.

The structural evaluation should be done after the structure is complete and accessible, but before mechanical systems are running (which results in higher ambient levels). Whenever possible, we carry out an evaluation during the evening, after construction activities have been reduced or have ceased. The effects of impact-causing construction activities on drive point measurements can be overcome, but they can be quite troublesome for propagation measurements.

4. BASIC STRUCTURAL MEASUREMENTS

An accelerance curve is a spectrum showing the acceleration response resulting from a unit force, the frequency of which is swept over a selected frequency range. Likewise, mobility and receptance curves are the corresponding spectra showing the velocity and displacement response, respectively, from that same unit force. An acceleration, velocity, or displacement spectrum can be obtained using any swept force, and then divided by the force spectrum to “normalize” it to that associated with a unit force.

We typically use, for these studies, a sinusoidal force for which the frequency is swept from a low frequency to a higher frequency over the duration of one FFT time sample. The swept force is repeated for each FFT sample (a "chirp"), and at least ten samples are taken and averaged. (When multiple single-channel analyzers are used, the sweep rate of the swept sine is decreased considerably, so that the time required to traverse the frequency range takes many FFT time samples.)

Figure 2 shows the basic data we obtain from a drive point measurement. It consists of three graphs, all plotted against a linear frequency axis. In the top graph, (a), is shown the measured drive point accelerance magnitude spectrum. The middle graph, (b), shows the measured phase between the acceleration of the shaker armature and the acceleration of the floor. The bottom graph, (c), is the measured coherence spectrum, which indicates in numerical form the "quality" of the measured accelerance spectrum. [The value of coherence is between 0 and 1. A coherence of 1 at a particular frequency indicates complete repeatability (from one FFT sample to the next) of the accelerance spectrum at that frequency; a value of zero—or nearly zero—indicates poor repeatability and thus a poor-quality accelerance value.]

It is a simple transform using $1/(2\pi f)$ and $1/(2\pi f)2$ to obtain mobility and receptance from accelerance. Figure 3 shows accelerance, mobility, and receptance spectra for a measurement on a fab floor; the latter two were calculated from the accelerance. Two uses of these spectra are illustrated. Accelerance is the form in which the support criterion for one photolithography tool is given. The accelerance spectrum of Figure 3(a) is shown with this criterion. Also, we find that at low frequencies, the receptance spectrum is asymptotic to the inverse of the static stiffness of the structure at the test location, as shown in Figure 3(c).

Two-channel spectrum analyzers can be configured to obtain these quantities automatically using the frequency response function (FRF). The "response" signal (either the measured floor acceleration, velocity, or displacement) is measured simultaneously with the "input" signal (the dynamic force, or in our case, the acceleration of the moving mass generating the dynamic force). The two signals are digitized and an FFT calculation is used to produce the magnitude and phase of the FRF.

It is also possible to obtain these quantities using two unsynchronized spectrum analyzers, but the phase and coherence cannot be calculated. To obtain an accelerance, mobility, or receptance curve in this manner, the procedure is as follows. Both analyzers are set to “maximum rms” mode. One analyzer will store the maximum floor response (either acceleration or velocity) as the function generator signal is advanced through the frequency range. The other analyzer stores the maximum acceleration of the shaker armature (which is proportional to force) as a function of frequency. The resulting acceleration, velocity, or displacement spectrum from the floor is divided by the force spectrum to obtain the accelerance, mobility, or receptance spectrum using a spreadsheet. For the results to be valid, it must be verified that the response measured is significantly above the ambient vibration level.

4.1 Drive Point Properties

The “drive point” accelerance, mobility, or receptance spectrum shows the acceleration, velocity, or displacement at the *point of application* of the dynamic force, and gives an indication of how the drive point will respond to a dynamic load as well as identifying the resonances which that loading can excite. The receptance at low frequencies can also be used to obtain the point stiffness.

An accelerance spectrum $A(f)$ is calculated using Equation (1),

$$A(f) = [\ A_o(f) \ /\ F(f)\ ], \tag{1}$$

where $A_o(f)$ is the measured acceleration spectrum, and $F(f)$ is the measured force spectrum. We calibrate the system in terms of *g* (the gravitational constant) for acceleration and pounds for force. To be consistent with published accelerance requirements, we express accelerance in metric units of 1/kg, equivalent to m/sec^2-N. The force spectrum is calculated with Equation (2),

$$F(f) = A_i(f) \times W, \tag{2}$$

where $A_i(f)$ is the acceleration spectrum measured on the shaker armature, expressed in *g*, and W is the weight of the moving armature, expressed in units of force.

The mobility spectrum $M(f)$ and receptance spectrum $R(f)$ are calculated using Equations (3) and (4), respectively.

$$M(f) = A_o(f) \ /\ (2\pi f) \tag{3}$$

$$R(f) = A_o(f) \ /\ (2\pi f)^2 = M(f) \ /\ (2\pi f) \tag{4}$$

4.2 Fundamental Resonance Frequency

The peaks in the drive point spectra indicate resonance frequencies of the floor system. The "fundamental" mode of the structure associated directly with the drive point is at the lowest frequency at which there is a peak and the accelerance phase is 90°. In Figure 2, the fundamental is shown with arrows on the accelerance and phase plots. (This illustrates an important reason why analysis with a multiple-channel analyzer is inherently "better" than the use of measurements from two single-channel analyzers. An identical accelerance spectrum might be obtained with the latter approach, but confusion might arise as to which of the "humps" was actually the fundamental. It is much more clear with the assistance of the phase spectrum.)

4.3 Floor Stiffness Measurements

We have historically taken two approaches to measuring vertical stiffness at a point in a structure, one using a fixed-frequency dynamic load, and the other using a swept-frequency load.

The simplest approach is to use the shaker to apply a sinusoidal force at a fixed, low frequency, such as 3 Hz, and measuring the structure's sinusoidal response to that force. The input force is measured directly, by placing an accelerometer on the inertially loaded shaker armature. The response of the floor to the input force is measured on the floor adjacent to the shaker. Then, the static stiffness is calculated using Equation (5),

$$K_S \approx F_i/\delta_o, \qquad (5)$$

where F_i is the input force, and δ_o is the response displacement. Ideally, the driving frequency should be as low as possible so that the static stiffness is closely approximated.

A more rigorous approach is to calculate the receptance spectrum and determine the value to which the curve asymptotes. That value is the inverse of stiffness. This approach offers improved reliability owing to the fact that we can work with a trend, reducing the problems associated with poor or marginal low-frequency coherence.

4.4 Transfer Properties

The "transfer" accelerance, mobility, or receptance spectrum shows the acceleration, velocity, or displacement at a *remote* location due to the applied force. It gives an indication of how that remote point will respond to a dynamic load at the drive point. It can be used to predict the acceleration or velocity response at a distance from a known applied force from a source such as an out-of-balance fan, pump, or chiller.

A transfer spectrum can be compared with the corresponding drive point spectrum to obtain the *attenuation* or *amplification* provided by the structure between the two locations. The *amplification factor* spectrum $AF(f)$ can be calculated using Equation (6),

$$AF(f) = A_{TRANSFER}(f) \,/\, A_{DRIVE\ POINT}(f) \qquad (6)$$

where $A_{TRANSFER}(f)$ and $A_{DRIVE\ POINT}(f)$ are the transfer and drive point accelerance spectra. (Mobility or receptance spectra can be substituted for the accelerance spectra.) $AF(f)$ is unitless. Amplification is represented by values greater than 1, attenuation by values less than 1. The *amplification level* spectrum $AFL(f)$ is calculated using Equation (7).

$$AFL(f) = 20 \log AF(f) \qquad (7)$$

$AFL(f)$ has units of decibels. Amplification is represented by positive values, attenuation by negative values.

Figure 4 shows a typical set of transfer measurements from a swept-sine test. Figure 4(a) shows transfer mobility (heavy line) and drive point mobility (light line). Figure 4(b) shows the coherence for the transfer mobility. Figure 4(c) shows the change in vibration level from the source to the receiver. On the left vertical axis we see the "amplification factor," the ratio between the transfer response and drive point response, as defined in Equation (6). On the right axis, we see the corresponding change in decibels, as defined in Equation (7).

Occasionally, it is necessary to make transfer measurements under less-than-desirable conditions, in which the ambient vibrations are severe enough that a swept-sine measurement leads to poor coherence. In some of these cases, improved coherence can be obtained by using sinusoidal vibration at a discrete frequency. When we use a swept frequency tone, the force actually "dwells" in a particular frequency band for only a small fraction of a second. (For example, if we are sweeping from 0 to 100 Hz in 8 seconds, then the "sweep rate" is 12.5 Hz/sec. The frequency resolution of an 800-line, 0-100 Hz spectrum is 0.125 Hz. Thus, the tone dwells in a particular band for only 0.01 second out of the eight.) If the ambient amplitude is high, it dominates the response averaged over 8 seconds. However, if the forcing frequency and amplitude are constant and the response amplitude only slightly exceeds the ambient, the average is dominated (in that band) by the forced response.

For example, the data shown in Figure 5 were obtained from measurements that had to be made during the day while construction was on-going. Swept-sine measurements were contaminated by earthwork outdoors and by installation of large exhaust ducts in the subfab, resulting in coherence on the order of 0.5 or less. Transfer functions were measured while the shaker was excited with a sinusoid at eleven discrete frequencies (the one-third octave band center frequencies from 6.3 to 63 Hz). Figure 5(a) shows the transfer and drive-point mobility data (the drive point data could be measured using the swept sine). Figure 5(b) shows the coherence of the transfer measurements. Figure 5(c) shows the change in level, which is defined only at the discrete frequencies. In all three parts of Figure 5, solid symbols represent data for which the coherence is greater than 0.8.

5. EXAMPLES OF "TYPICAL" DRIVE POINT MEASUREMENTS

We can define a "soft" floor as one in which the midbay point stiffness is significantly less than the column stiffness, or the fundamental floor resonance frequency is significantly less than that of the column. In this case, the columns act as rigid support points, and nodal lines pass through the columns. When the midbay point stiffness or slab resonance frequency is nearly equal to that of the column, we call this a "stiff" floor,[*] in which case the columns supporting the driven bay do *not* act as rigid supports. They participate in the fundamental modeshape, and the nodal line for the fundamental lies beyond the columns closest to the drive point. Floors designed to achieve optimal performance for submicron semiconductor production are usually "stiff;" a support area floor is most likely "soft." Whether "stiff" or "soft", the response of a floor when excited at midbay includes interaction with adjacent bays, and is to some extent a function of the geometry of the fab floor as a whole (not just the geometry of the driven bay).

The mobility spectra shown in Figure 6(a) were measured in a "stiff" fab, at a midbay location on a waffle slab and above a column supporting it, and on the mat foundation supporting the columns. The primary resonances at the three locations are identified. Figure 6(b) shows similar data measured at a midbay and column location on a "soft" waffle slab in a fab support area. The latter floor was supported on spread footings. Separate mobility spectra for the foundation are not available.

In Figure 6(a) we see the cumulative effect of resonances as we move up from the mat foundation to the column and to the waffle slab. The mat foundation resonance at 19.0 Hz is visible in the spectra measured at the top of the column and the midbay location. The column resonance at 42.3 Hz is not apparent in the mat foundation spectrum, but is present in the midbay spectrum. There are three peaks to the right of the column resonance in the waffle response. The waffle resonance at 57.5 Hz was identified as the fundamental resonance from the phase spectrum (see Figure 2). It does <u>not</u> show up in the other spectra.[†]

In Figure 6(b) we see the condition more typical of "soft" floors, in which there are many resonance peaks between the slab fundamental (12.0 Hz) and the column resonance (45.4 Hz), and no evidence of the column resonance in the slab mobility spectrum. This is because the column acts as a nodal point in the fundamental and intervening modeshapes. It is interesting to note the several extra peaks between 13.3 Hz and 15.0 Hz, indicating modeshapes that involve varying forms of interaction with immediately adjacent bays.

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[*] The notion of "stiff" and "soft" floors is discussed in more detail in Amick, H., *et al.*, "Design of Stiff, Low-Vibration Floor Structures", Proceedings of International Society for Optical Engineering (SPIE), Vol. 1619 (November, 1991).

[†] We should note that there is a multitude of closely-spaced modes near what is identified as the "fundamental" resonance frequency. Therefore, the spectrum shape presented is reflective of this fact. One should not infer damping values from these curves.

Figure 7(a) shows drive point mobility curves (in log-log format) measured on five different floors of different stiffness and support conditions. They range from a concrete deck on steel framing composite floor sized only for fan deck equipment load to a mat foundation. We see that the mobility at resonance is spread over more than two orders of magnitude, and that damping increases (the peak is shorter and fatter) as we move from "soft" to "stiff." Some of this change is due to the fact that in a "soft" floor the columns just act as "rigid" support points for the modeshapes, but in a "stiff" floor the columns participate in the modeshapes. Thus, in stiff floors (and in the mat foundation) the damping of the soil (via the foundation) contributes to the overall system damping at resonance. If the top of the column does not move (as is the case with the "soft" floor at resonance), then the foundation is not subjected to a loading that will generate damping forces.[*]

Figure 7(b) shows the same measurements transformed to receptance. We clearly see the asymptotic nature of the low-frequency end of each spectrum. The asymptote is the inverse of stiffness. The asymptotes decrease as the floor stiffness increases.

6. PROPAGATION IN FLOOR SLAB

Figure 8(a) shows drive point mobility at midbay location on a stiff fab floor and transfer mobility at midbay for one, two, and ten bays away. The coherence spectra corresponding to the three transfer mobility spectra are shown in Figure 8(b). The changes in amplitude (from the drive point to the transfer location) are shown in Figure 8(c). We see that in the distance of one bay from the drive point the amplitude is attenuated about 10 decibels at most frequencies. As we move one more bay away, the attenuation is between 10 and 30 decibels. The greatest attenuation is between 60 and 70 Hz. At ten bays distant, the vibration levels at all frequencies have been attenuated at least 20 decibels, and at some frequencies, as much as 50 decibels. These variations with frequency are a function of the individual dynamic properties of the building under test.

It is sometimes useful to know how vibrations at specific frequencies are attenuated with distance across a floor slab. For instance, if we have a motor of a particular rpm, we can predict how the vibration characteristic of its shaft speed will diminish with distance. We can measure transfer mobility at many different distances from the drive point; several such mobility spectra are shown in Figure 8. We can extract from each transfer mobility spectrum the magnitude at the selected frequency and plot this as a function of distance.

Curves can be fit to the attenuation data which will simplify the prediction process. Figure 9 shows this procedure at two frequencies, 30.0 Hz and 48.5 Hz. Data were measured in 20 sequential midbays, but only the data points for which coherence is greater than 0.7 have been shown. Three types of propagation models have been used to fit these data:

- A "linear" model, in which we plot the log of amplitude as a function of distance in a straight line, for which the slope can be given as decibels (dB) per unit length, say dB/ft or dB/bay.
- A "power" model, in which the data are fit to the equation $M = M_1 L^g$, where L is the distance from the source and g is the slope of the best-fit curve on a plot of the log of amplitude versus the log of distance. The slope may be expressed in decibels per doubling of distance (abbreviated herein as "dB/dbl"). It accounts for both geometric attenuation (spreading) and material attenuation (damping).

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[*] Again, the reader is reminded that in a "stiff" structure, there exists a multitude of closely-spaced modes (that is, the modal density around the fundamental resonance frequency is quite high). Therefore, the broader shape of the spectra does not represent only the effect of greater damping from the structure and foundation system.

- A "Rayleigh" model,* in which we assume both geometric spreading (amplitude varying as the square root of the distance) and material damping, with a damping coefficient a . This model can be interpreted as a special case of the "power" model, with g =-0.5. We can state the model as

$$M = M_2 L^{-0.5} \times e^{-aL} \tag{8}$$

Table 1 lists the coefficients obtained for least-squares fits of data at these two frequencies using the three models given above. The figures indicated in Table 1 show the measured data and the "best fit" propagation models, using only data for which coherence was greater than 0.7. The term R^2 indicates the quality of the fit, with 1 showing a perfect fit and 0 showing completely uncorrelated data. The most consistent fit appears to be with the "Power" model. The exponent, which is the slope on a log-log plot, is in the range -0.8 to -0.9. A general "rule" for this facility might be that vibrations at these frequencies drop off about 5 decibels per doubling of distance. However, it is important to note that 5 dB/doubling is not a "generic" behavior representative of propagation at all frequencies; one might find γ values as low as -0.2 or less, or as high as -2. Likewise, one might easily find cases in which the "Power" model did *not* provide the best fit (as demonstrated later in this paper).

Table 1. Propagation coefficients for stiff fab floor at two frequencies

Freq., Hz	"Linear"			"Power"			"Rayleigh"	
	dB/ft	dB/bay	R^2	γ	dB/dbl	R^2	α	R^2
30	-0.40	-4.8	0.88	-0.81	-4.8	0.91	0.021	0.82
48.5	-0.14	-1.6	0.60	-0.87	-5.2	0.90	0.0078	0.77

Figure 10 compares the propagation characteristics of the stiff and soft floors at the common rotating equipment frequencies of 30 and 60 Hz (shaft rotation rates of 1800 and 3600 rpm, respectively). The closed symbols represent mobility values for which the coherence was greater than 0.7; the open symbols represent other less-reliable values of interest, which are shown for reference only. The "best-fit" "power" curves are shown for comparison. Table 2 gives the coefficients.

Table 2. Comparison of propagation coefficients for stiff and soft floors

Freq., Hz	Type	"Power"		
		γ	dB/dbl	R^2
30	Stiff	-0.81	-4.8	0.91
	Soft	-0.41	-2.4	0.34
60	Stiff	-0.70	-4.2	0.92
	Soft	-0.87	-5.2	0.77

At both frequencies, the soft floor has a higher drive point mobility than the stiff one, but the difference is greater at 30 Hz. The slopes of the "fit" curves are about the same at 60 Hz, but at 30 Hz the slope is greater for the stiff floor. Both of these characteristics may be due to the same reason: the 30 Hz frequency is *greater than* the fundamental resonance frequency in the soft floor, and *less than* the fundamental in the stiff floor.

We should restate that these attenuation characteristics are unique to the particular fabs in which they were measured, even when one fab structure is replicated on several sites. Perhaps a *rough* estimate of attenuation effects can be assumed, but because of radical variations in attenuation from one structure to

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[*] We are not suggesting that this form of wave propagation involves Rayleigh or "surface" waves, merely that the model is similar to that commonly used to model the propagation of Rayleigh waves.

another, one should be careful not to assume that *any* measured propagation values can be used in a structure other than the one in which they were measured (or shared among different components of the same structure).

7. PROPAGATION FROM OTHER PARTS OF A FACILITY

It is quite common in modern fabs to have mechanical equipment in the subfab, the space beneath the fab. Typically this subfab equipment consists of process support equipment used as part of the production effort, such as vacuum pumps. There are two types of subfab arrangements currently in use: single-level subfabs and two-level subfabs. Generally in the latter case the upper level is referred to as the "clean" subfab (because it is usually in the "clean" air stream, acting as a return air plenum) and the bottom level serving as a "utility" space.

Figure 11 shows the data measured at three locations when a shaker was placed on the mat foundation forming the floor of a subfab space. Vibrations were measured near the shaker, at a fab midbay location above the shaker, and in the fab atop the nearest column. Except for dips at anti-resonances between 55 and 70 Hz, the coherence is good at all frequencies between 15 and 100 Hz. We see that there is very little attenuation of vibration (less than 10 decibels except at the anti-resonances), and even slight amplification at midbay[*] at frequencies between 35 and 50 Hz.

Figure 12 shows the transmission between midbay of a suspended "clean" subfab floor and midbay of the fab floor directly above. We see that the attenuation is 20 decibels or more at frequencies below the subfab floor fundamental resonance (about 29 Hz) and about 10 decibels at frequencies between 30 and 55 Hz.

Suppose we had a steel-framing and concrete-slab fan deck off to one side of a fab and at an elevation one level above fab level.[†] If we had fans operating at 922 rpm, how would the vibration associated with this shaft rotation speed be attenuated between the fan deck and the closest fab location? How would they be further attenuated by distance as one moved across the fab? These questions can be addressed through measurements such as those shown in Figure 13, where the shaker was placed on the fan deck and swept from 5 to 60 Hz, and measurements made at several distances on the fab floor. Figure 13(a) shows the drive point mobility and the transfer mobility measured in the fab in the first bay nearest the source. Figure 13(b) shows the change in level between the two locations, typically between 30 and 40 decibels. Figure 13(c) shows the change of the component at 15.375 Hz, the component corresponding to 922 rpm. At that frequency we see 47 decibels attenuation between the fan deck and the fab, and about 2.4 decibels per bay (12 ft bays) as we move across the fab away from the fan deck. Upon inspection, it is clear that the best fit to the data on the fab floor in this case is the "Linear" model.

A discussion we commonly have during the design phase is whether or not to include structural isolation breaks (SIBs) around the perimeter of the fab area. We recently had the opportunity to compare several facilities with and without SIBs. (There are structural differences other than the presence or absence of a SIB, but the major differences in performance, we believe, are due to the SIB.) Figure 14(a) shows drive point and transfer mobility across a SIB. Figure 14(b) shows drive point and transfer mobility between a support area and the fab in a facility where the SIB was considered, but eliminated during the design phase. The distances between drive point and receiver are similar for the cases illustrated in Figure 14(a) and Figure 14(b). Figure 14(c) shows the change in level (in decibels) for the measurements, plus the change in level from Figure 5. With our assumption that the SIB is the only significant structural

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[*] It should be noted that this situation (amplification from subfab to fab) is the exception, rather than the rule. Usually there is some degree of attenuation at all frequencies. In the two fabs in which we have observed amplification, however, we were able to alert the owners and hookup designers that vibration isolation of subfab equipment was even more critical than usual.

[†] In effect, there is a complete SIB between the fan deck and the fab floor, because all of the vibration energy is forced to go down to the foundations and propagate through the soil to the fab structure.

difference between the fabs, we see the presence of the SIB provides virtually no benefit at frequencies between 20 and 40 Hz, and only a marginal benefit at frequencies below 20 Hz. However, at frequencies greater than 40 Hz, the benefit provided by the SIB is quite dramatic, on the order of 40-50 decibels. Indeed, in the facility without a SIB, the attenuation at frequencies near 60 Hz (where there are many motorized sources) is less than 10 decibels.

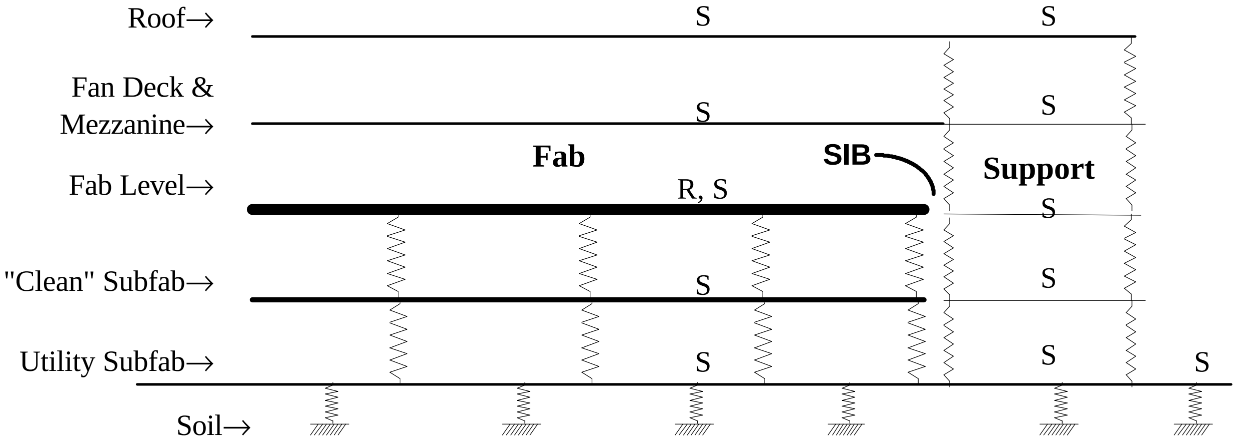
SUMMARY

A mathematical model has been defined which includes various components affecting the vibration performance of a facility. Experimental data obtained from several fabs using forced excitation were presented, based upon the test methods used by the authors. Data were presented which illustrate the effects of structural isolation breaks (SIBs), floor to floor propagation, building-to-building propagation, and other transfer functions. A study of this sort confirms the predicted structural stiffnesses and resonance frequencies by means of in situ measurements. The study can also characterize the manner in which vibrations are propagated through the structure.

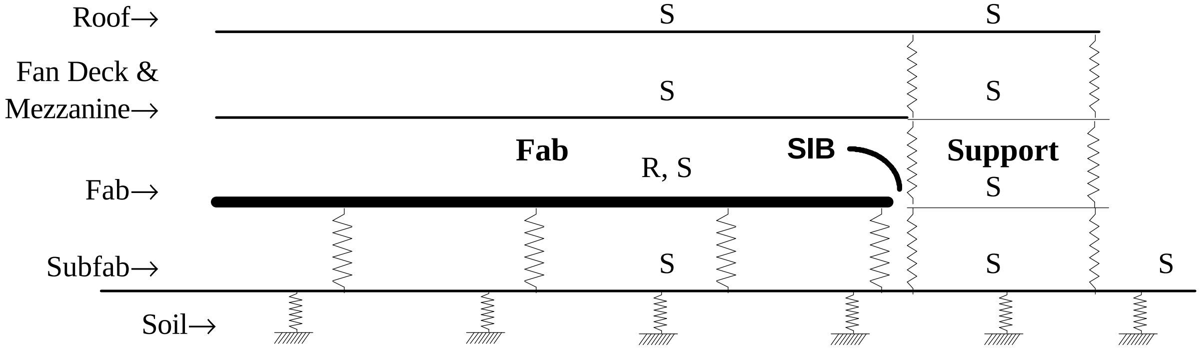
It is important to note that the purpose of a structural evaluation is to determine *in situ* properties. This paper presented data from several such evaluations, but the reader is cautioned that the specific results of those studies should not be taken as generally applicable to all facilities. Even structures that are nearly identical can yield different results because of variations in soil properties.

Figure 1: Study of structural properties using *in situ* measurements
Idealized Structural Configurations

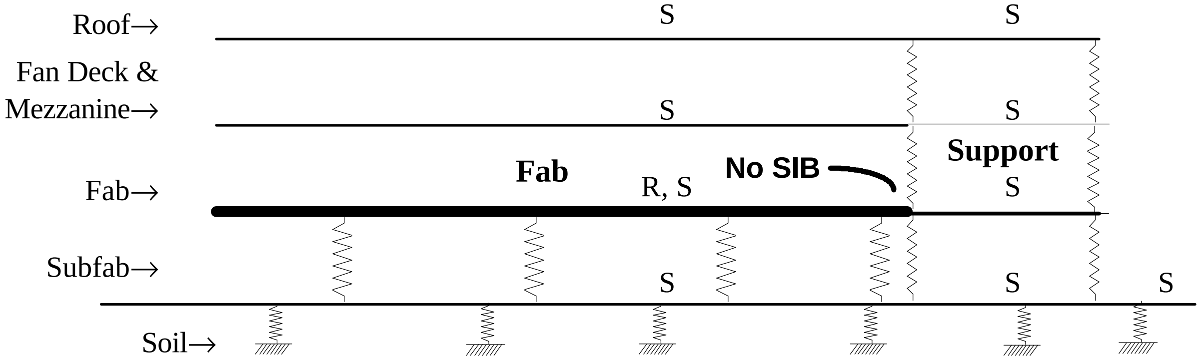
a) Two-level Subfab with SIB



b) Single-level Subfab with SIB



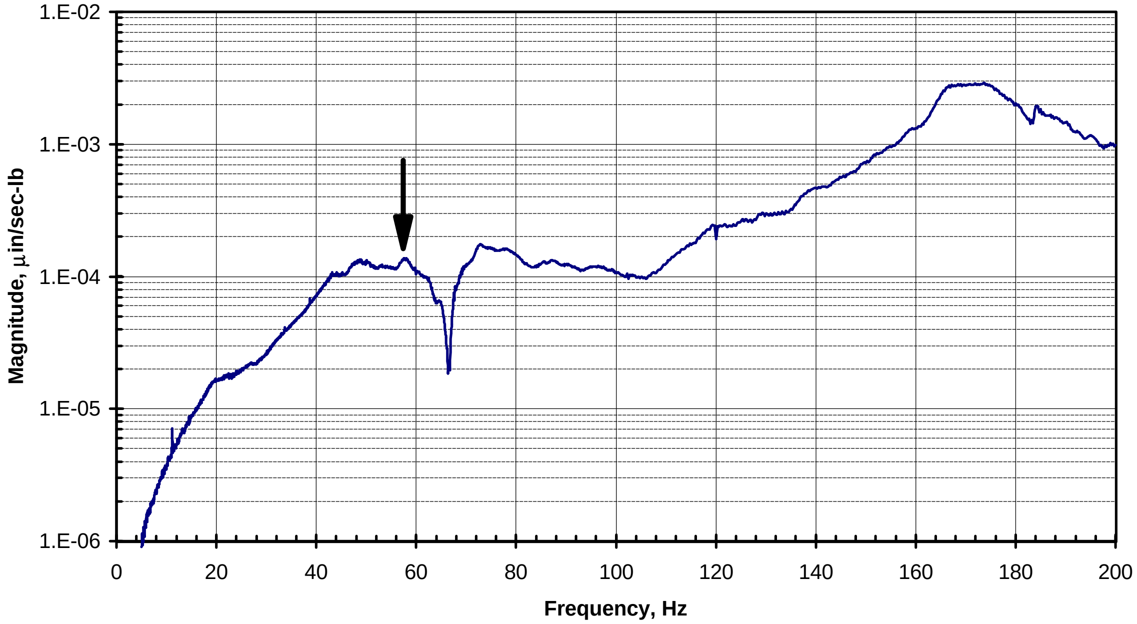
c) Single-level Subfab without SIB



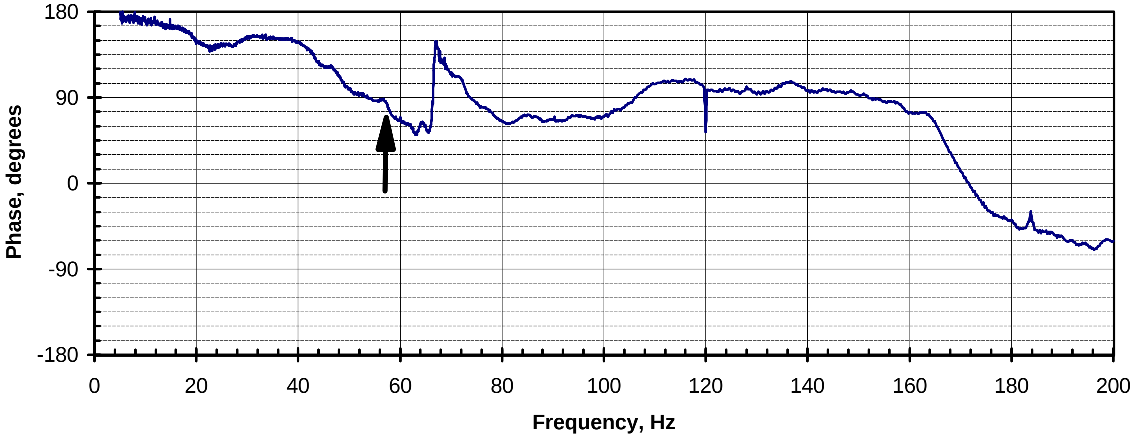
NOTE: "S" = potential vibration source(s); "R" = potential vibration receiver(s)

Figure 2: Study of structural properties using *in situ* measurements
Measured Data of Drive Point Properties

a) Accelerance Response Function



b) Accelerance Phase



c) Coherence

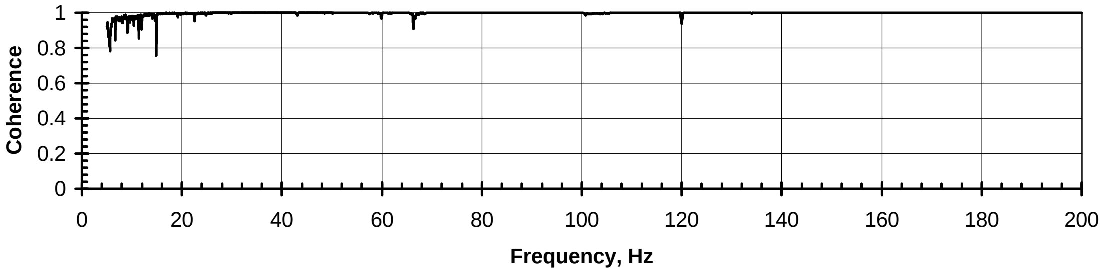
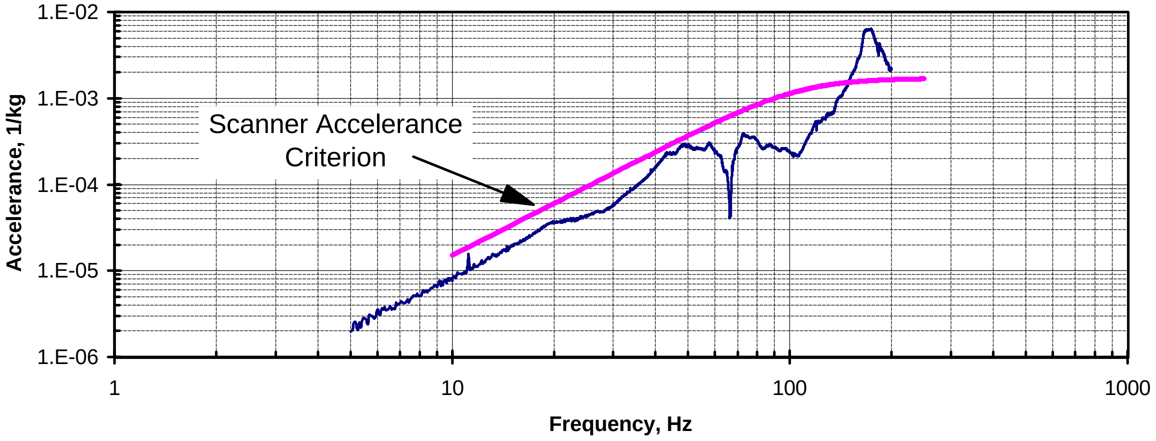
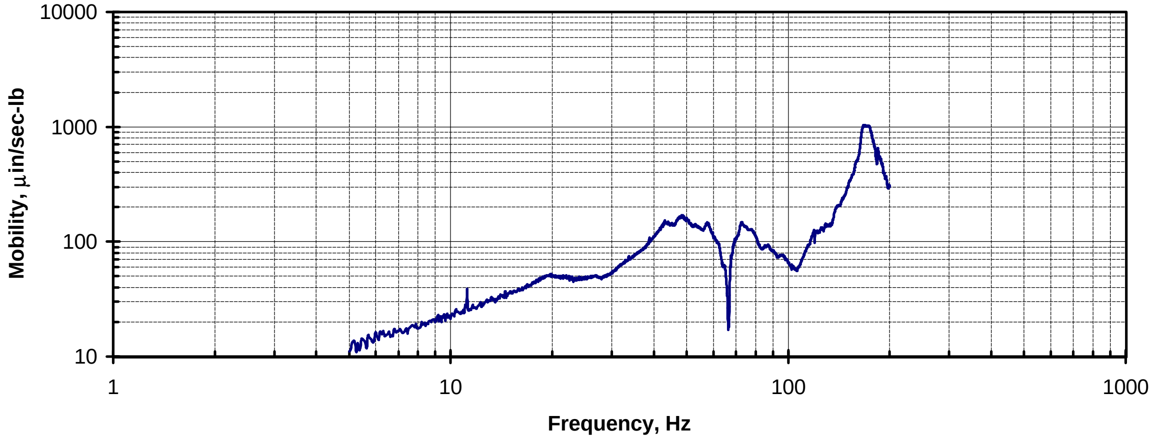


Figure 3: Study of structural properties using *in situ* measurements
Several Forms for Frequency Response Function Magnitude

a) Accelerance



b) Mobility



c) Receptance

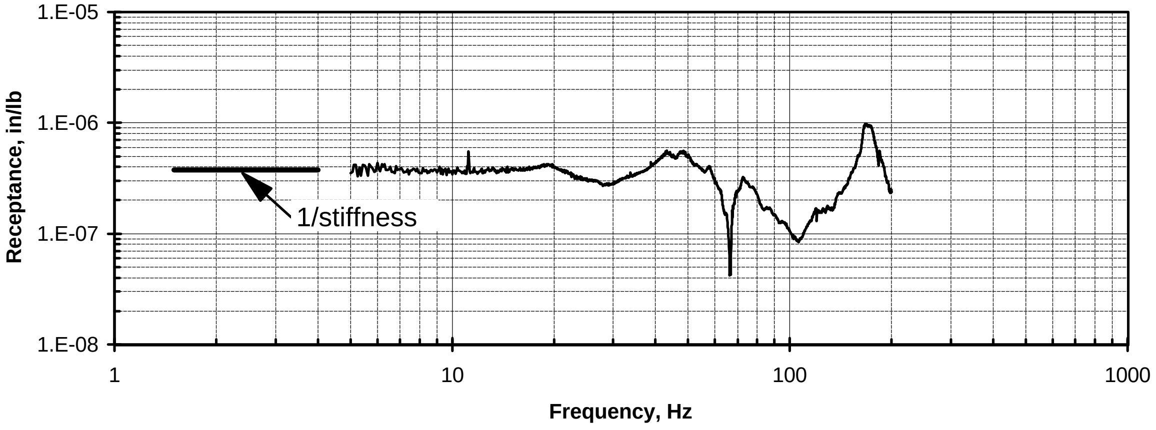
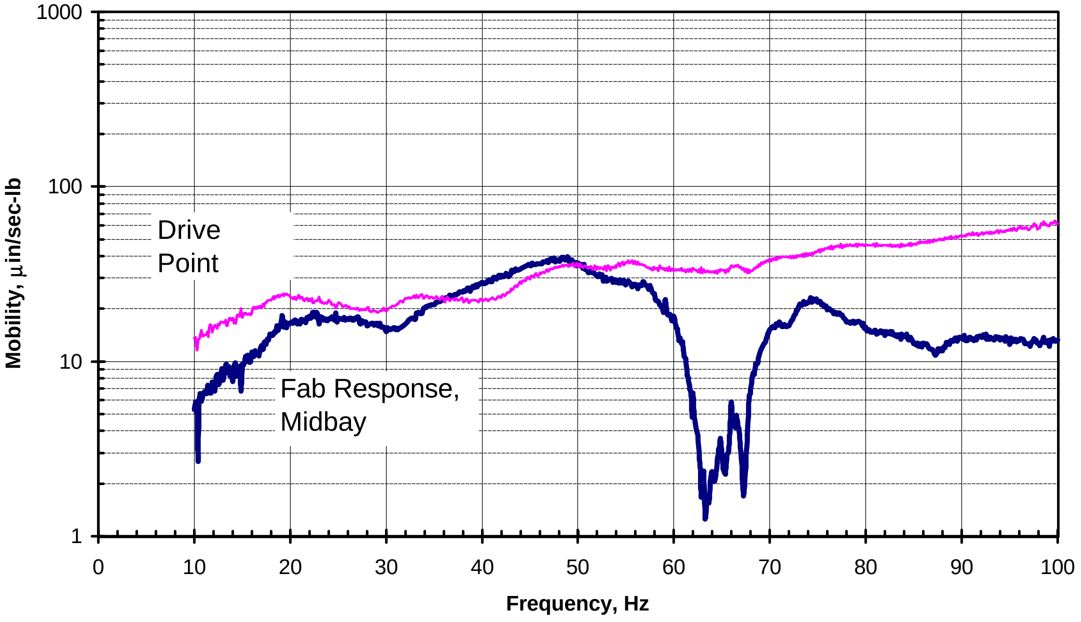
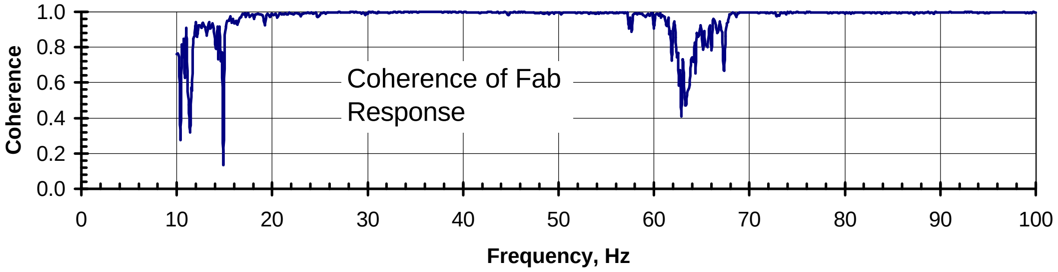


Figure 4: Study of structural properties using *in situ* measurements
Measured Data of Transfer Properties using Swept Sine

a) Mobility



b) Coherence



c) Change in Vibration Level

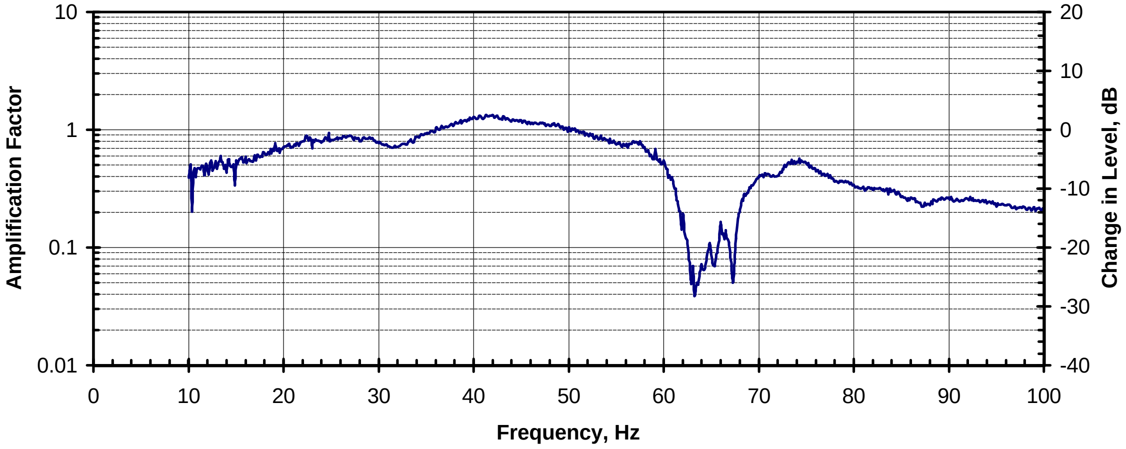
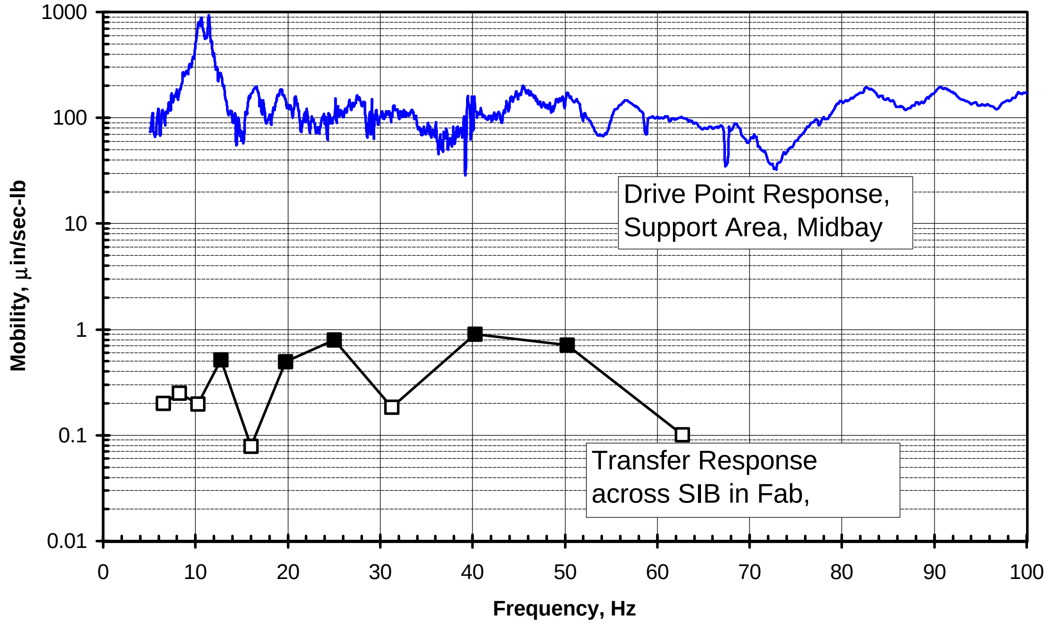
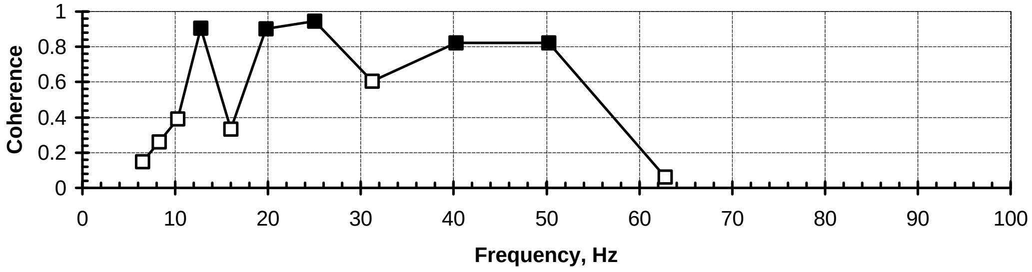


Figure 5: Study of structural properties using *in situ* measurements
Measured Data of Transfer Properties using Fixed Tones

a) Mobility on Two Sides of Structural Joint



b) Coherence of Transfer Response



c) Change in Vibration Level

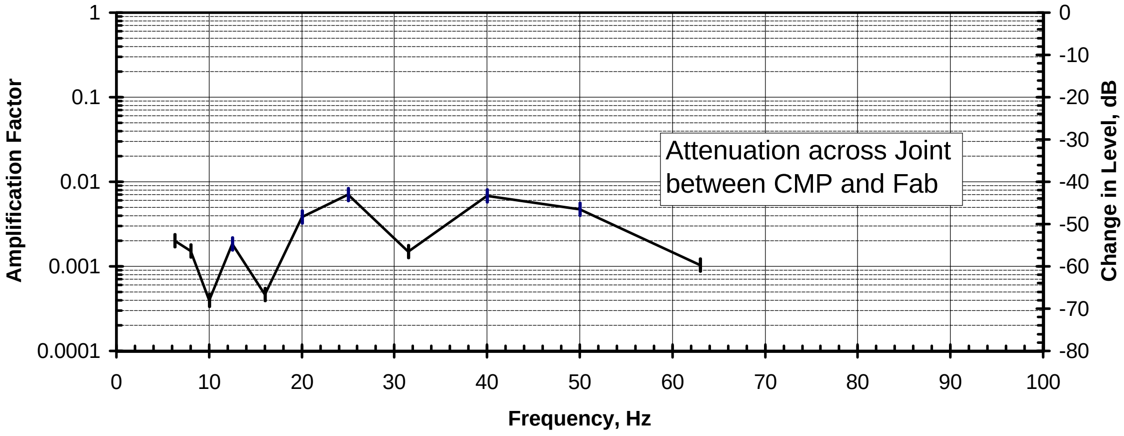
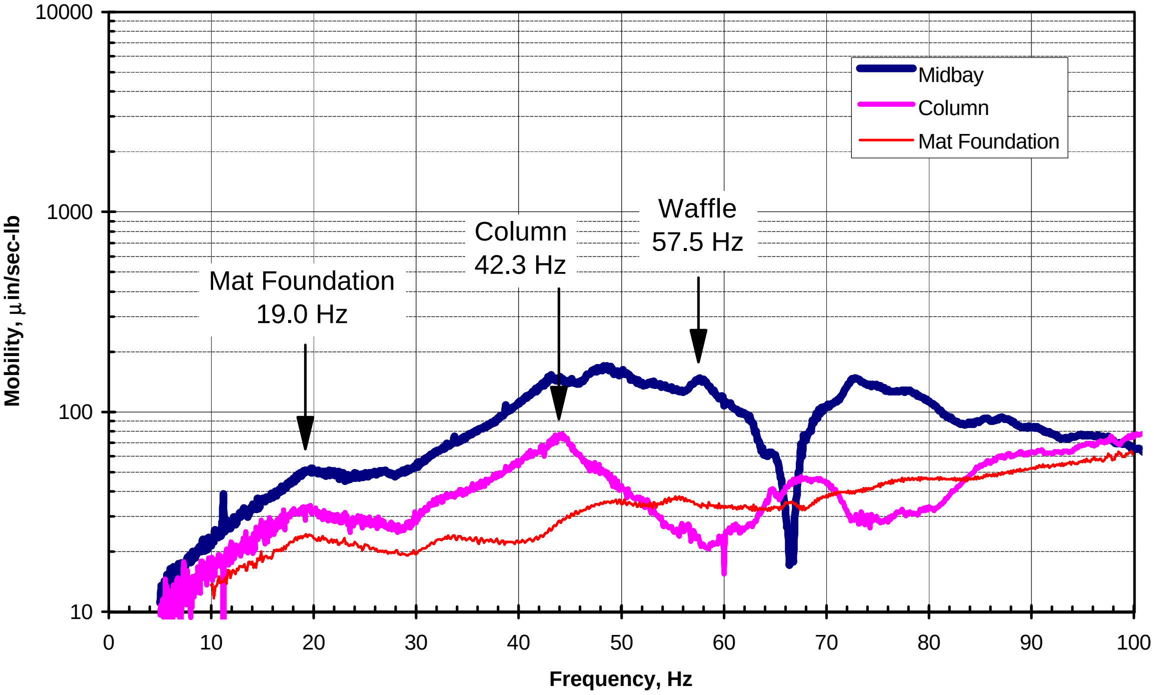


Figure 6: Study of structural properties using *in situ* measurements
Drive Point Properties - Midbay and Column

a) Stiff Floor



b) Soft Floor

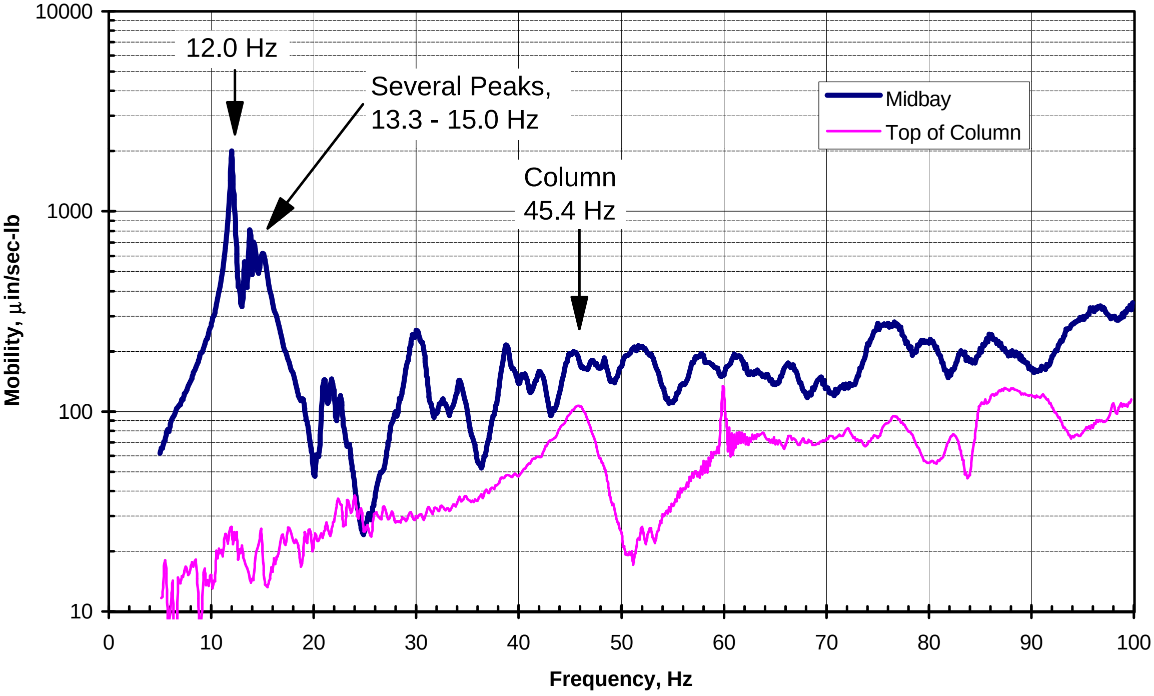
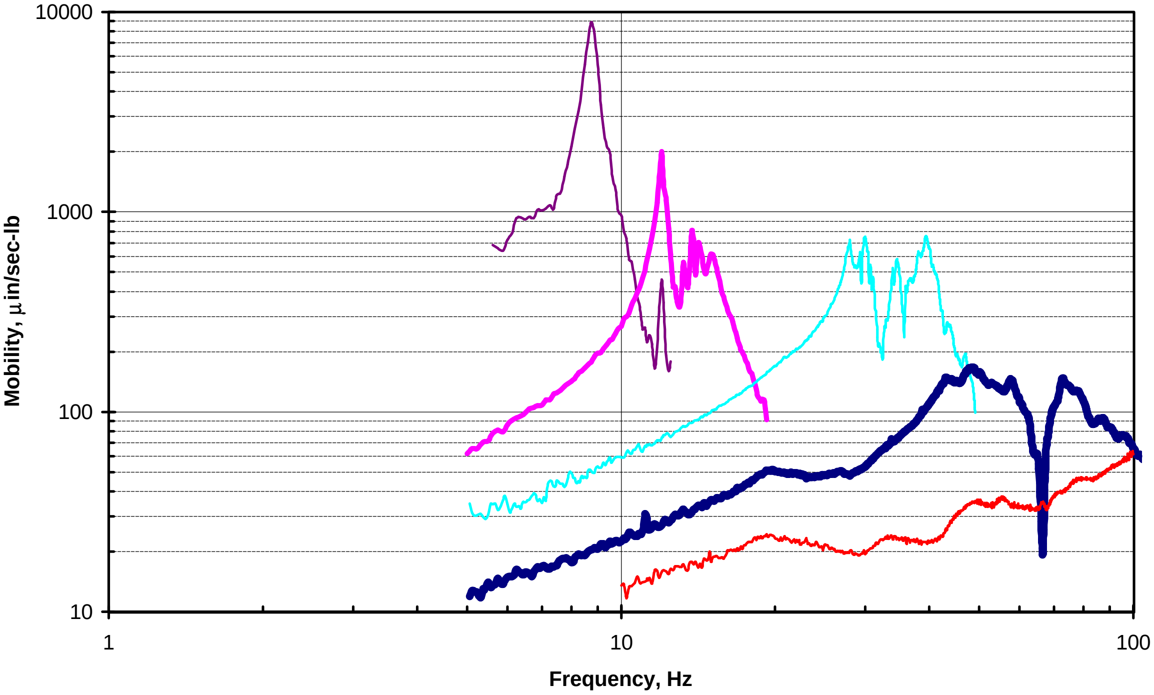


Figure 7: Study of structural properties using *in situ* measurements
Mobility and Receptance of Several Floors


a) Mobility of Several Floors of Varying Stiffness

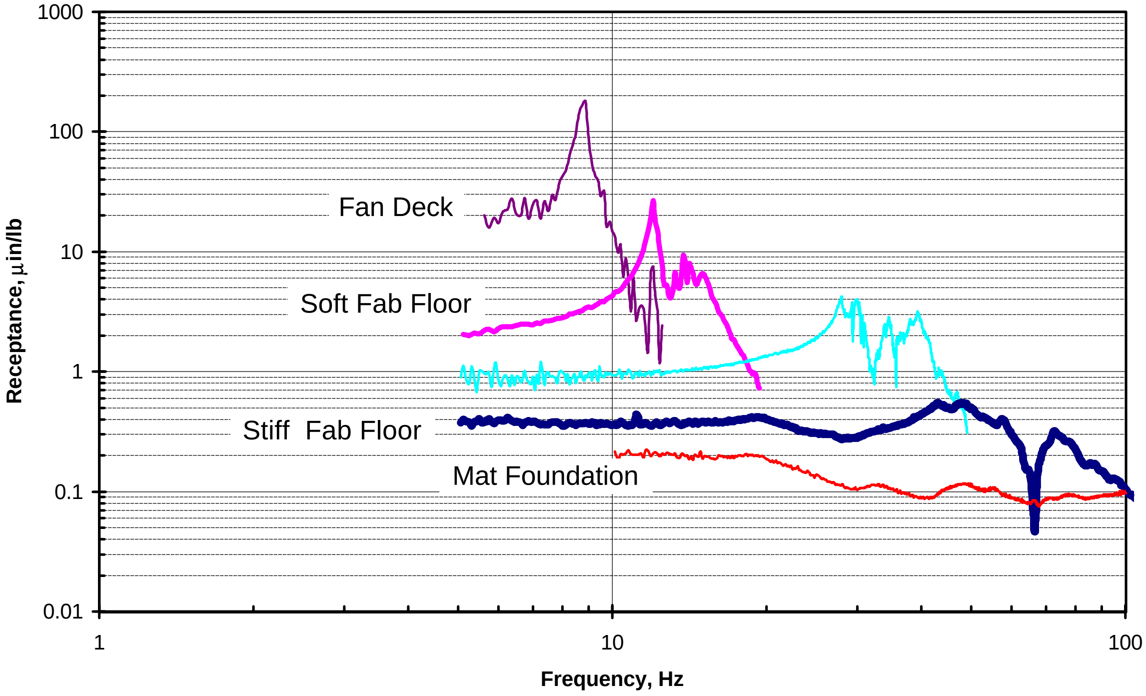
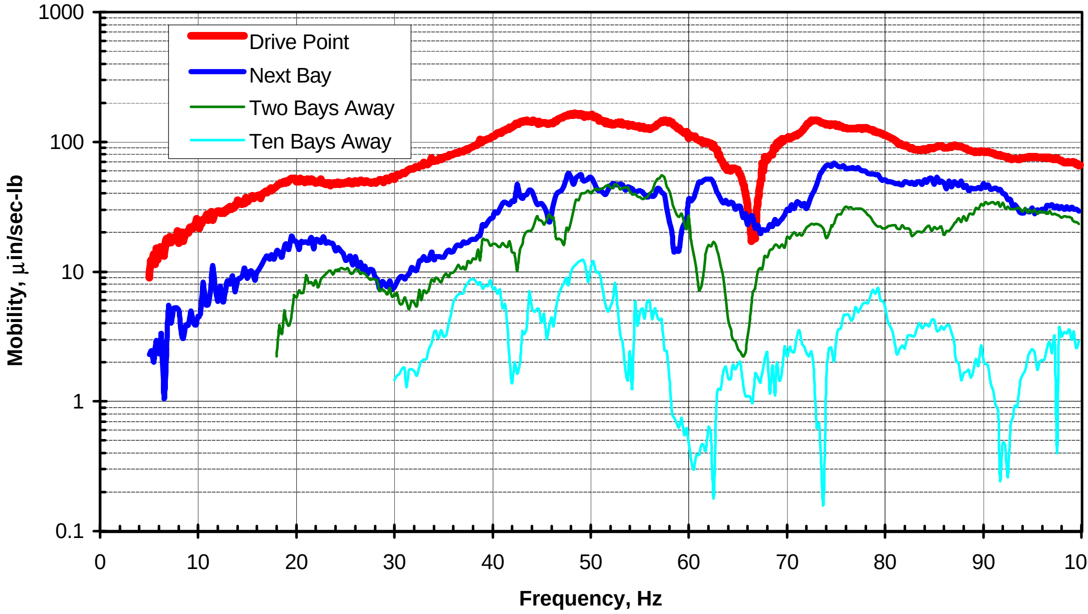
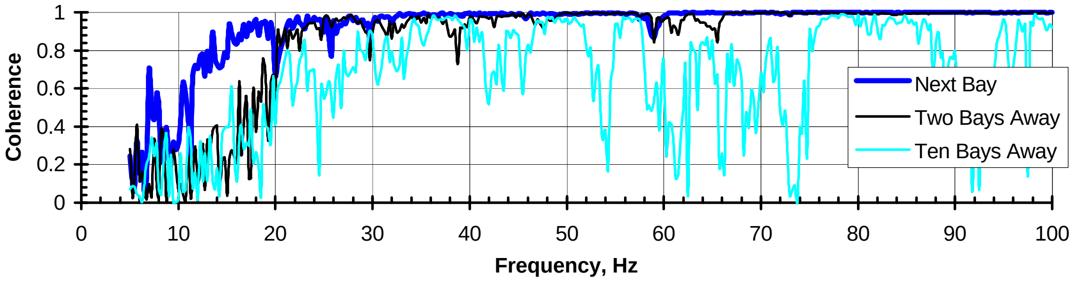

b) Receptance of Several Floors

Figure 8: Study of structural properties using *in situ* measurements
Mobility in Propagation through Several Bays - Stiff Floor

a) Mobility



b) Coherence



c) Change in Amplitude

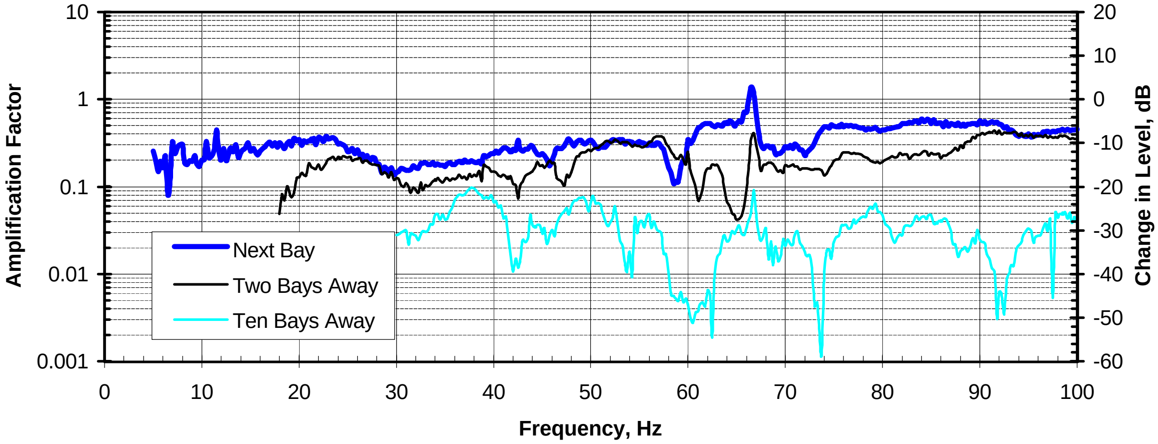
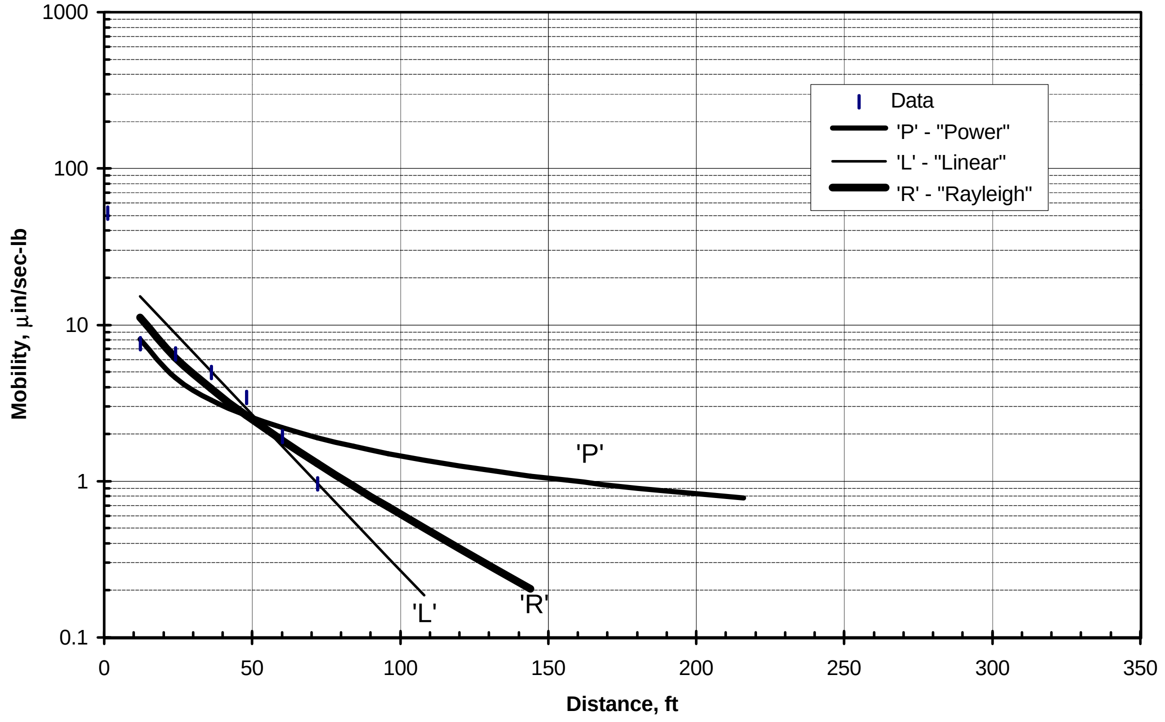
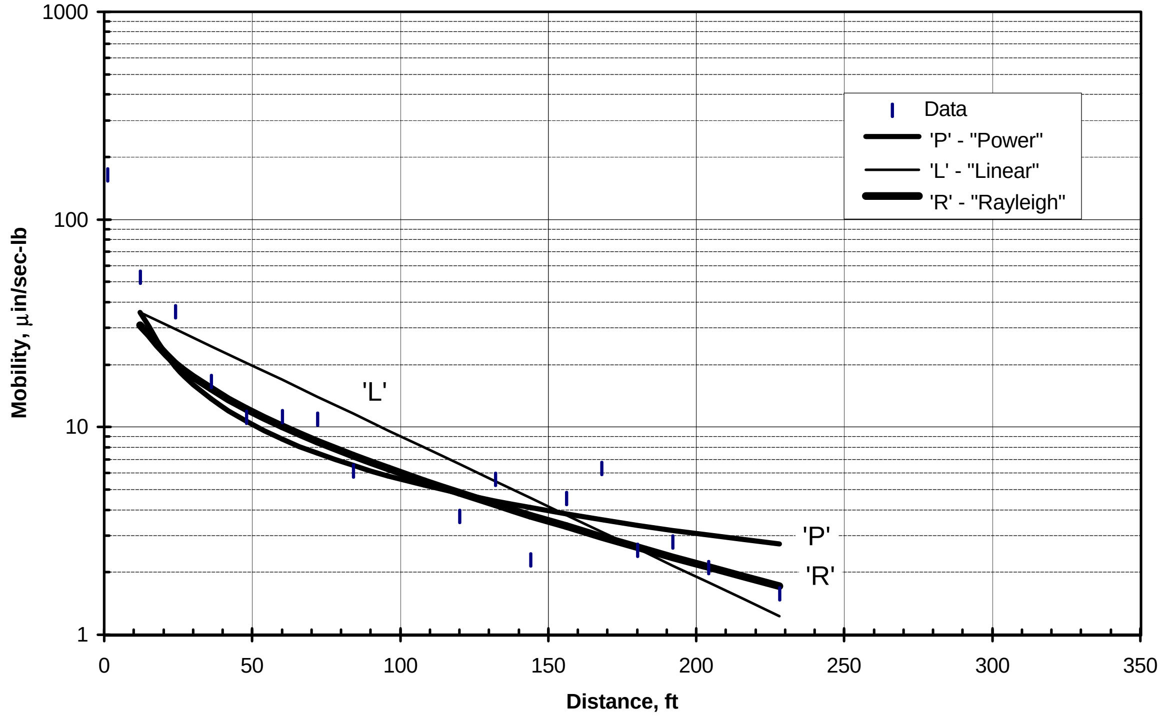


**Figure 9: Study of structural properties using *in situ* measurements
Fitting of Propagation Curves for Stiff Fab Floor**

a) 30.0 Hz



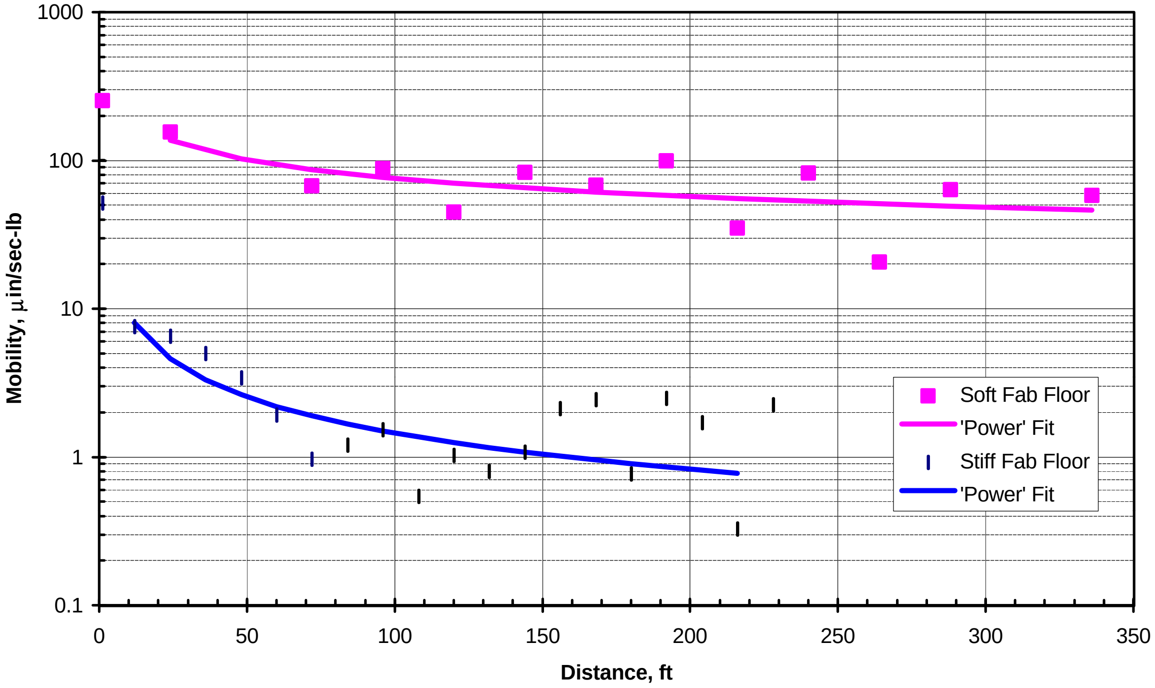
b) 48.5 Hz (Waffle Resonance)



b) 48.5 Hz

Figure 10: Study of structural properties using *in situ* measurements
Compare Vibration Propagation in Stiff and Soft Floors

a) 30.0 Hz



b) 60 Hz

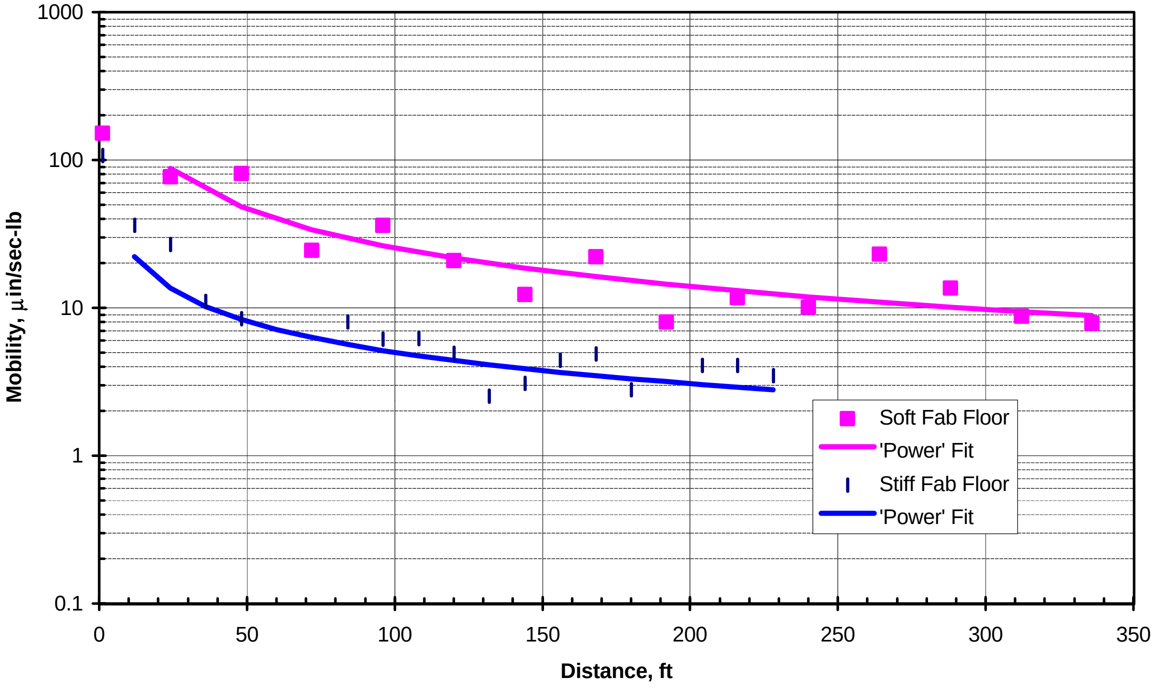
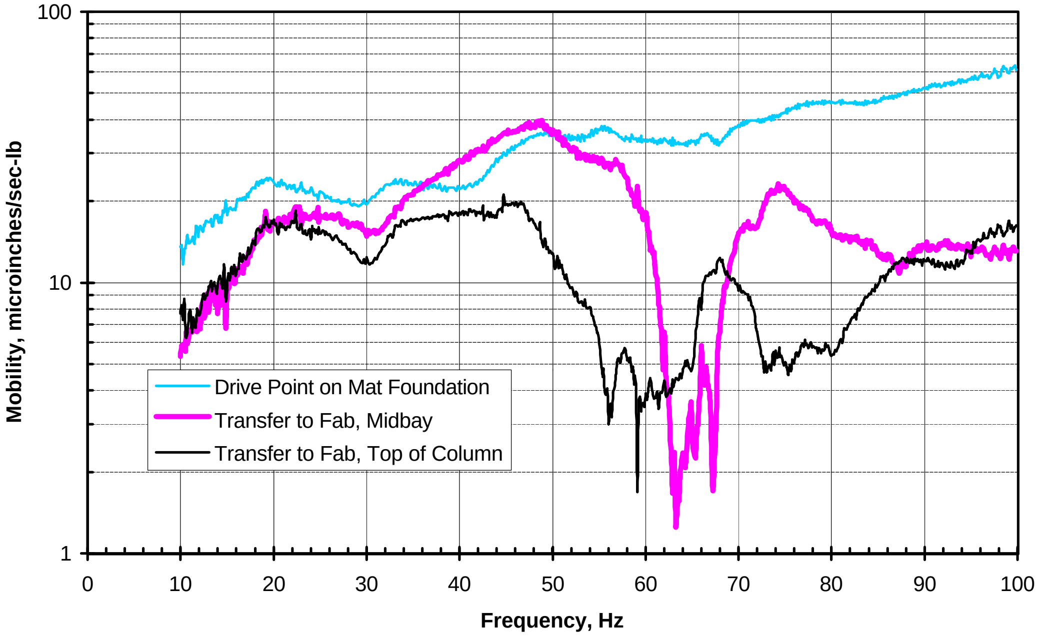
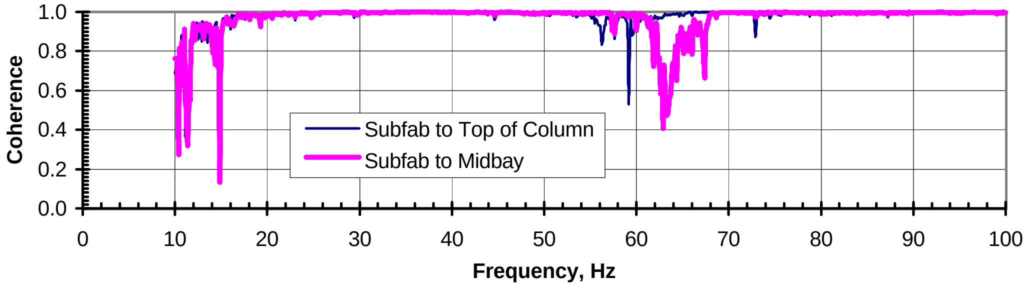


Figure 11: Study of structural properties using *in situ* measurements
Transmissibility from Grade Level Subfab to Fab Level

a) Measured Mobility



b) Coherence



c) Change in Vibrations from Subfab to Fab

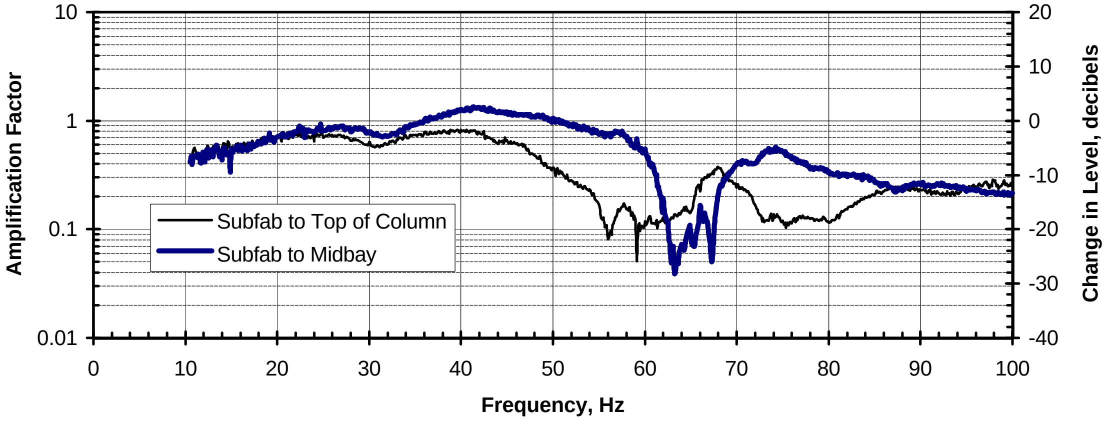
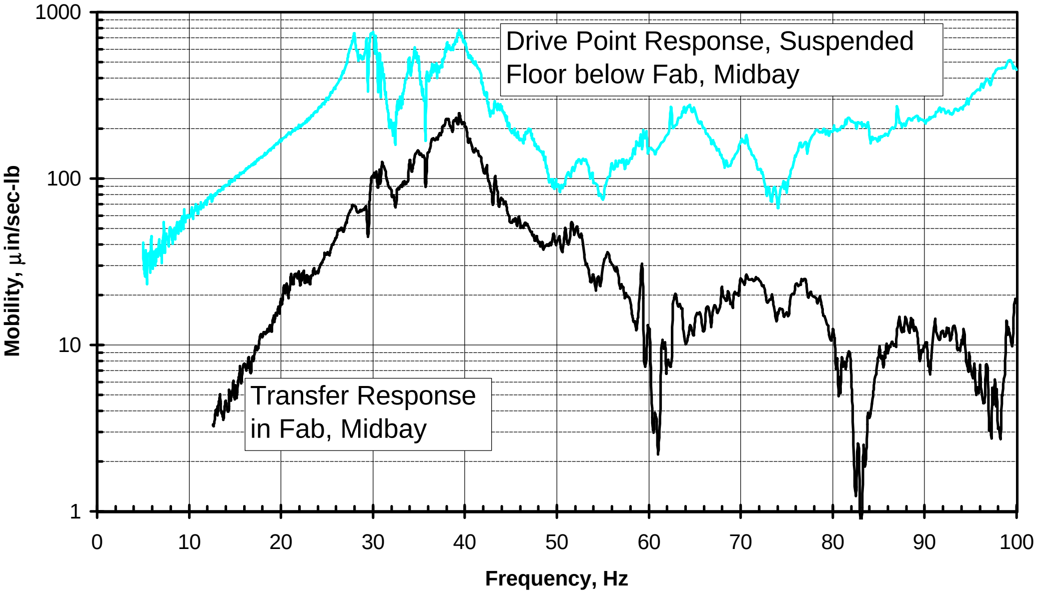
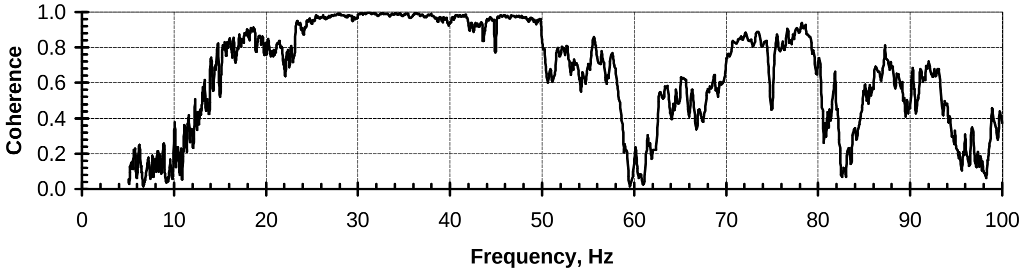


Figure 12: Study of structural properties using *in situ* measurements Transmissibility from Suspended "Clean" Subfab to Fab Level

a) Measured Mobility



b) Coherence of Transfer Response



c) Vibration Attenuation

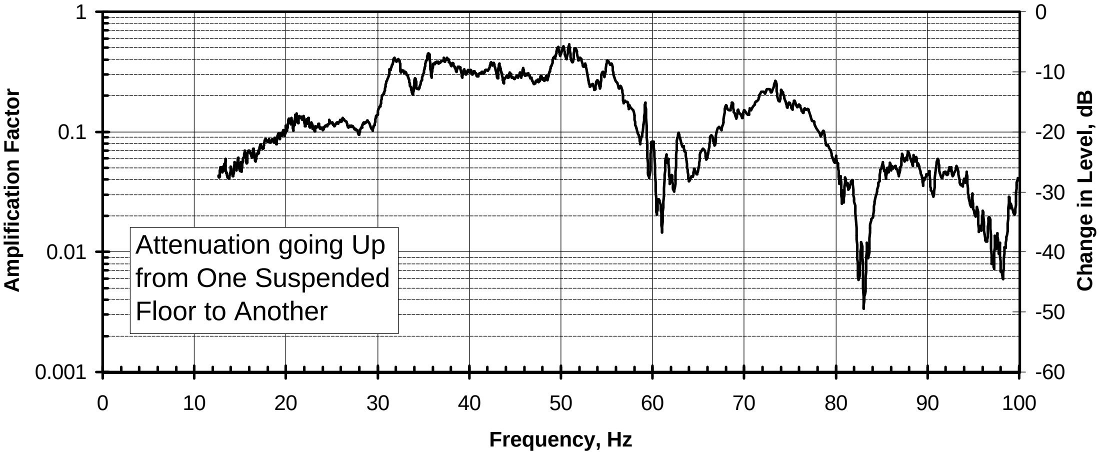
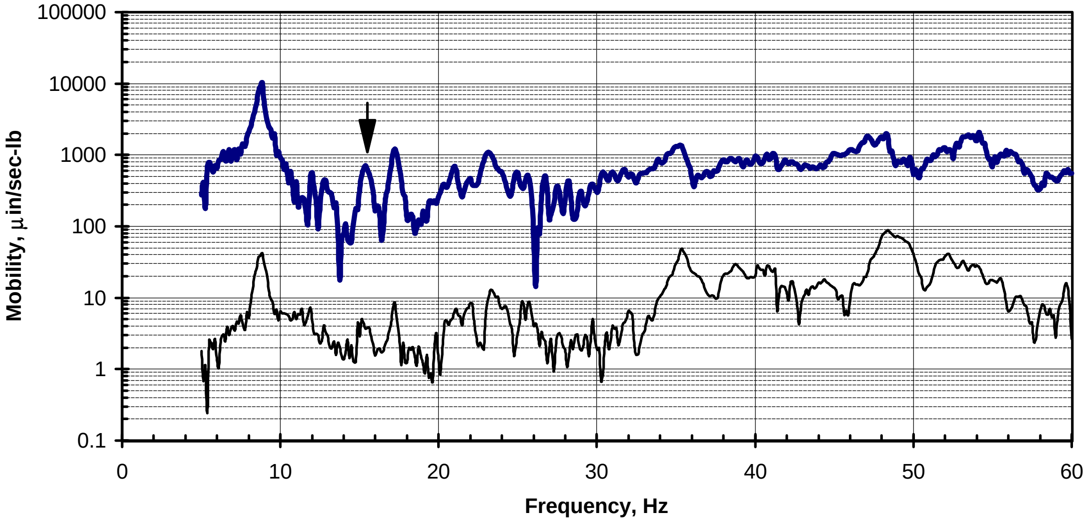
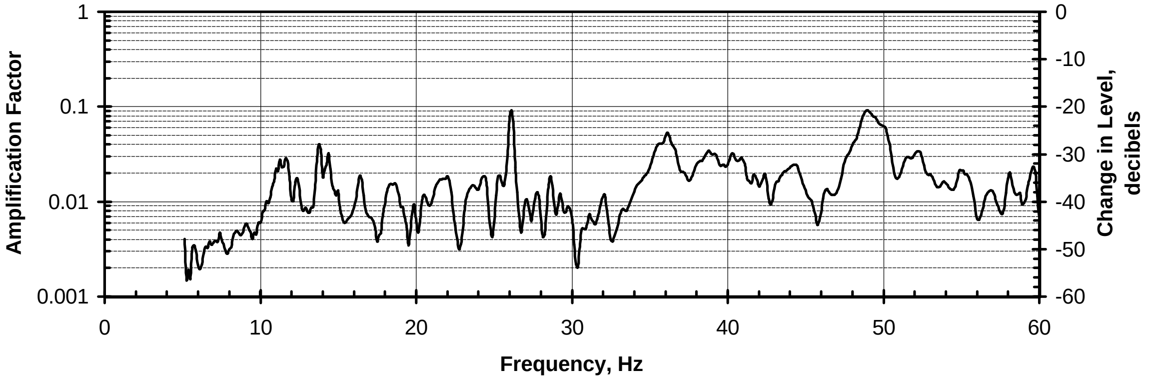


**Figure 13: Study of structural properties using *in situ* measurements
Transmissibility from Fan Deck to Fab Level**

a) Fan Deck and First Bay in Fab



b) Change in Vibrations, Fan Deck to First Bay in Fab



c) Fan Deck to and Across Fab at 15.375 Hz (922 rpm)

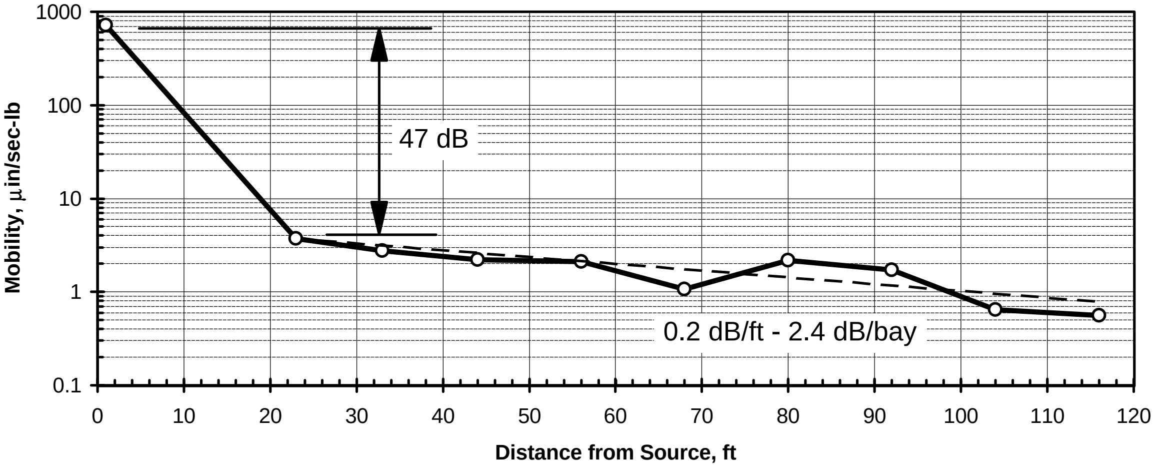
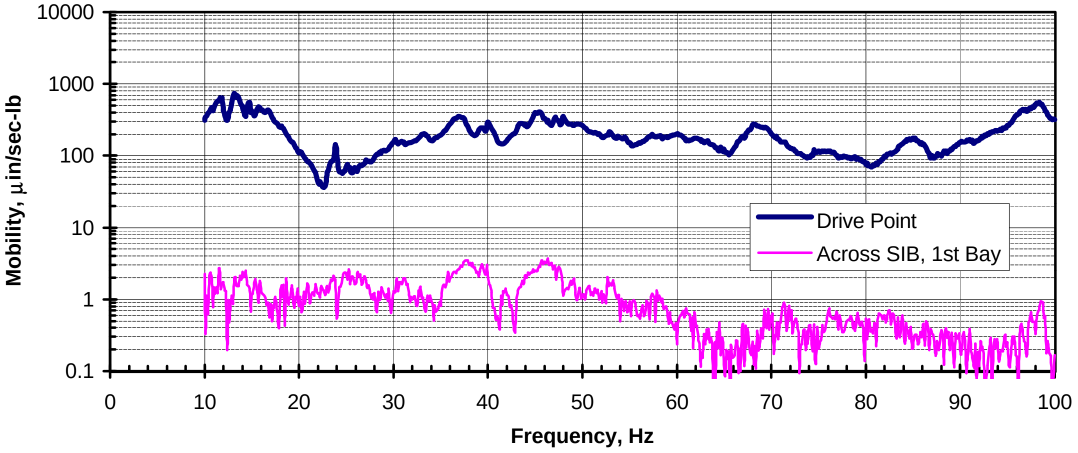
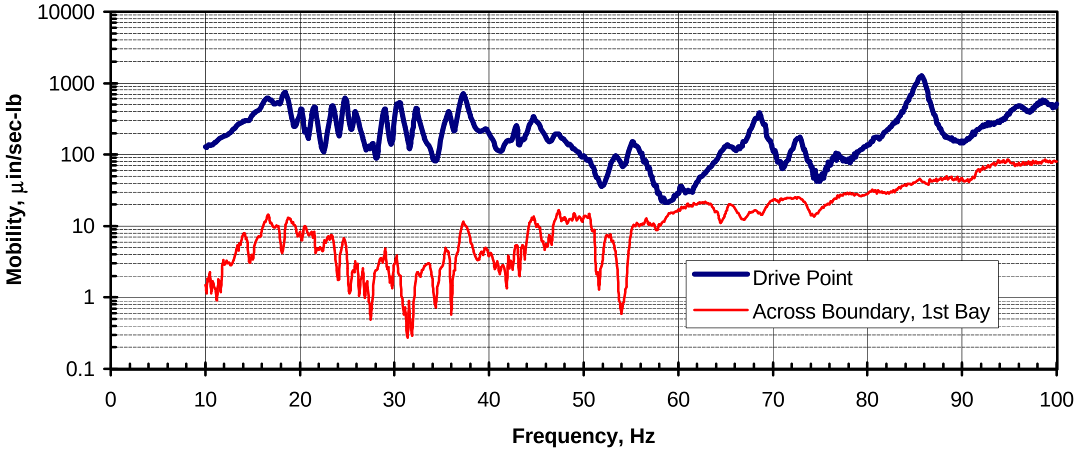


Figure 14: Study of structural properties using *in situ* measurements
Comparison of Transmissibility with and without SIB

a) Facility A with SIB



b) Facility B without SIB



c) Change in Vibration, with and without SIB