Vibration transfers to measure the performance of vibration isolated platforms on site using background noise excitation

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This article demonstrates a quick and easy way of quantifying the performance of a vibration-isolated platform. We measure the vibration transfer from floor to table using background noise excitation from the floor. As no excitation device is needed, our setup only requires two identical sensors (in our case, low noise accelerometers), a data acquisition system, and processing software. Background noise excitation from the floor has the additional advantage that any non-linearity in the suspension system relevant to the actual vibration amplitudes will be taken into account. Measurement time is typically a few minutes, depending on the amount of background noise. The (coherent) transfer of the vibrations in the floor to the platform, as well as the (non-coherent) acoustical noise pick-up by the platform are measured. Since we use calibrated sensors, the absolute value of the vibration levels is established and can be expressed in vibration criterion curves. Transfer measurements are shown and discussed for two pneumatic isolated optical tables, a spring suspension system, and a simple foam suspension system. © 2011 American Institute of Physics. [doi:10.1063/1.3602331]

I. INTRODUCTION AND BACKGROUND

As research and production focus on smaller dimensions, disturbances due to vibrations and acoustics play a significant role. Vibration levels can be limited by proper building design in combination with vibration isolated platforms. Similar actions can be taken to limit acoustic noise.

The growing need for vibration control manifested in the early 1980s in the vibration criterion (VC) curves (“BBN” curves at that time), by Ungar and Gordon.1 These curves define floor vibrations to meet the requirements of generic groups of equipment. VC levels are defined as rms speed values, integrated over one third octave bands. Figure 1 gives an overview of the VC-curves, including all modifications made at later dates and, for comparison, the ISO guidelines for people in buildings.2 The VC-A and -B curves are unchanged from their original definitions. Their definitions cover the frequency range from 4 to 80 Hz, showing a decrease in vibration amplitude from 4 to 8 Hz. In the 1980s, VC-E was added, mainly due to the growing demands of the semiconductor industry. The VC-C, -D, and -E curves were flattened in 2002,3 because pneumatic springs, used in many items of the equipment (and optical tables), show an amplification of low frequency vibrations. In 2005, the low frequency limit was lowered from 4 to 1 Hz (for the same reason) and the VC-F and -G curves were added.4 The latter two were used to accommodate the need for scientists and engineers to characterize extremely vibrationally quiet places.

In practice, manufacturers of extremely demanding equipment try to limit the VC-demand on the floor by implementing vibration isolation in their products. Other vibration criterions exist, such as the National Institute of Standards and Technology-A criterion.5 Manufacturers may use different units compared to VC curves to quantify vibration requirements for their equipment. For comparison, recalculation of vibration data may be necessary.

A well known way of measuring transfers is the use of a dual channel spectrum analyzer.11,12 The transfer follows directly from the division of the two channels. Excitation of the structure under investigation is done actively with an instrumented hammer or an electromagnetic or pneumatic shaker. Active excitation however has the disadvantage of measuring at higher amplitudes compared to regular levels.

In our case, characterizing vibration isolated platforms, measured at relatively high amplitudes does not take the effect of nonlinearity of the suspension system under normal working conditions into account. For this reason we propose using the background noise as excitation source. This also opens the opportunity to use coherent transfers to discriminate acoustic pick-up from floor vibrations, based on a difference in coherence. Acoustic pick-up is generally not specified by table manufacturers.

An additional advantage is the fact that an excitation device can be omitted. As a disadvantage, the need for low noise sensors and a somewhat longer measurement time must be mentioned.

There is a commonly held belief that the FFT of a stochastic signal does not exist so that random input signals such as ambient noise cannot be used for spectral analysis. One consequence is that the power spectral density has to be derived from the FFT of the autocorrelation function, rather than the FFT directly. This belief holds for pure random signal of infinite length, where there is no power for any frequency and only a power spectral density for any frequency range. Finite signals with a limited spectral content, however, do contain power at selected frequencies over the finite time period and can be analyzed using a FFT as the measure of the spectral content over that time period.

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Transfer functions, which contain the division of the FFT of two limited time series, can be derived correctly as long as both series are transformed using the same parameters; the series must be measured simultaneously, have the same filtering, and have the same record length to ensure coherence between the two series, independent of the coherence length of the two signals themselves. Conceptually, one can think of any finite record (realization) of a stochastic signal, as a deterministic signal which was purposely created in that way precisely. Mathematically, the FFT of the autocorrelation for a discrete finite signal is equivalent to the absolute value of the FFT, since convolution in one domain is equivalent to multiplication in the other. In this article we only use the FFT of two simultaneously measured traces and ensure equal treatment of both series so that the coherent and incoherent transfer functions can be derived correctly. The calculation of the power spectral density of the total ambient noise is based on the FFT directly. Due to the shape of the spectral content of the noise (sharply decreasing with frequency), the Hamming filter, and the large sampling array, this calculation of the power spectral density from the limited dataset is an accurate procedure which we verified by also comparing with calculations using the linear autocorrelation from a padded array and by averaging multiple slices.

Applications of ambient noise measurements have been reported for decades in the field of seismic behavior and civil engineering. Using sufficiently sensitive sensors, we show that the field of vibration isolated platforms can benefit from these techniques as well.

To understand the performance of vibration isolated platforms, the damped mass-spring model is presented here. The math behind this model can be found in many textbooks or at www.Wikipedia.org. The theoretical transmissibility curve versus frequency is presented in Fig. 2, for a range of values for the damping (Q). For frequencies smaller than the resonance frequency, the transmissibility is close to one, meaning that the isolated platform follows the movements of the floor. At the resonance frequency, the Q-factor determines the peak height. Above the resonance frequency, the Q-factor determines the slope of the transmissibility. For vibration isolation, Q values are a mixed blessing; low Q means limited peaking but a moderate suppression of vibration (about \(-20\) dB/decade) for frequencies higher than the resonance frequency. Higher Q shows higher suppression in the first decade above the resonance frequency (about \(-40\) dB/decade) at the expense of a higher peak at the resonance frequency. For the highest frequencies, the slope always converges to \(-20\) dB/decade, as can be seen in Fig. 2.

The low frequency resonance behavior, combined with the growing number of equipment applications using pneumatic springs, resulted in a flat VC curve starting as low as 1 Hz.

II. SENSOR

Motion sensors can measure acceleration, velocity, or displacement. Acceleration can be measured directly and independent of a reference frame as it is accompanied by a force. Measurement of velocity or motion requires a reference point. Since we are interested in the transfer of motion from the floor to a platform, one cannot serve as the reference for the other. Furthermore, as the frequency increases, the background noise amplitude decreases rapidly as most buildings can be seen as mass spring systems with very low resonant frequencies. Acceleration, as the second derivative of motion, is much more flat with respect to frequency so that measurement of acceleration is much easier over a larger frequency range. For low to medium frequencies, velocity sensors are best suited. For very low frequencies, displacement sensors give the highest accuracy.

The sensor requirements in our case are based on measuring low vibration levels in the frequency range of 1 to several hundreds of Hertz. Sensor size and weight are hardly a constraint because floors and isolated platforms are meant to support large weights. We choose to build a custom low noise
accelerometer based on a large mass and a low noise charge amplifier.

Our sensor is shown in Fig. 3 and consists of a brass rod of 1.4 kg that stably rests on three piezoelectric elements. This sensor is primarily sensitive in the vertical (z) direction. Each multilayer piezo\(^5\) is connected to a charge amplifier. The outputs of the three charge amplifiers are summed and amplified 10 times. This signal is fed to a data acquisition system. Our sensor shows sufficient signal-to-noise behavior for VC-G classification from \(~3\) Hz (due to \(1/f\) noise) up to several hundreds of Hz. Due to shielding, electro-magnetic interference (EMI) is negligible. This can easily be verified by monitoring the detector’s response in the upside down position; in which case, the sensitivity for vibrations goes down at least two orders of magnitude due to the absence of the 1.4 kg seismic mass.

Different dimensioning of the sensor is possible by selecting other piezo’s or a different mass. The sensor itself is also a mass-spring system; however, since the stiffness of the piezoelectric actuators is high, resonances in the frequency range of interest are easily avoided. In this case (17.4 \(\mu\)m displacement at 850 N), the resonance frequency is 1.6 kHz, well above the upper limit of the VC curves of 80 Hz.

Calibration was done against an accelerometer from Oceana Sensor,\(^6\) type TR1BCN, designed for low noise operation but, in practice, showing relevant \(1/f\) noise below 10 Hz in low level applications. The accuracy is specified within \(+/−5\%\), sufficient for the VC classification of floors where measurements of floor vibrations are repeatable only within 1 or 2 dB (12 or 26\%).\(^7\)

## III. MEASUREMENT SETUP

An overview of the measurement setup is shown in Fig. 4.

To measure the transfer of vibrations from the floor to the platform we use two accelerometers, one on the floor and one on the platform. The signal from the floor is regarded as the excitation for the transfer to the table. It can also be used to calculate the quality of the floor (the VC value).

The two accelerometers are simultaneously measured for a fixed number of points in time and low pass filtered at 500 Hz (4th order Butterworth), to prevent high frequencies from being undersampled. The filter outputs are connected to the inputs of a data acquisition card (NI USB-6211 OEM).\(^8\) Programming is done in LABVIEW.

Figure 5 gives a schematic overview of the processing steps used to extract the transfers, noise pick-up, and VC-curves. A Hamming window is applied first in all cases.

For the VC calculation, only the signal from the floor sensor is required. The single-sided auto-power spectrum is calculated, followed by a spectrum unit conversion to obtain the spectral density (acceleration) data. These are converted to velocity and displacement, through division by \(ω\) (\(ω = 2πf\)) and \(ω^2\), respectively. VC-values are obtained by integrating the velocity spectral density values over one third octave bands.

After the Hamming window, the signals from the floor and the platform are Fourier transformed using a FFT routine.
FIG. 6. (Color online) Results of averaging on vibration transfers of a foam suspended steel plate after multiple consecutively measured records, making use of excitation from the background floor noise. Acquisition rate = 2 kHz and record length = 4096 points. Below 3 Hz, the detectors 1/f noise becomes visible in the coherent transfer. Further analysis is presented in Fig. 7. (a) Vibration transfers after 1 record; (b) Vibration transfers after 10 records; (c) Vibration transfers after 100 records; and (d) Vibration transfers after 999 records.

The signal from the platform is expected to contain two components: the (coherent) transfer of motion from the excitation of the floor and (incoherent) the pick-up of acoustical noise from the surroundings. These two contributions can be separated by averaging the data from the response sensor in different ways.

The amplitude ratio follows directly from division of the moduli of the single sided spectra of the FFT of both channels.

FIG. 7. (Color online) Quantifying the influence of averaging on vibration transfers by calculating the standard deviation of the difference with the most accurate measurement (N = 999) for two frequency bands, 3–80 Hz and 80–300 Hz. Efficient averaging is obtained after 20 averages, except for the coherent transfer at frequencies with low coherence (<3 Hz and >80 Hz).

FIG. 8. (Color online) An aluminum plate on foam.
This way, the coherent as well as the non-coherent contributions are summed and cannot be discriminated.

To rule out non-coherent contributions, the phase relation between the two sensors must be taken into account. The excitations from the floor are spontaneous and thus have a random phase. To obtain the phase of the response with respect to the excitation, the phase of the excitation is subtracted from the phase of the response. We then average the complex spectra from a number of measurements. The incoherent part of the spectrum will average out to zero as long as there is no phase relation between the random acoustic pick-up and the transferred vibrations from the floor. The complex averaging thus results in a spectrum that only contains the (coherent) part of the excitations from the floor. The coherent transfer itself can now be calculated by dividing this averaged spectrum by the averaged spectrum from the floor.

For a decent signal-to-noise ratio, most measurements are conducted over several minutes.

One should be aware that vibrations occur in all directions. We are measuring only the (most obvious) z-component, and the total vibration values will be higher and more complex.

A second error source is sensor noise, mainly present in the low frequency region in the case of accelerometers, caused by 1/f current and voltage noise of the amplifiers. From the nature of this noise source, there is no coherence at all, and it will average out to zero over time. The sensor signal is however interpreted as some vibration level independent of the sign. Electronic noise will thus result in higher readings of the vibration sensor. One could argue that this can be subtracted from the FFT data of the stimulus signal, which is no longer complex after setting the phase of this signal to zero.

The effect of low frequency noise compensation will however always be limited because the noise reduces the sensors sensitivity, per definition.

Low frequency noise will deform an amplitude transfer to values close to one (or zero dB), which look plausible, but are, in fact, inaccurate.

In the case of a coherent transfer, low frequency noise will be averaged out, resulting in a much too low transfer value.

IV. AVERAGING IN PRACTICE

The effect of averaging was investigated by measuring the transfers of a foam suspended steel plate in a setup similar to Fig. 4. Our excitation source was a relatively quiet floor (VC-E) and acoustic disturbances were very low. Measurement settings: acquisition rate $= 2$ kHz, record length $= 4096$ points, total number of consecutive records: 1000. The resulting transfers are presented in Fig. 6 for the number of averages $N = 1, 10, 100, and 999$.

As averaging progresses, the 1/f detector noise (below 3 Hz) shows most clearly in the coherent transfer as a large negative value and a value close to zero in the amplitude ratio.

To quantify the effect of averaging, the standard deviation of the difference with the most accurate measurement ($N = 999$) is calculated for two frequency bands (Fig. 7), the 3–80 Hz band, which resembles the VC-criterion (with omission of the noisy 1–3 Hz part) and the 80–300 Hz band, where coherence is lower and vibrations become gradually less important due to their rapidly decreasing amplitude. From Fig. 7 we see that the frequency part from 3 to 80 Hz (relevant for vibrations) is efficiently averaged after some 20 records, which takes about 40 s of the measuring time.
As coherence decreases (<3 Hz and >80 Hz), the coherent transfer takes much longer (up to more than an order of magnitude) to converge.

V. VIBRATION MEASUREMENTS OF THREE TYPES OF ISOLATED PLATFORMS

Measurements were taken on three different platforms: A metal plate on foam, a metal plate suspended on elastic bands, and (two) optical tables on pneumatic isolators.

A. Metal plate on foam

A solid metal plate on a few pieces of foam as shown in Fig. 8 provides a simple and cheap vibration-isolated platform. The measurement results are given in Fig. 9.

From the transfer (Fig. 9(a)) we can conclude that the aluminum plate on foam behaves like a mass-spring system with a Q-factor of almost 10. Peaking occurs around 17 Hz and the total transfer approximates a $-40$ dB/decade slope for one decade. The floor is rated almost VC-E, a rather demanding criterion to achieve.\(^7\) The 1 to 3 Hz part of the measured VC curve is allowed to exceed the VC-E range because of low frequency detector noise.

Above $\sim 200$ Hz, there is an increased influence of acoustical disturbance. For comparison and making the acceleration data more intuitive, the rms noise level of the displacement is calculated for two (somewhat arbitrary) frequency ranges: 20–200 Hz: 23.58 nm (floor: 22.58 nm) and 50–200 Hz: 0.57 nm (floor: 7.82 nm). As the displacement is calculated from acceleration through division by $\omega^2$, low frequencies dominate the displacement amplitude.

B. Metal plate suspended by springs

A second low cost method of vibration isolation uses elastic springs. In Fig. 10, a relatively small aluminum platform is suspended by elastic springs. The measurements are shown in Fig. 11.

The spring suspended aluminum plate behaves like a mass-spring system, peaking at 4.4 Hz with a Q of $\sim 3$. Due to the relatively low resonance frequency and the acoustical damping, low total vibration levels are obtained at higher frequencies. The peak in the amplitude ratio indicates acoustic pick-up at 50 Hz, as EMI is ruled out.

Root-mean-square noise levels of the displacement are 20–200 Hz: 0.95 nm (floor: 20.11 nm) and 50–200 Hz: 0.25 nm (floor: 10.08 nm).
C. Optical tables isolated by pneumatic isolation mounts

Pneumatic isolated optical table tops are widely used. The low spring constant in combination with a large mass results in a low resonance frequency, typically 1 to 3 Hz. The damping as well as the table height can be controlled, which makes these kinds of tables very convenient.

A common way of characterizing optical table tops is the measurement of the static and dynamic rigidity. The latter is characterized by the compliance curve which is measured by striking the corner of a table with an impact testing hammer and measuring the response with an accelerometer. With pneumatic isolation mounts, one would expect the table surface to react as a mass-spring system, at least until the first resonance in the compliance plot.

We measured the transfers of a Newport technical series sealed hole laboratory table top, size 244 cm × 122 cm × 20 cm, isolated by 4 NRC pneumatic isolation mounts, type XL-G. Sensor positions: table sensor, approximately in the middle of the table and floor sensor underneath the table sensor. The results are presented in Fig. 12.

Moderate peaking occurs at 2.4 Hz, and at 20 Hz already a factor of 100 of transfer suppression is obtained. Above 20 Hz, instead of a decreasing amplitude ratio, a slight increase is observed. Also from 20 Hz, the acoustic pick-up starts to dominate the coherent transfer by some 10 dB.

As acoustic disturbances are unlikely to occur in the 20 Hz region, the main difference is expected in the use of pneumatic isolators. The pneumatic isolators are, in principle, susceptible to room air pressure fluctuations. Also, friction may be an element. Further investigation of the modeling of pneumatic isolators is required, but outside the scope of this article.

The rms noise levels of the displacement are: 20–200 Hz: 0.79 nm (floor: 25.42 nm) and 50–200 Hz: 0.66 nm (floor: 5.28 nm). Figure 12(b) shows that this measurement was performed on a very quiet floor, meeting the VC-E vibration classification. It should be noted that during the measurement, acoustical noise was at a relatively low level.

We also measured the transfers of a TMC 780 series table top, size 360 cm × 150 cm × 30 cm, isolated by 6 gimbal piston isolators, type 14-146-00. The table sensor was positioned approximately in the middle of the table and the floor sensor was positioned underneath the table sensor.

The results, obtained in a relatively quiet lab (acoustically), are presented in Fig. 13.

The TMC table shows a close resemblance to the smaller NRC table, depicted in Fig. 12. Due to the higher mass of the TMC table, its resonance frequency is somewhat lower (1.9 Hz), but the overall transfer is similar. The floor is rated VC-D and the remaining rms noise levels of the displacement are 20–200 Hz: 0.62 nm (floor: 44.44 nm) and 50–200 Hz: 0.47 nm (floor: 4.80 nm).

VI. CONCLUSIONS

Characterization of vibration isolated platforms can be done quickly and easily on-site by measuring the vibration transfer from floor to platform, making use of background noise excitation from the floor.

Measuring at realistic amplitudes takes possible non-linearities of the suspension system into account and enables to discriminate non-coherent (acoustic) pick-up.

By using calibrated sensors, absolute measurement values allow for floor characterization. VC-curves were calculated from the acceleration spectral density data of the floor sensor.

Measurements were performed on three different suspension systems, all in the vertical direction. The foam- and the spring-suspension systems behave like a mass-spring system. The pneumatic isolated optical tables deviate from such a system above 20 Hz. Its low resonance frequencies result in superior low frequency behavior at the expense of increased noise levels above 20 Hz.

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