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THE EFFECT OF VARYING ACOUSTIC PRESSURE ON VIBRATION ISOLATION PLATFORMS SUPPORTED ON AIR SPRINGS

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Air spring isolation systems are commonly used to support sources of vibration, or equipment sensitive to vibration, in order to reduce vibration transmission from source to receiver. In the latter case, commercial isolation tables or custom-designed "plinths" may use isolation systems to provide reduced vibration environments at the base of sensitive equipment supported on them. For passive isolation systems this occurs in some range above the resonance frequency of the isolator but below the compliance modes of the table or plinth. The achievable vibration environment is generally determinant given the existing ambient environment on the supporting structure plus some attenuation factor due to the isolation performance of the table or plinth, as a function of frequency. However, empirical data presented in this study show that the sound pressure acting on the isolation system also affects the achievable vibration environment for passive systems. The mechanism of this impact is discussed, as well as implied limitations in isolation table performance as a function of ambient acoustic pressure.

1. Introduction

A "plinth," as referenced in this paper and elsewhere, is a device, usually custom built, including a concrete mass supported on elastic vibration isolation systems. This is designed to provide a stable work surface that has relatively low vibration amplitudes with respect to the installation environment. An isolation plinth may be required, for example, for improved operation of advanced technology research instrumentation or processes. Photographs of two typical plinth designs are shown in Fig. 1. The concepts discussed shall apply in addition to more conventional systems, such as commercial isolation systems, optical tables, etc. A plinth (or optical table) may be supported on various types of isolators: active, passive, or combinations of the two. As plinth isolation systems tend to include massive concrete blocks for various reasons,[1] these are most commonly supported on passive isolation hardware (air or steel springs, or elastomeric isolators) since active isolation systems are, to our knowledge, not commercially available for use with very high loads. This paper therefore focuses primarily on passive isolation systems, especially as used in physics and nanotechnology research and instrument design.

Very briefly, in the design of a passive plinth isolation system, the goal is to provide a very low-vibration environment in the mid- to upper-frequency range (say, from 10 to 100 Hz) for sensitive instrumentation. This is done with the understanding that there will be limited degradation of the vibration environment at low frequencies associated with the isolation system resonance, and improvement in the environment above these frequencies, at least until reaching the compliance frequency range of the plinth (or optical table) or isolation systems. The low frequency degradation depends on the isolation hardware used: air springs typically have a resonance in the 1 to 3 Hz range, steel springs can have a resonance in the same range or higher (depending on the static deflection), and elastomerics may have a broader resonance (due to higher damping) in the 10 to 30 Hz range, again dependant on static deflection. As regards compliance of the plinth, table, or isolator structure, these are modal frequencies at which vibration is amplified. For example in the design of a custom concrete plinth, it is usually preferred that the first unrestrained whole body modes be above 100 Hz, unless lower frequencies are allowable given the known sensitivity of the equipment to be supported by the plinth.

Figure 1. Vibration isolation plinths (concrete blocks supported on air springs). From the top: (a) two identical plinths with steel top plates; (b) two plinths supported below floor level and accessed by raised flooring (partially removed); and (c) a typical air spring for the plinths shown in (b).

Air springs are, perhaps, most commonly specified for use in plinth or optical table isolation systems for various reasons, including low resonance frequency (equating to higher theoretical isolation efficiencies at a given forced frequency), higher damping compared to steel springs of similar frequency, and a lack of “surge” (longitudinal) frequencies found in steel spring systems. These are all significant advantages, but there are disadvantages that should also be considered. The obvious one is a need for a constant air supply; another, the possible degradation in performance due to the ambient acoustical environment, is discussed in this paper. Note, however, that it is not our intent to disparage plinth or air spring isolation systems in any way—for they have proved to be extremely useful and beneficial in many cases—but to provide information on an effect to be considered in designs optimizing vibration performance.

A separate consideration in the design of facilities for advanced technology research is the acoustic pressure in the laboratory. Air flow is usually necessary, as may be needed for control of air quality, temperature and humidity, cleanliness, etc., and these requirements may have to be balanced against those for noise control, both in the audio (20 Hz to 20 KHz) and infrasonic ranges (below 100 Hz). The noise requirements are usually considered as an independent variable to the vibration requirements, since vibration-sensitive research is more or less sensitive to acoustic noise, and this also varies as a function of frequency. Thus, there are often laboratory designs for very low environmental vibration, but which also require a high degree of cleanliness (say, ISO Class 6 or more stringent), with its attendant high acoustic noise level due to the great quantity of filtered air required. This paper will demonstrate that the vibration performance of a facility may be related to the acoustic noise performance, in certain cases.

2. The influence of acoustic noise on resiliently-supported structures

The performance of plinths supported on air springs may be degraded by the acoustical environment in which they are situated. These systems are excited at the resonance frequency of the isolation system by acoustic pressure due to air handling systems, door motion, etc. Furthermore, our research shows that there is a large amount of acoustic pressure at the relatively low frequencies corresponding to these resonances in typical rooms served by forced air systems, doors, etc.

Fig. 2 shows the noise level in a room containing a concrete plinth on air springs (that shown in the lower portion of Fig. 1) with the air handling system serving the room at three different settings: 100% normal operation (in terms of flow rate), 50% normal operation, and 0% (powered off). This particular study only measured noise in the audio range (31.5 to 8000 Hz), but it is typical to have broadband increases in sound pressure at lower frequencies as well. At the same time that the pressure measurements were made, the vibration velocity on the structural floor and on the isolated plinth was determined. These data are summarized, as peak response at the frequency of highest amplitude, in Fig. 3. Several measurements were taken at each location and each fan setting. The average trend lines clearly show a relationship between overall sound pressure in the room and the vibration velocity at the isolation system resonance frequency (3 Hz) on the isolation plinth.

It is also important to note that the vibration amplitude on the structural floor is not substantially affected by the acoustic noise level. This is evidence that (1) the isolation plinth is unusually susceptible to pressure stimulation; and (2) the vibration measurement system itself is not significantly affected by acoustic pressure, which would be a source of measurement error. The vibration on the plinth, as a function of sound pressure level in the room, is disproportionate to the vibration on the floor that supports it.

The next set of figures (Fig. 4 through 8) show in more detail the acoustic pressure and response on two isolation systems in the same room, a concrete plinth supported on air springs (shown in the upper portion of Fig. 1) and a optical table supported on neoprene pads. Fig. 4 shows the one-third-octave band sound pressure in the room with the air handling systems that serve it powered on and off, and with a pressure impulse that occurs when the door to the room is opened and closed. The latter generates a significant infrasonic impulse below 4 Hz. The are also differences in the room sound pressure level with the air handling system operating and not, especially below 100 Hz.

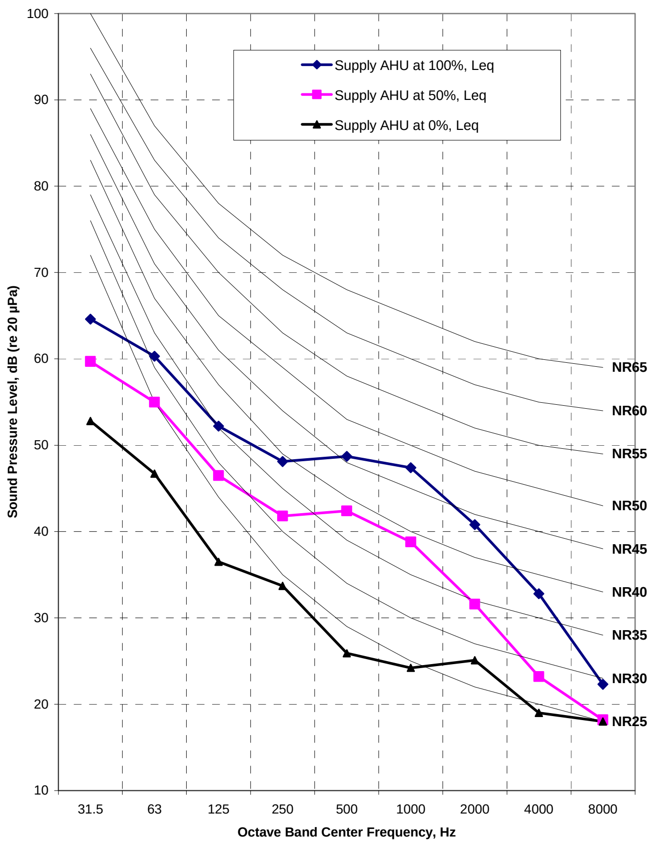

Figure 2. Equivalent-energy average (L_{eq}) sound pressure level in room with variation in supply air handling unit (AHU) operating point.

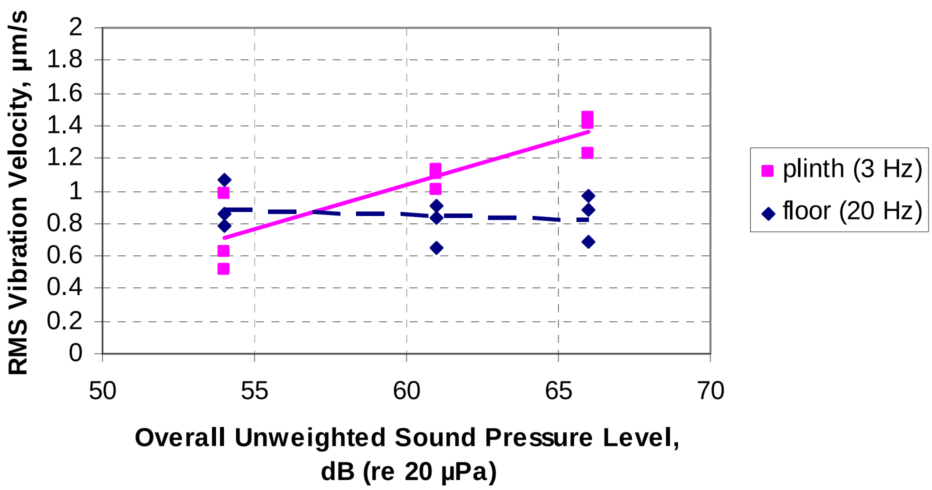

Figure 3. Vibration amplitude on floor and plinth versus overall sound pressure level (pressure variation shown in Fig. 2).

The vibration impact to the plinth supported on air springs (and to the structural floor upon which the plinth is located), due to the acoustic pressure sources represented in Fig. 4, is shown in Fig. 5 and 6. Both of these sources cause vibration and noise impact to the room. Our particular focus separates vibration impact, which if occurring on the floor would be seen in appropriate proportion on the isolation plinth, and acoustic impact, which, as discussed above, is seen disproportionately on the isolated plinth. In Fig. 5 it is clear that there is exceptional impact to the plinth at frequencies below 3 Hz—in the air spring isolator resonance range—that is not seen on the structural floor. There is a similar result shown in Fig. 6 due to operation of the local air handler; the effect is more subtle in this case. It is also important to note that the isolation system still works very well above the resonance frequency: the vibration on the concrete floor is increasingly attenuated on the plinth above 3.2 Hz.

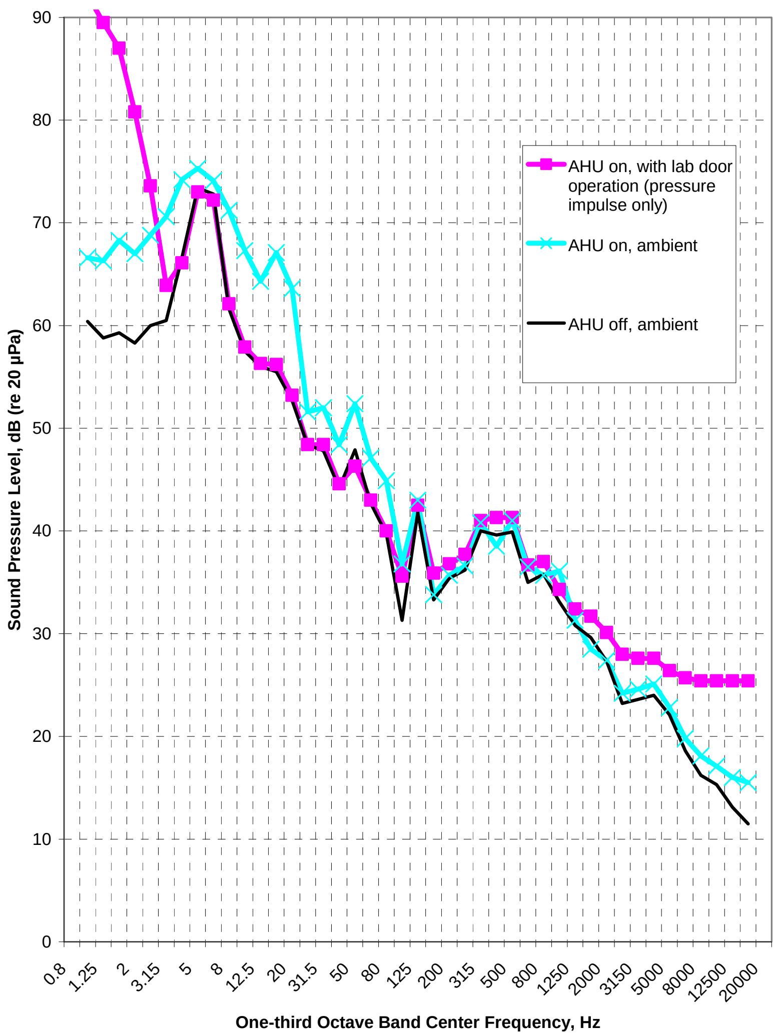

Figure 4. L_{max} sound pressure in room as a function of operation of supply air handling unit (AHU) and operation of door to room.

An important question is whether all isolation systems are affected in the same way, or if air springs in particular are more vulnerable to pressure fluctuations. Fig. 7 and 8 show the vibration amplitude on a table supported on ribbed neoprene pads also located in the same room with the pressure fluctuations shown in Fig. 4. This system has a resonance frequency of 25 Hz. At this frequency there is no amplification from the air handler variation or operating door pressure pulse. (There are vibration increases at other frequencies due to transmission of vibration through the floor and through the isolator below its resonance frequency.)

There are at least two possible explanations for the different effects seen on the different isolation systems. Either

- the amplification due to pressure events is a pressure (force) effect on the air springs (which cannot by the same mechanism affect solid neoprene); or
- the neoprene supported platform, because this isolation system has more damping, has not been subjected to adequate force to cause a significant response.

In both cases the laboratories in which the above data were collected are approximately 7m by 10m in floor area and about 5m in height (excluding lay-in ceiling systems). The dimensions of the plinths are such that any surface is less than a few meters square in area. The acoustic wavelength, then, is much larger than the room or the plinths at the exciting frequency. It seems unlikely, therefore, that what we are seeing is the effect of pressure on the surfaces of the plinth, since this would apply to all surfaces coherently. It is more likely that the results are due to the effect of varying outside pressure on the air bladder of the spring system.[*]

3. Summary and conclusions

Resiliently-supported plinths are an effective means for providing an improved vibration environment within their operating frequency range. When these are supported on passive isolation systems, it is normal to have a low-frequency resonance associated with the isolation system, which determines the isolation frequency. At the resonance frequency it is typical for the vibration in the supporting structure to be amplified to some degree.

There are cases in which one would prefer to minimize amplification at resonance. This paper has shown that the acoustic environment can affect the amplitude of this resonance, particularly in systems supported on air springs. This is thought to be due to outside pressure fluctuation interacting with the pressurized bladder of the air spring. For best performance of plinths and tables supported on air springs, it may be necessary to consider the impact of the acoustic noise in the room, especially in the infrasonic frequency range, due to such sources as air handling systems, door motion, and others.

REFERENCES

[1] H. Amick, B. Sennewald, N. C. Pardue, C. Teague, and B. Scace, Analytical / Experimental Study of Vibration of a Room-Sized Airspring-Supported Slab, *Noise Control Engineering Journal*, 1998, v. 46, no. 2, pp. 39-47.

[*] There is a test that could be carried out to verify that it is not the increased damping in the neoprene pad isolation system that keeps it from being excited at resonance due to pressure on the surface of the plinth. The test would compare the acoustic pressure response of plinths supported on air springs and on steel springs, which have more similar damping characteristics to air springs, as well as similar resonance frequencies for static deflections of 50mm or more.

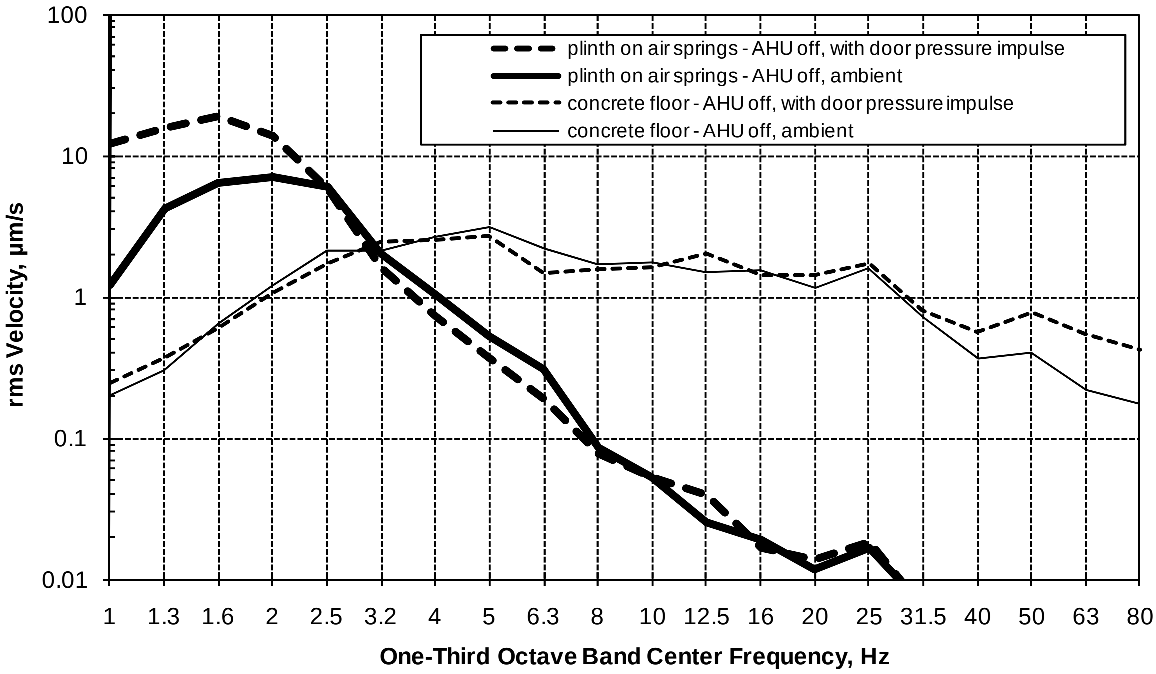

Figure 5. Maximum rms vibration amplitude on floor and plinth on air springs during operation of door to room (pressure variation shown in Fig. 4).

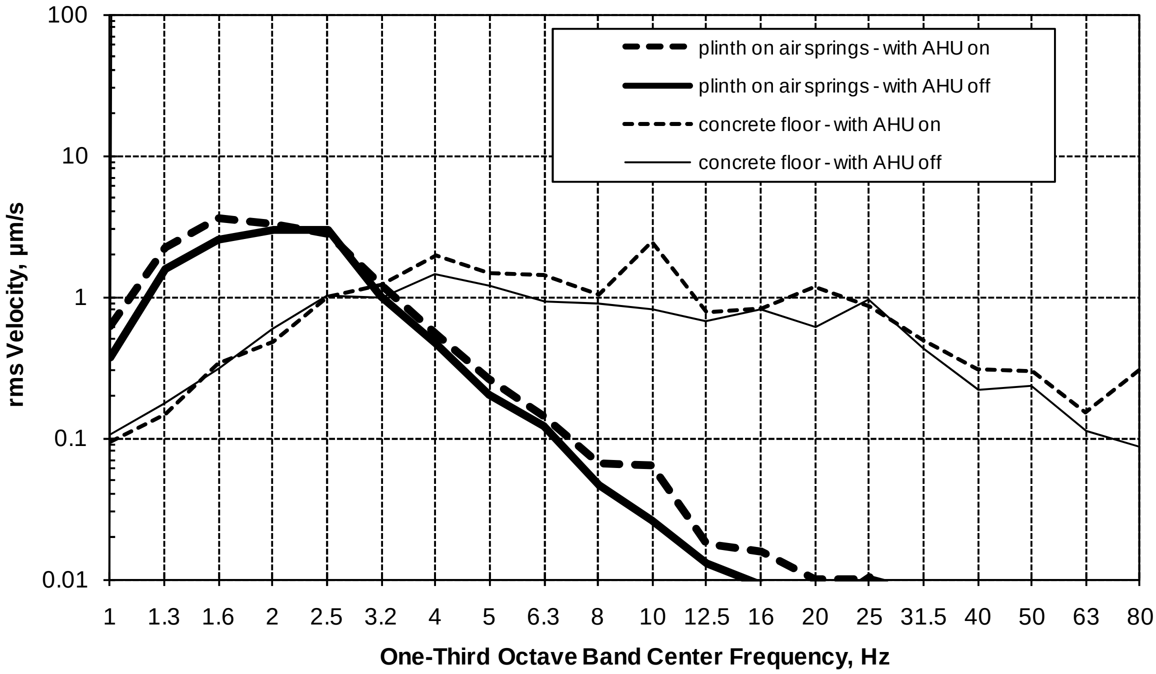

Figure 6. Linear average rms vibration amplitude on floor and plinth on air springs during operation of air handling unit (pressure variation shown in Fig. 4).

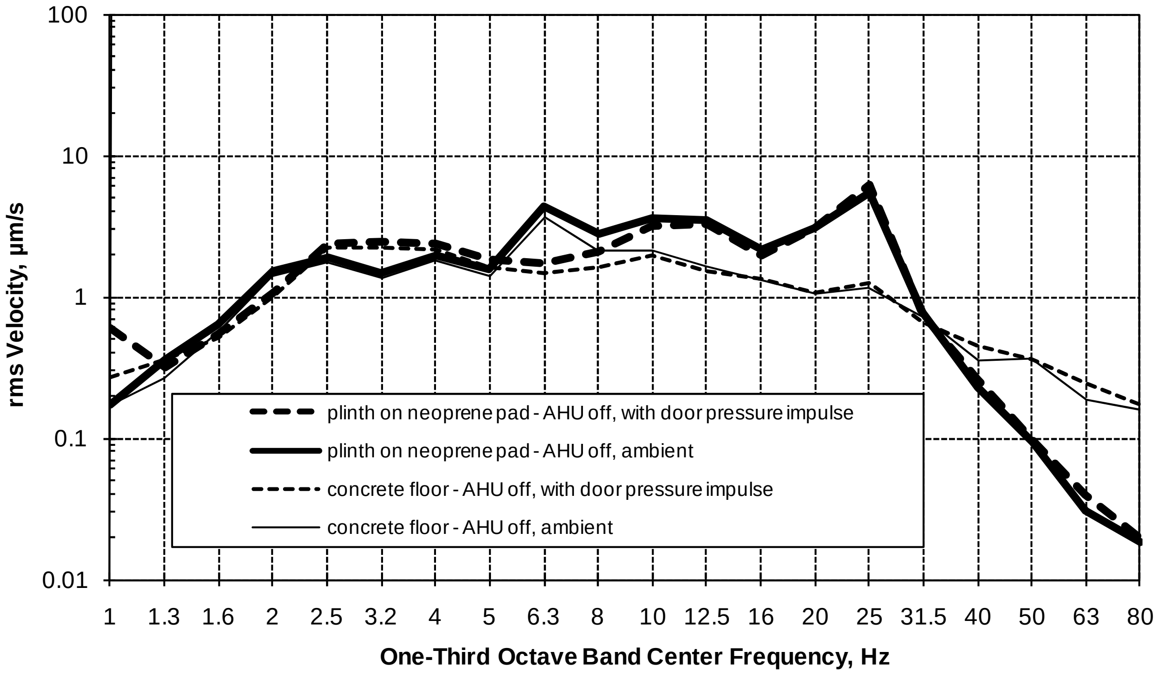

Figure 7. Maximum rms vibration amplitude on floor and plinth on neoprene pads during operation of door to room (pressure variation shown in Fig. 4).

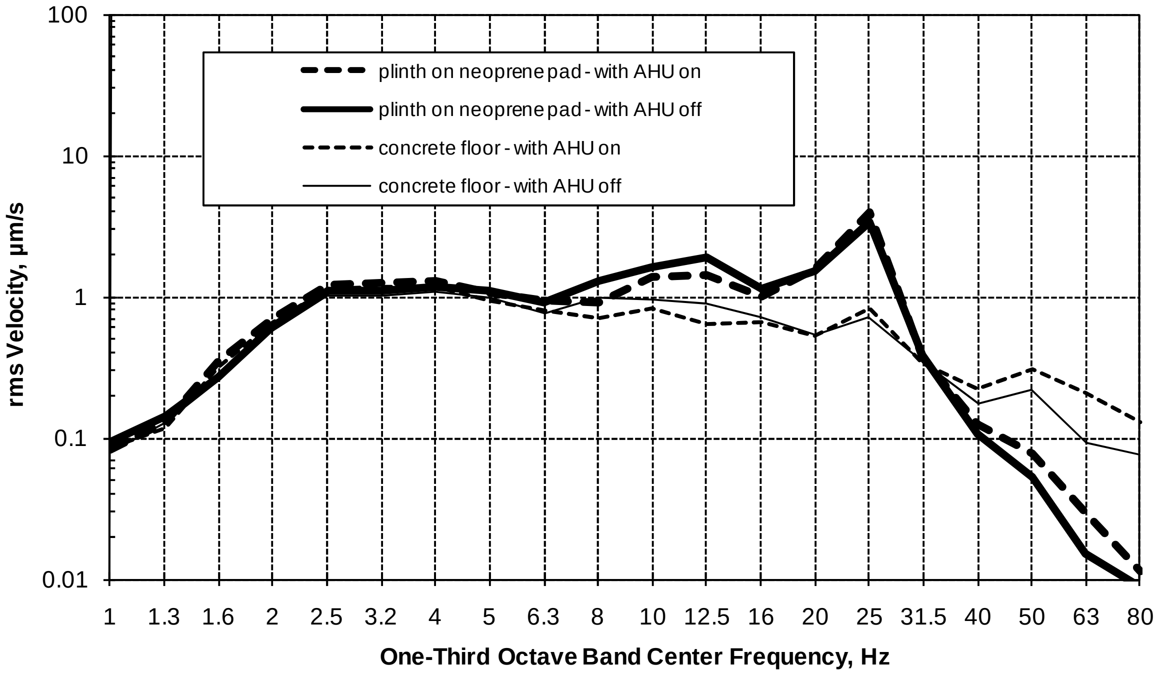

Figure 8. Linear average rms vibration amplitude on floor and plinth on neoprene pads during operation of air handling unit (pressure variation shown in Fig. 4).