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VIBRATION ISOLATION MEASUREMENTS ON A PRECISION MACHINE TOOL*

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Normal floor vibrations might be negligible for conventional machine tools; but for very precise machines, such as those used for diamond turning mirror finishes, even low level vibrations can cause detrimental effects. To isolate these floor vibrations it is common to use air or pneumatic systems, and the modified No. 3 Moore machine we use at the Los Alamos Scientific Lab (LASL) for precision turning is supported on three Barry air mounts for isolation and servo-leveling. Also we plan to use a similar system on a new larger precision turning machine which is scheduled for installation during 1979.

We have studied the air isolation system used on the No. 3 Moore machine, and we have concluded that the system is effectively isolating the floor vibrations; however, more work is required to describe the complete dynamic behavior of the machine as it interacts with the isolation system. With this work we hope to increase our understanding of isolation for machine tools so that we can specify or design systems for future machines.

As for any real system, the response of a machine tool to input vibration, such as floor motions, can be quite complicated with many different degrees of freedom or modes of vibration. Such a system can be analytically described by many different masses connected with springs and dampers, but for describing the operation of an isolation system a simple one degree of freedom spring-mass system is useful. The way a simple system responds to different harmonic motion inputs is illustrated in Fig. 1. The ratio of the distance the mass moves (y) to the maximum distance the base moves (x) is called transmissibility.

For a good isolator we should have a very low transmissibility, or the motion of the mass (for example a machine) should be smaller than the input or floor motion. To achieve low transmissibility it is necessary to have a large frequency ratio, or toward the right end of the horizontal axis of Fig. 1. In other words, we should have the natural frequency of the mass and spring ($\omega_0 = \sqrt{K/M}$, much lower than the forcing frequency).

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There are often practical limits on how low the natural frequency can be designed, and the driving frequency may be near the natural frequency of the system, and then it is necessary to have some damping in the system to limit the magnitude at resonance. The ideal situation would be to have a highly damped system for frequency ratios less than $\sqrt{2}$ (where the curves cross the axis), but have a lightly damped system for higher frequency ratios to attenuate the effect of floor motions.

The curves shown in Fig. 1 are for an elastic spring and parallel damper system, like an automobile spring system, which are limited because large static deflections are required for low natural frequency systems and they are not sensitive to very low level vibrations. Air systems are used because low natural frequencies (in the range of 1 Hz) are possible and they can respond to vibrations even at the micro-g level. According to Barry literature, air systems respond as a damped system (about 0.2 damping ratio) near resonance, but attenuate the higher frequencies as if the system were undamped.

In the case of machine tools, the ultimate concern is how any vibration might affect the distance between the workpiece and tool. Therefore, also the response of the machine itself is of concern, and it might be unnecessary to isolate the machine for all frequencies. The normal approach is to isolate the higher frequencies with the isolation system, and build the machine stiff enough so that low frequencies don't produce an unacceptable (approximately one microinch) movement between tool and workpiece.

As a first step in experimentally evaluating our system, I measured some floor motions in our shop area near the machine. An example of these measurements is presented in Fig. 2 as three different acceleration time histories. The acceleration signals were generated by a Sundstran servo accelerometer and recorded as a function of time on a digital oscilloscope. The top curve shows a sample of low level vibrations, about 1 milli-g peak with a frequency of about 120 Hz. It was interesting to note that at the same location a few hours later the signal looks different as shown in the second trace. Also there are indications that a lower frequency is also present, therefore, I took a very slow sweep data sample as shown in the third time history of Fig. 2. These three data samples indicate that many different frequencies exist, from above 120 Hz to well below 1 Hz.

A convenient way to examine a vibration signal such as the floor motions, is to produce a curve of acceleration amplitude versus frequency. Such a plot is called a frequency spectrum of the vibra[illegible]n and can be produced by an electronic device called [illegible]rum analyzer. The frequency spectrum representation o[illegible] acceleration output from the same location as the time histo[illegible] is shown in Fig. 3. This spectrum is the vertical accelerat[illegible] but a horizontal acceleration spectrum in different directi[illegible]s could also be produced. The 120 Hz and higher harmonics appear predominant; however, there are several peaks in the range below 50 Hz.

To predict how effectively an isolation system would eliminate such floor vibrations it is necessary to know the natural frequency and damping of the system. To measure these quantities for our machine, we placed an accelerometer on the machine table and pushed on the machine to cause a disturbance. The response from the disturbance was recorded, and is shown in Fig. 4. This response is for the horizontal direction parallel to the spindle axis, and other curves could be obtained for another horizontal or vertical direction. From the period of this response the natural frequency is estimated as 2.5 Hz, and by the method of log decrement a damping ratio of about 0.3 was calculated. This data indicates the system should isolate frequencies above about 4 Hz; for example, the transmissibility of a 20 Hz floor vibration would be 0.1, and an acceleration peak of one milli-g floor motion would produce an acceleration of only 0.1 milli-g in the machine.

Another approach to evaluate an isolator, which is often used for mechanical equipment, would be to put the machine and isolation on a large shaker and sweep through a range of frequencies. For the case of a diamond turning lathe, a very large shaker would be required and such a test would be expensive and time consuming. Therefore, a simple test was conducted to further evaluate the effectiveness of the isolation system.

A series of measurements were performed with the air released from the air suspension system and compared to results with the standard air pressure in the system. With the machine base setting on blocks instead of the air isolator, a response spectrum of the machine table acceleration was taken and is shown in Fig. 5. Horizontal direction acceleration was selected for these tests, but the 120 Hz component is still present as in the vertical. The range from 0-50 Hz is expanded in Fig. 6, which illustrates a peak at 30 Hz (or 1800 rpm, a common motor speed) and other peaks which should be attenuated with our isolation system.

The spectrum of Fig. 6 was taken with the machine sitting still on the floor with no isolation; however, with the spindle running the spectrum changes as shown in Fig. 7. The peak of 20 Hz (1200 rpm) indicates there is an unbalance of the spindle causing motion in the direction of the spindle axis.

Figure 8 is the spectrum with the spindle still running, but the isolation system operating. The 30 and 40 Hz peaks of Fig. 7 are clearly eliminated along with some of the lower peaks, but not the 20 Hz peak which is generated within the machine itself. Also of interest is the low frequency peak which appears in Fig. 8 for the isolated case, which was not present in the unisolated case (Fig. 7). This probably indicates the machine and isolator are vibrating at a natural frequency of the system (about 4 Hz).

As a concluding test we diamond turned two small aluminum mirrors, one with isolation and one without. Both mirrors look good with visual observation; however, from an interference microscope picture the mirror turned with an air isolation system is superior. It was surprising how little difference the isolation system improved the mirror, but the results could have been different if a large disturbance (for example, a fork lift driving by) had occurred while the mirror was being turned without isolation.

Using the accelerometer and spectrum analyzer for evaluating the performance of our existing machine and isolation system has been useful, and the methods will be used further to investigate vibration related machining problems on this and other machines. We plan to use a similar vibration measuring system while cutting parts on our new machine to provide immediate data on vibration or isolation problems.

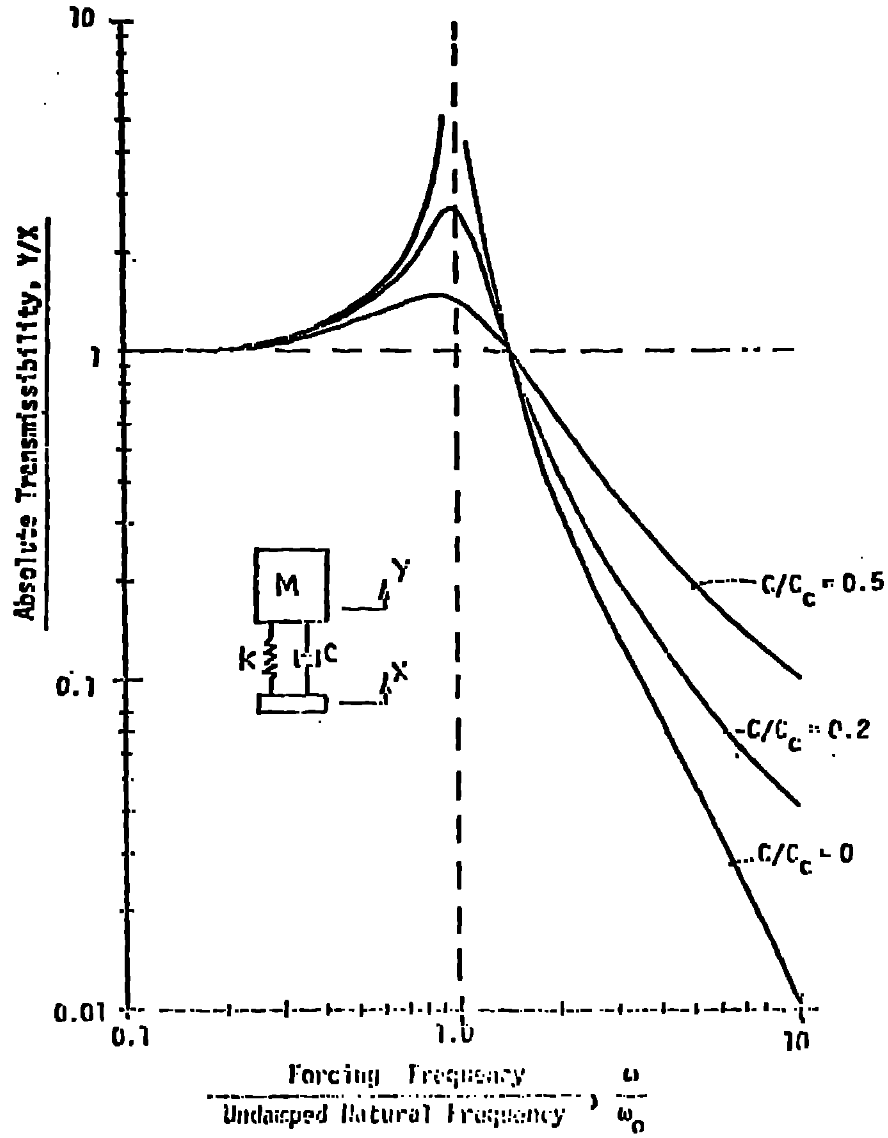

Fig. 1. Isolation for Simple Vibrating System

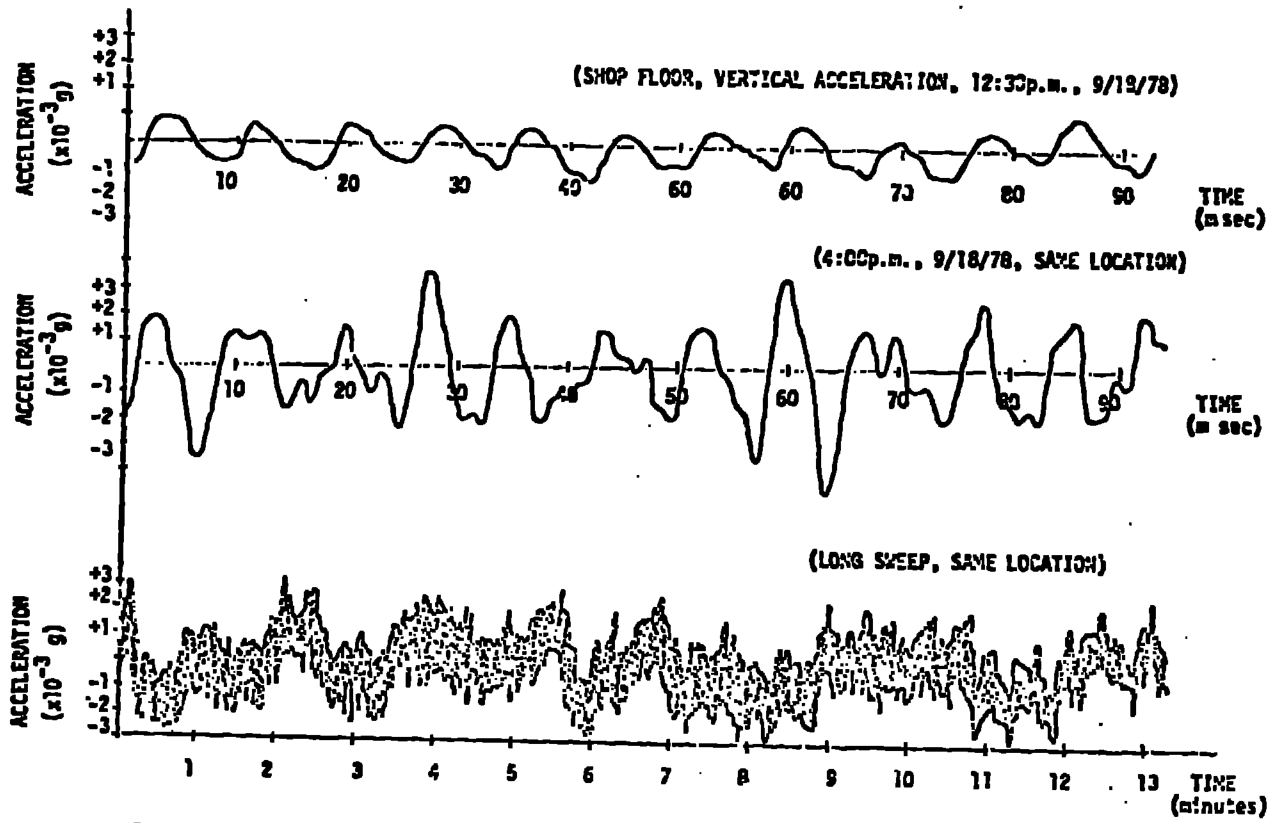

FIG. 2 Typical Shop Floor Motions

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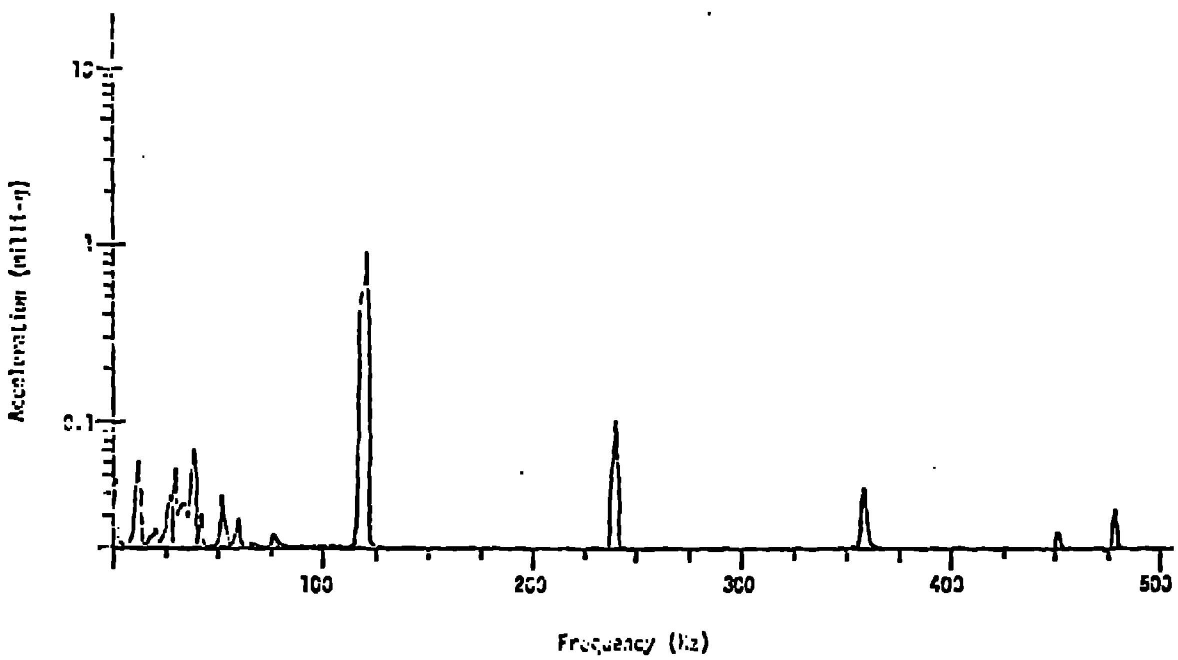

Fig. 3. Spectrum - Vertical Floor Acceleration

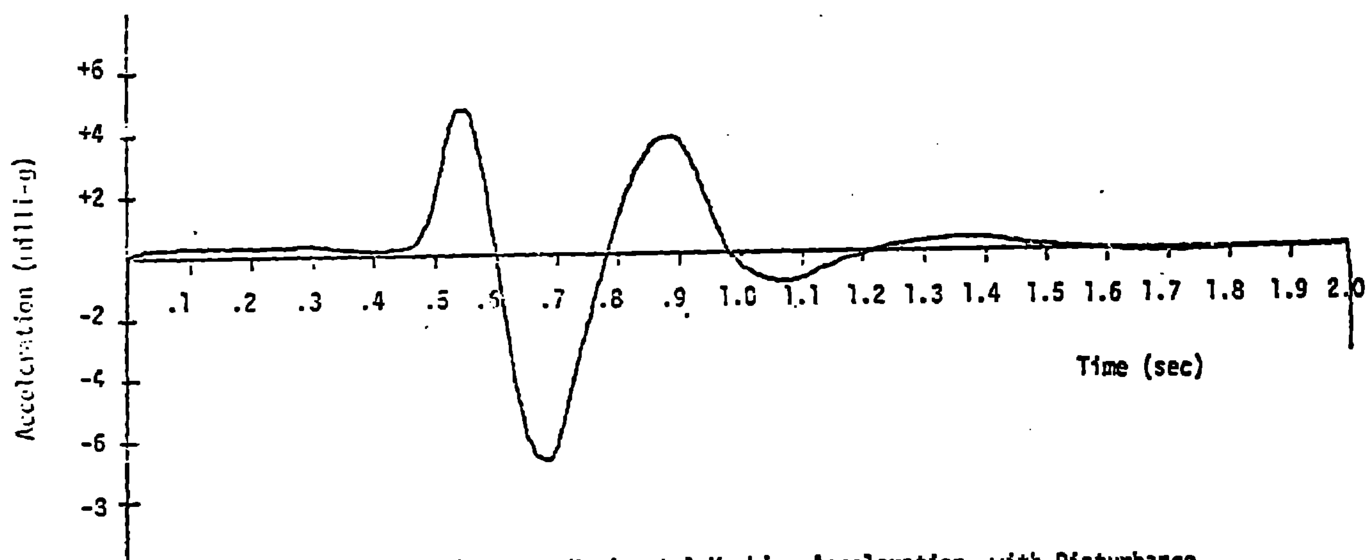

Fig. 4. Time History - Horizontal Machine Acceleration, with Disturbance

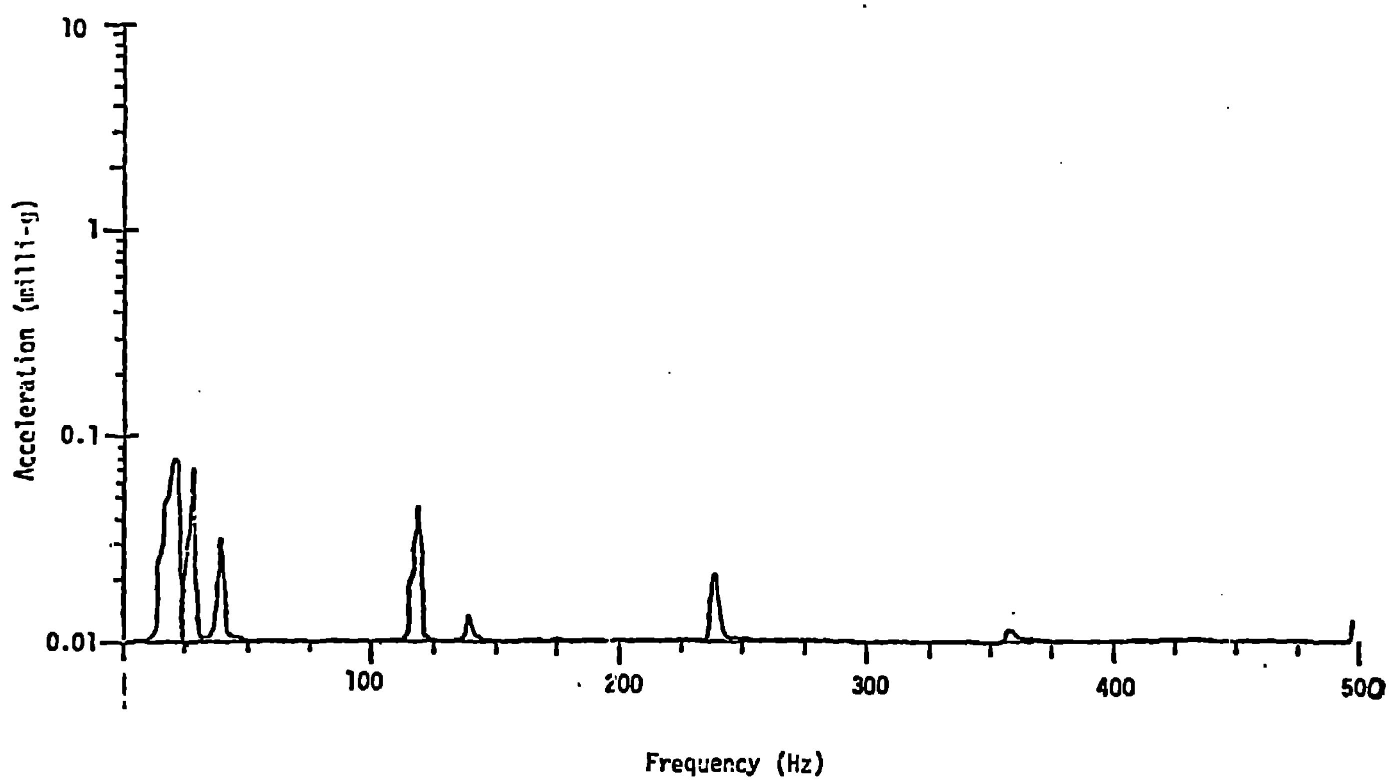

Fig. 5. Spectrum - Horizontal Machine Acceleration, No Isolation System

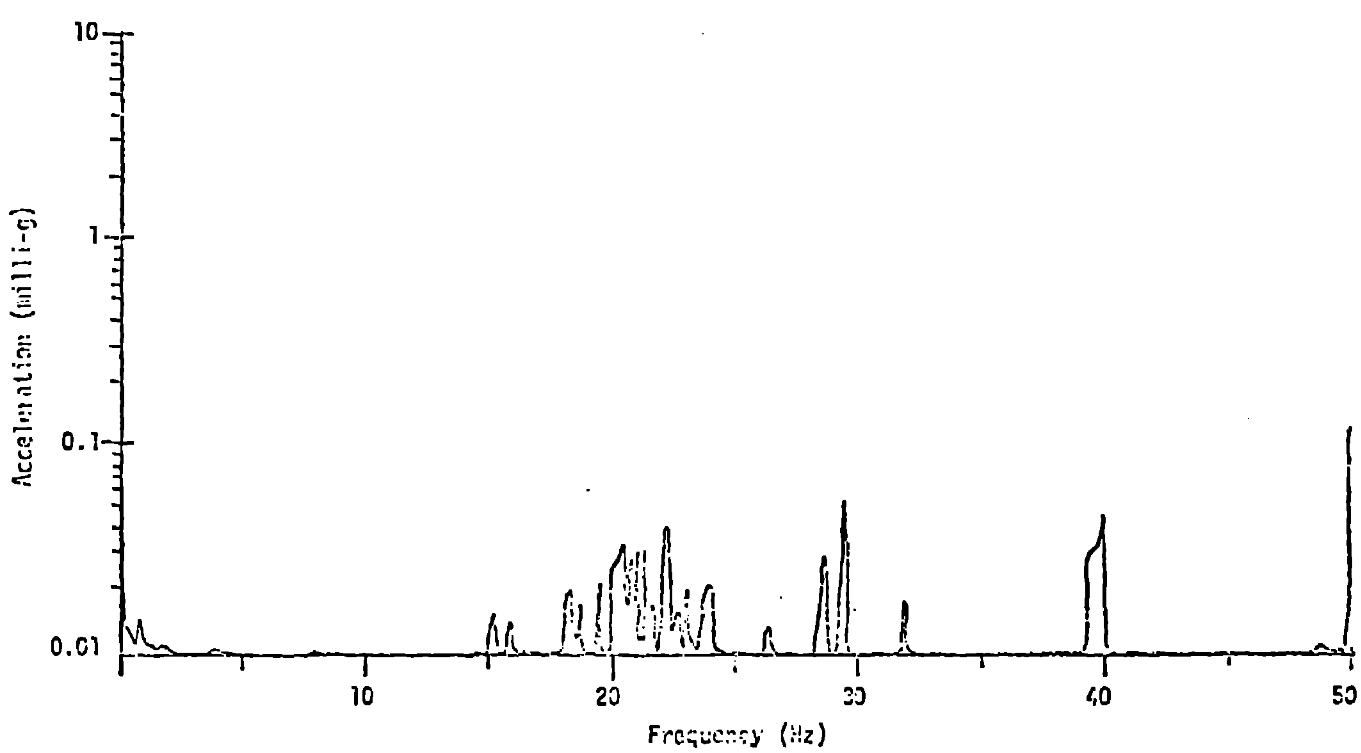

Fig. 6. Spectrum - Horizontal Machine Acceleration, No Isolation System

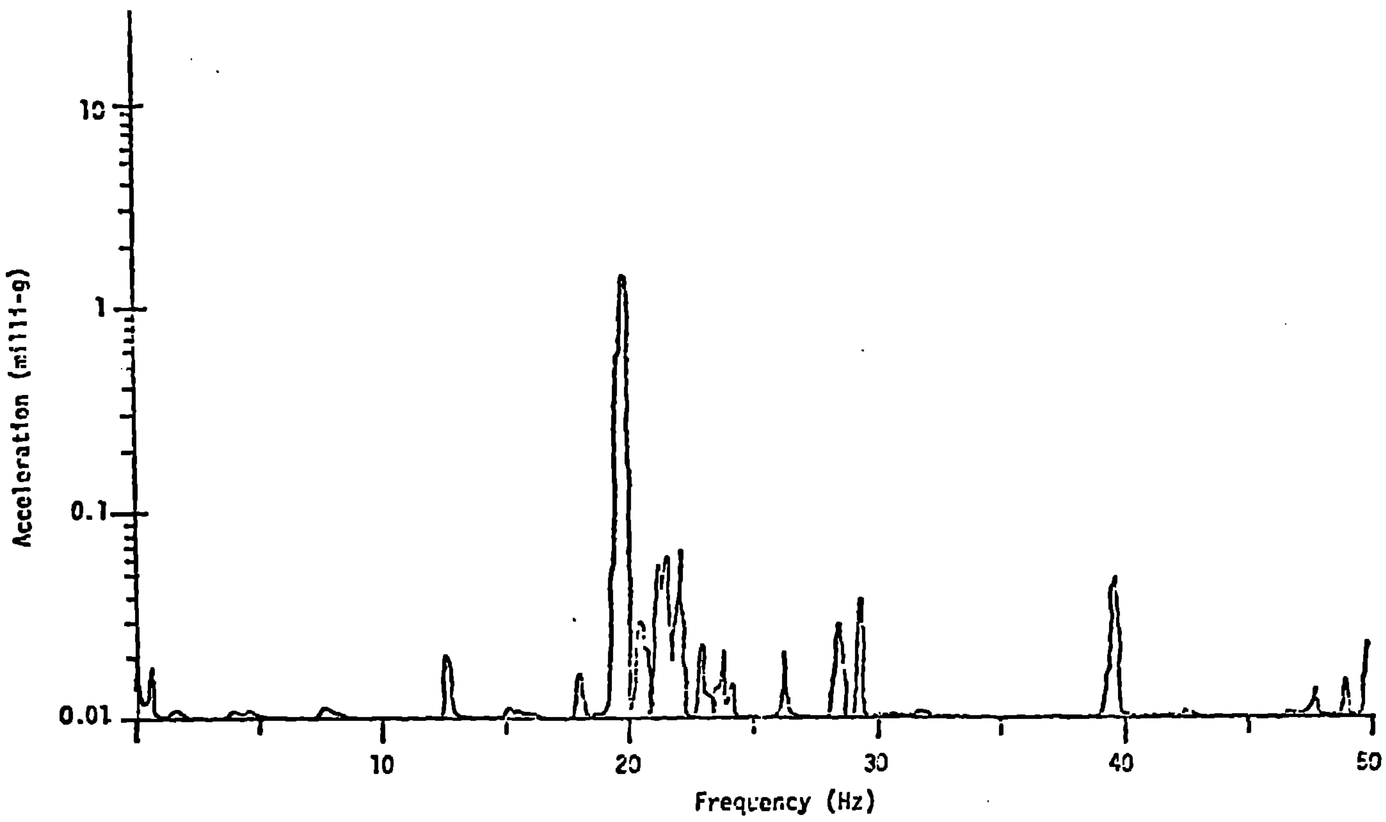

Fig. 7. Spectrum - Horizontal Machine Acceleration, No Isolation, Spindle Running

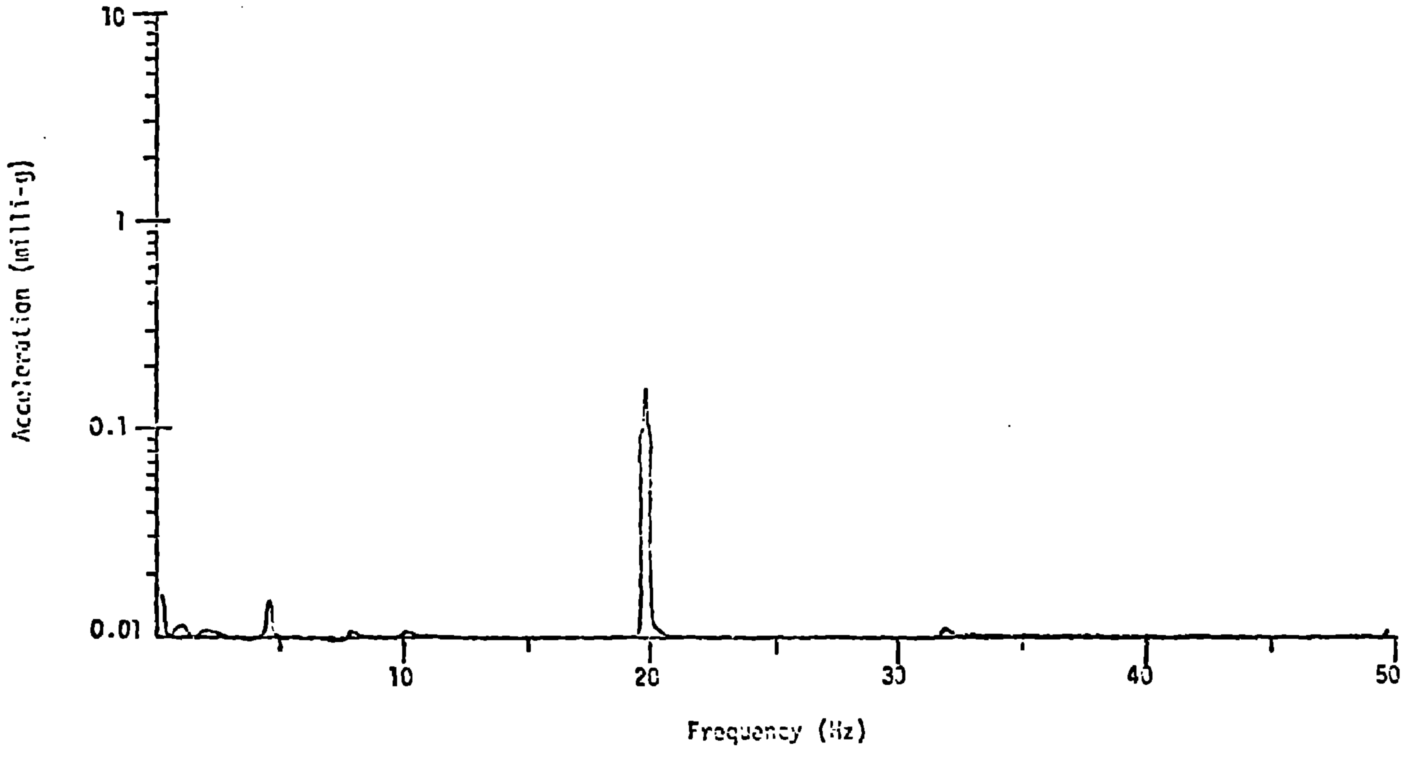

Fig. 8. Spectrum - Horizontal Machine Acceleration, With Isolation, Spindle Running