CHAPTER 26

VIBRATION

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INTRODUCTION

Exposure to vibration is frequently associated with exposure to noise in industrial processes since the two often originate from the same operation. However, the adverse effects resulting from exposure to noise and to vibration are quite different in nature, the former having a more substantial basis than the latter for establishing a cause-and-effect relationship both qualitatively and quantitatively. As more information concerning industrial exposures to vibration becomes available, particularly within the United States, and appropriate exposure criteria are established and standards adopted, the now common practice of taking noise surveys in industrial situations will likely be extended by the use of a vibration sensor to assist the investigator in evaluating the exposures of workers to both noise and vibration.

The effects of exposure to noise have been thoroughly investigated and the results of these studies are reflected in current legislation. Although a significant amount of research is underway on the relationship between exposure to vibration and the health and well-being of the persons exposed, sufficient evidence for the establishment of occupational health standards has not yet been developed.

Therefore the purpose of this chapter is to acquaint the reader with the general principles involved in recognition, evaluation and control of workers' exposure to vibration. Except to the extent necessary, the chapter will not discuss considerations of vibration in noise control efforts, since this is done in detail in Chapter 37.

EFFECTS OF VIBRATION ON MAN

The human body is an extremely complex physical and biological system. When looked upon as a mechanical system, it contains a number of linear and non-linear elements, the mechanical properties of which differ from person to person. Biologically, and certainly psychologically, the system is by no means any simpler than it is mechanically. On the basis of experimental studies, as well as documented reports of industrial experience, it is apparent that exposure of workers to vibration can result in profound effects on the human body — mechanically, biologically, physiologically and psychologically.

It should be noted at this point that relatively few studies of industrial exposures to vibration have been conducted in the United States; most of the available literature on documentations of effects of vibration in industrial situations has been published in European countries. There have been considerable numbers of military and, relatively recently, agriculturally applied research programs in the United States. The application of such studies to general industrial situations is obviously limited. However, they have been valuable in determining some of the parameters of importance in investigating the response of people exposed to vibration and, for this reason, results of some of these studies will be discussed later in this chapter.

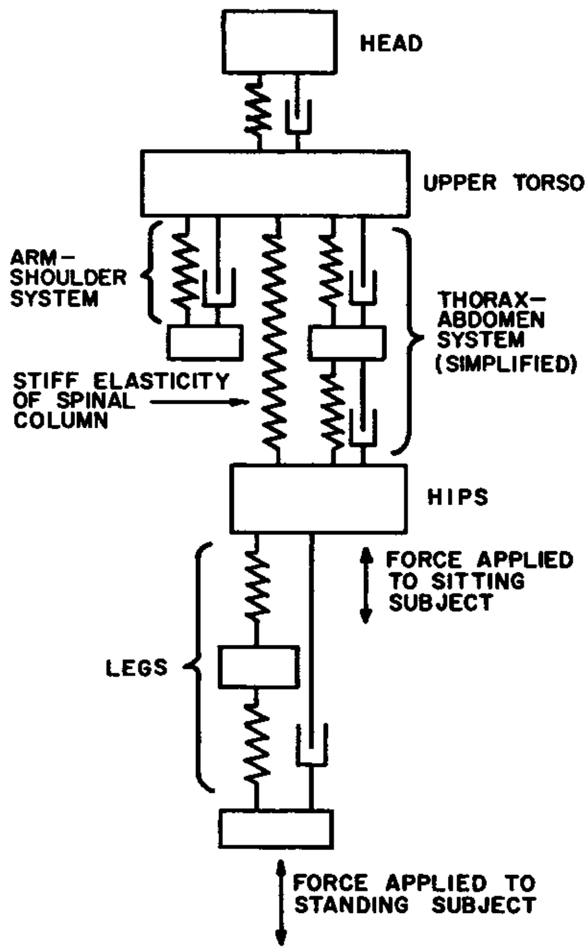

Figure 26-1. Simplified Mechanical System Representing the Human Body Standing (or Sitting) on a Vertically Vibrating Platform.[1]

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When considering the effects of vibration on man, it is necessary to classify the type of vibration exposure into one of two categories on the basis of the means by which the worker contacts the vibrating medium. The first category is referred to as "whole body" vibration and results when the whole body mass is subjected to the mechanical vibration as, for example, from a supporting surface such as a tractor seat. The second category is usually referred to as "segmental" vibration and is defined as vibration in which only part of the body, for example the hand or hands operating a chain saw, is in direct contact with the vibrating medium and the bulk of the body rests on a stationary surface. This classification of vibration does not necessarily mean that parts of the body other than those in direct contact with the vibrating surface are not affected.

If the whole body is considered as a mechanical system, at low frequencies and low vibration levels it may be approximated roughly by a simplified mechanical system such as that depicted in Figure 26-1.

Results of some of the research studies conducted in the United States are presented in Figures 26-2 and 26-3.[2,3,4] These studies have shown that, for whole body vibration, the tolerance of a seated man is lowest in the frequencies between approximately 3 and 14 Hertz. As with any tangible object, it is possible to apply externally generated vibrations to the human body at certain frequencies and in such a way that the body becomes more in resonance with the vibrating source than at other frequencies. These studies have indicated that such whole body resonances occur in the frequency range 3-6 Hertz and 10-14 Hertz. The studies have also indicated the presence of resonant effects in some of the sub-systems of the body as a result of exposure to whole body vibration. For example, resonance of the head-shoulder sub-system has been found in the 20-30 Hz range; disturbances which suggest eyeball resonance have been indicated in the 60-90 Hz range; and a resonance effect in the lower jaw and skull sub-system has been reported for the 100-200 Hz range.

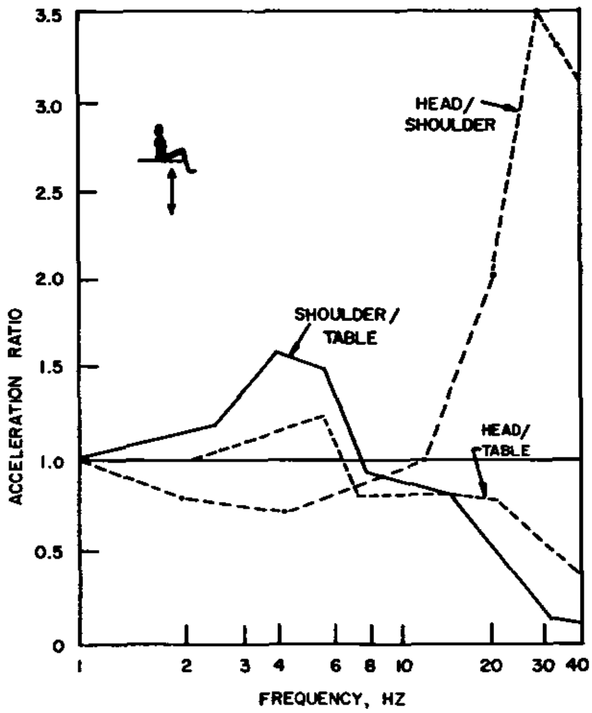

Figure 26-3. Transmissibility of Vertical Vibration from Supporting Surface to Various Parts of a Seated Human Subject as a Function of Frequency.[2,4]

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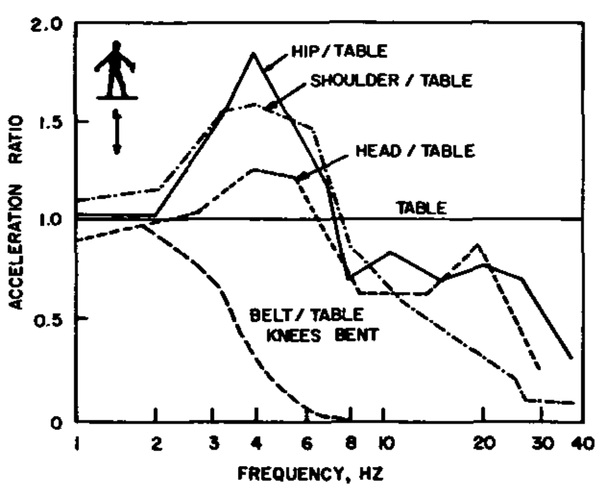

Figure 26-2. Transmissibility of Vertical Vibration from Supporting Surface to Various Parts of the Body of a Standing Human Subject as a Function of Frequency.[2,3,4]

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The preceding discussion has dealt with mechanical responses to vibration, however, there are pronounced physiological and psychological effects resulting from exposures to whole body vibration as well. Although these effects are rather complex and usually difficult to measure, the subjective responses of man to whole body vibration have been fairly well documented in the European literature. A few causal relationships between the biomechanical effects of whole body vibration and consequent physiological changes in the body are apparent. These physiological observations have included evidence of a slight acceleration in the rate of oxygen consumption, pulmonary ventilation and cardiac output.[5,6,7,8] There is evidence of an inhibition of tendon reflexes and an impairment in the ability to regulate the posture, possibly by actions through both the vestibular and spinal reflex pathways.[9,10] Alterations have been recorded in the electrical activity of the brain and there has been evidence of effects on visual acuity and performance at various levels of motor activity and task complexity during exposure to whole body vibration.[11,12] These and other studies conducted in Europe have indicated that whole body vibration has effects on the endocrinological, biochemical and histopathological systems of the body as well.

The most extensive investigations of industrial exposures of workers to vibration have been concerned with repeated exposure to low frequency vibration transmitted through the upper extremities of the worker; that is, during the use of handheld power tools which incorporate rapidly rotating or reciprocating parts. Such studies therefore consider the effects of "segmental" vibration on the worker. Unfortunately, most of these studies present only general descriptions of the clinical evidence of overexposure to vibration and very few contain any controlled observations by quantitative techniques; consequently there is a large variability between the observations reported by different investigators. However, the main features of what can be called a vibration syndrome are evident.

The clinical evidence of overexposure to vibration during the use of hand tools can be conveniently grouped into four categories.[13] These four types of disorders, in decreasing order of their appearance in the published literature, are:

1. A traumatic vaso-spastic syndrome in the form of Raynaud's phenomenon (discussed in more detail below);
2. Neuritis and degenerative alterations, particularly in the ulnar and axillar nerves, that is, a loss of the sense of touch and thermal sensations as well as muscular weakness or even paralysis, and abnormalities of the central nervous system;
3. Decalcification of the carpal and metacarpal bones; fragmentation, deformation and necrosis of carpal bones; and
4. Muscle atrophy, tenosynovitis.

As indicated earlier, the prevalence of the different symptom groups varies tremendously in the reports of different investigators. However, a fair estimate of the average overall prevalence of the vibration syndrome, at least as typified by the Raynaud phenomenon, appears to be around 50%; that is, about half of the workers exposed to segmental vibration exhibit clinical symptoms characteristic of the Raynaud phenomenon.

Raynaud's Syndrome

Raynaud's syndrome or "dead fingers" or "white fingers" occurs mainly in the fingers of the hand used to guide a vibrating tool. The circulation in the hand becomes impaired and, when exposed to cold, the fingers become white and void of sensation, as though mildly frosted. The condition usually disappears when the fingers are warmed for some time, but a few cases have been sufficiently disabling that the men were forced to seek other types of work. In some instances, both hands are affected.

This condition has been observed in a number of occupations involving the use of fairly light vibrating tools such as the air hammers used for scarfing and chipping in the metal trades, stonecutting, lumbering and in the cleaning departments of foundries where men have a good deal of overtime work. Obviously, prevention of this condition is much more desirable than treatment. Preventive measures include directing the exhaust air from the air-driven tools away from the hands so they will not become unduly chilled, use of handles of a comfortable size for the fingers, and in some instances, substituting mechanical cleaning methods for some of the hand methods which have produced many of the cases of "white fingers." In many instances, simply preventing the fingers from becoming chilled while at work has been sufficient to eliminate the condition.

The appearance of the syndrome appears to be a function of the cumulative absorption of vibration energy, its harmonic content and on personal factors such as the age of the worker. Vibration in the frequency range 40-125 Hz has been implicated most frequently in reported cases of vibration disorders.[14,15] In most studies it appeared that an exposure time of several months was generally needed before symptoms appeared. With continued exposure there was a progressive diversification and intensification of the symptoms; some improvement of symptoms has been reported, but rarely complete recovery, with cessation of exposure to vibration. It is the opinion of many investigators in Europe that the vibration syndrome is a widespread and alarmingly common occupational disorder.[13] Symptoms of overexposure are reportedly grave or moderately grave in about half of those affected, and in all cases result in a varying loss of working capacity.

Results of the European studies of industrial vibration exposures are summarized in Table 26-1. These data were obtained from reports issued by investigators in several countries: Austria, Czechoslovakia, France, Finland, Germany, Great Britain, Italy, the Netherlands, Russia, Sweden and others. In general, these studies revealed abnormal changes in the vascular, gastric, neurological, skeletal, muscular and endocrine systems as well as definite effects on visual acuity and task performance. The presence of Raynaud's syndrome was common to practically all studies of exposures to segmental vibration. It is beyond the intent of this chapter to discuss any of the hundreds of articles published in the literature on these studies; bibliographies on the subject have been published and should be reviewed by the interested reader.[13,16]

As can be seen from Table 26-1 the vibration syndrome does indeed appear to be a widespread occupational disorder in European industry. It must be noted here again that similar studies in industries in the United States have been very limited. However, it is reasonable to assume that United States workers employed in occupations

TABLE 26-1

EUROPEAN INDUSTRIES IN WHICH CLINICAL EVIDENCE OF OVEREXPOSURE OF WORKERS TO VIBRATION HAS BEEN REPORTED

Industry	Type of Vibration	Common Vibration Sources
Agriculture	Whole body	Tractor operation
Boiler Making	Segmental	Pneumatic tools
Construction	Whole body Segmental	Heavy equipment vehicles, pneumatic drills, jackhammers, etc.
Diamond cutting	Segmental	Vibrating hand tools
Forestry	Whole body Segmental	Tractor operation chain saws
Foundries	Segmental	Vibrating cleavers
Furniture manufacture	Segmental	Pneumatic chisels
Iron and steel	Segmental	Vibrating hand tools
Lumber	Segmental	Chain saws
Machine tools	Segmental	Vibrating hand tools
Mining	Whole body Segmental	Vehicle operators rock drills
Riveting	Segmental	Hand tools
Rubber	Segmental	Pneumatic stripping tools
Sheet metal	Segmental	Stamping equipment
Shipyards	Segmental	Pneumatic hand tools
Stone dressing	Segmental	Pneumatic hand tools
Textile	Segmental	Sewing machines, looms
Transportation (operators and passengers)	Whole body	Vehicle operation

similar to those listed in Table 26-1 are being potentially exposed to excessive levels of vibration during their routine work activities. The National Institute for Occupational Safety and Health (NIOSH) has initiated a comprehensive program in which the occupational exposures to vibration in American industries is being investigated.[16]

VIBRATION EXPOSURE CRITERIA

It is obvious from the preceding discussion that the occurrence of vibration disorders in a wide cross section of industry is significant enough, both in terms of prevalence and magnitude, that appropriate standards for allowable exposure to vibration are desirable. However, at the present time there are no generally accepted limits for safe vibration levels and, in fact, the available literature, because of the variability of reported findings, does not permit the reliable construction of a vibration exposure standard. All this is not to say that individual standards and criteria, as well as corrective methods for dealing with excessive levels of industrial vibration, have not been attempted; in fact, they have. However, the approaches to establishment of criteria and/or im-

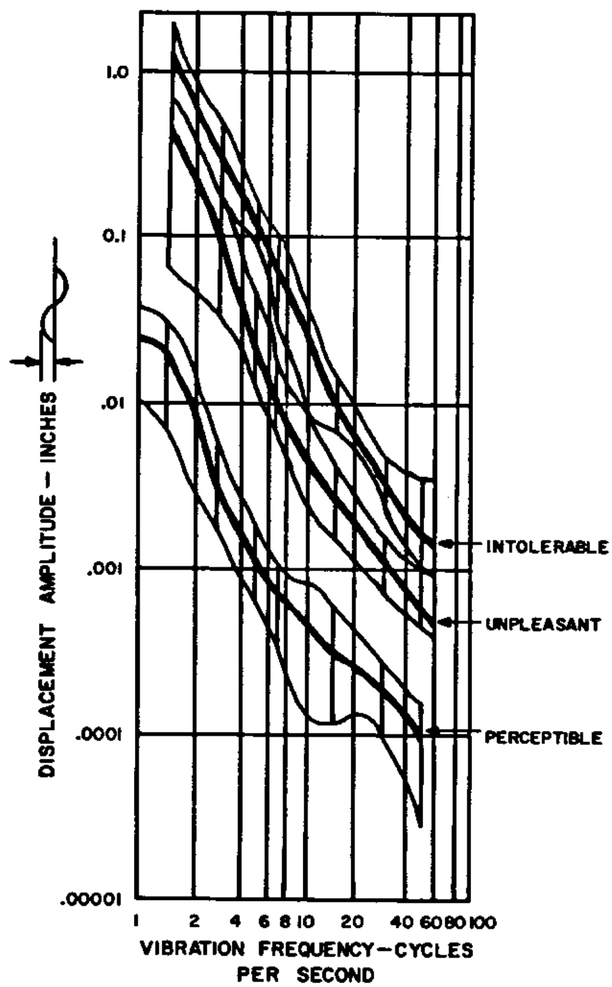

Figure 26-4. *Subjective Responses to Vibratory Motion.* The chart is based on the averaged values of various investigators and is valid for exposures up to a few minutes. The vertical arrows represent one standard deviation above and below the means.

McFarland RA Human engineering and industrial safety.

Reprinted from Vol. 1, F. A. Patty "Industrial Hygiene and Toxicology", 2nd edition. Courtesy Interscience Publishers, Inc., New York, N.Y., 1958.

plementation of corrective methods have been quite variable even within a given country.

Many studies have been conducted in which the subjective responses of exposed personnel to various levels of vibration have been documented. The results of experimental studies with human volunteers conducted by three different investigators are presented in Figure 26-4.[17] It must be emphasized that these data and, in fact, all such data in which *experimental* exposures have been documented are useful only over a very limited time duration. Most of these studies have been conducted for military and/or aerospace programs and are concerned primarily with acute exposures; that is, relatively short duration exposure such as impact or shock type of vibrations encountered in military situations. The subjects selected for such studies, therefore, are normal young, physically fit men, such as pilots. There are obvious pitfalls, therefore, in using such information as a basis for establishment of standards for industrial exposures to vibration. In the occupational workplace the normal work force is comprised of persons constituting a wide spectrum of characteristics. The industrial worker can be female as well as male, is not necessarily as physically fit as the subjects used in experimental studies, and certainly encompasses a greater range of age and other physical characteristics. Perhaps of more significance, however, is the fact that these experimental investigations represent acute or short-term exposures to vibration and certainly do not permit direct extrapolation to the typical industrial exposure which is comprised, for the most part, of exposure to relatively low frequency vibration of varying amplitudes for extended periods of time. In the case of the industrial worker then, one must be concerned with the cumulative effects of exposure to vibration over a working lifetime which can be comprised of exposure to vibration for, in an extreme example, 40 hours per week, 50 weeks per year for 40 years. Suffice to say, therefore, that the types of human exposure studies that have been conducted in the United States have limited application in the industrial environment.

Attempts *have* been made to establish vibration exposure standards and, of these, perhaps the single best vibration exposure criteria guides are those proposed by the International Standards Organization.[18] These criteria, presented in Figure 26-5 as a family of curves, are valid for vibrations transmitted to the torso of a standing or sit-

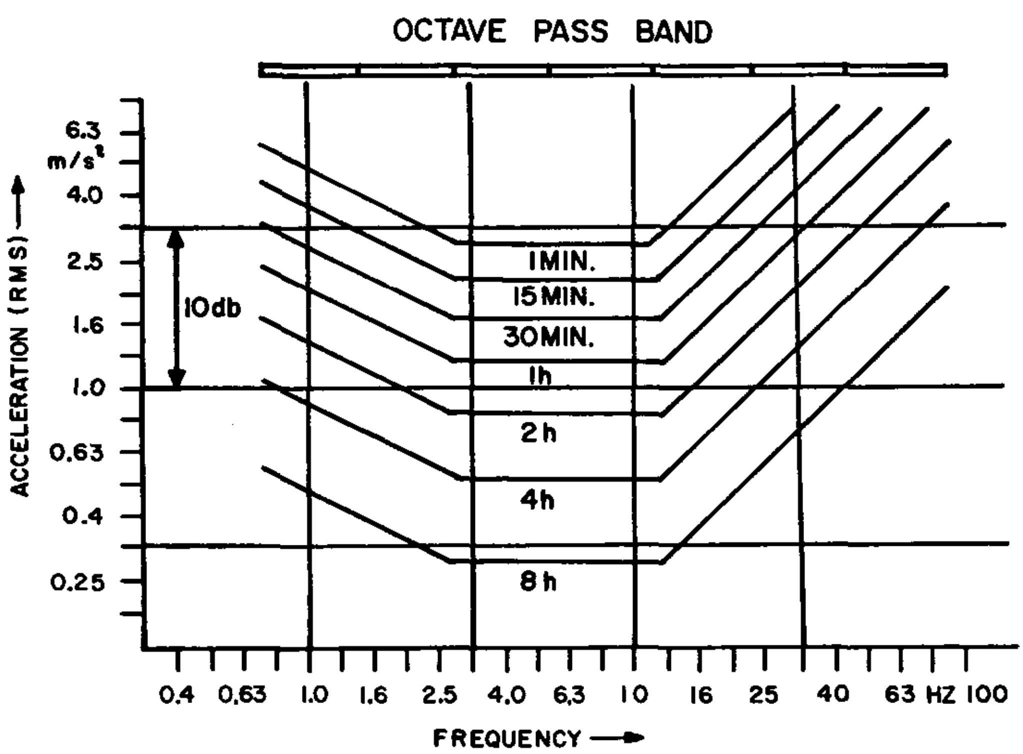

Figure 26-5. *Vibration Exposure Criteria Curves.* The vibration levels indicated by the curves in Figure 26-5 are given in terms of RMS acceleration levels which produce equal fatigue-decreased proficiency. Exceeding the exposure specified by the curves, in most situations, will cause noticeable fatigue and decreased job proficiency in most tasks. The degree of task interference depends on the subject and the complexity of the task.

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ting person and are, therefore, to be considered a guide for whole body vibration. A tentative ISO vibration guide has been proposed for segmental (hand-arm system vibrations).[19] The ISO whole body vibration guide is based to a great extent on studies conducted in the aerospace medical research laboratories in formulating guides for short-term exposures, usually of a military nature and thus again may have only limited application in industrial work environments.

The vibration levels indicated in Figure 26-5 are given in terms of the root mean square acceleration levels which produce equal "fatigue-decreased proficiency" over the frequency range of 1-100 Hz. Vibrations in frequencies below 1 Hz produce annoyances which are individually unique; for instance, cinetosis or air sickness. For frequencies above 100 Hz the vibrational perceptions are mainly effective on the skin and depend greatly upon the influenced body part and on the damping layer; for example, clothing or shoes. It seems, therefore, practically impossible to state generally valid vibration exposure criteria for frequencies outside the range indicated in Figure 26-5, that is, 1-100 Hertz. Exceeding the exposures specified by the curves in most situations will cause noticeable fatigue and decreased job efficiency. The degree of task interference depends on the subject and the complexity of the task being performed. The curves indicate the general range for onset of such interferences and the time dependency observed. An upper exposure considered hazardous to health as well as performance is considered to be twice as high (6 decibels higher) as the "fatigue-decreased efficiency" boundary shown in Figure 26-5 while the "reduced comfort" boundary is assumed to be about ⅓ (10 decibels below) the illustrated levels.

Again, these criteria are presented as recommended guidelines or trend curves, rather than firm boundaries of classified quantitative biological or psychological limits. They are intended solely for situations involving healthy, normal people considered fit for normal living routines and stress of an average work day. A program for establishing vibration exposure based upon identification and characterization of exposed workers in industries in the United States and implementation of relevant studies to ascertain the extent of the industrial vibration hazard and determination of criteria for these standards to prevent adverse exposures to industrial vibration has been initiated.[16]

CHARACTERISTICS OF VIBRATION

In general, vibration can be described as an oscillatory motion of a system. The "motion" can be simple harmonic motion, or it can be extremely complex. The "system" might be gaseous, liquid or solid. When the system is air (gaseous) and the motion involves vibration of air particles in the frequency range of 20 to 20,000 Hertz (Hz), sound is produced. For the purposes of this chapter, only the effects on the worker caused by mo-

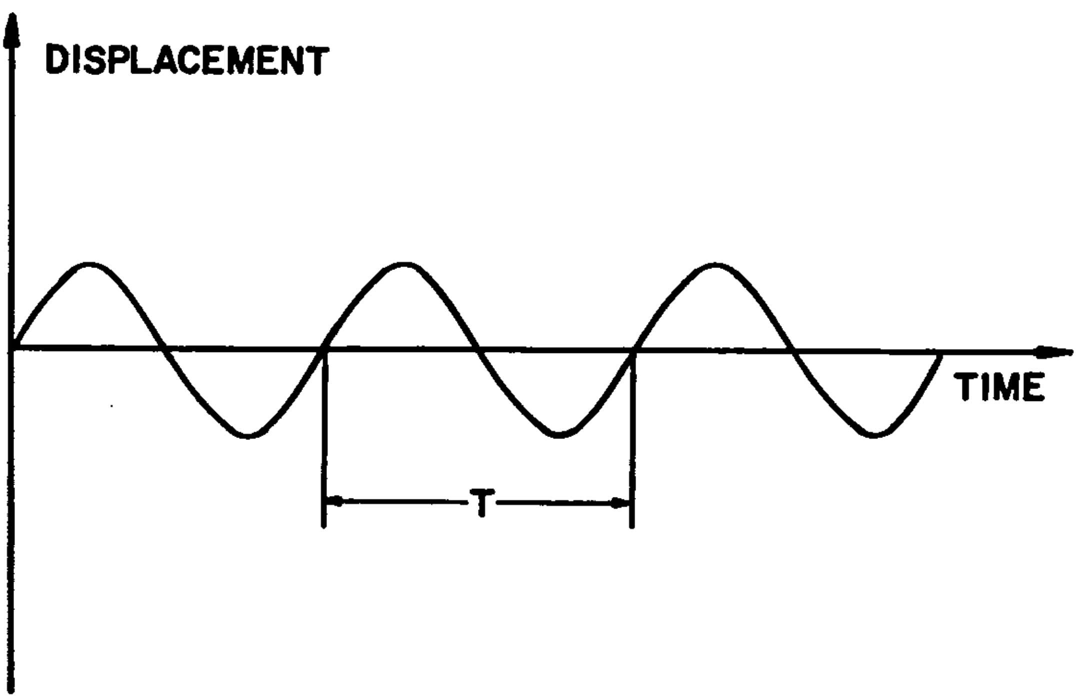

Figure 26-6. Representation of Pure Harmonic (Sinusoidal) Vibration.

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tion of *solid* systems will be considered.

The oscillation of the system may be periodic or completely random; steady-state or transient; continuous or intermittent. In any event, during vibration one or more particles of the system oscillate about some position of equilibrium.

Periodic (Sinusoidal) Vibration

Vibration is considered periodic if the oscillating motion of a particle around a position of equilibrium repeats itself *exactly* after some period of time. The simplest form of periodic vibration is called pure harmonic motion which as a function of time, can be represented by a sinusoidal curve. Such a relationship is illustrated in Figure 26-6, where T = period of vibration.

The motion of any particle can be characterized at any time by (1) displacement from the equilibrium position, (2) *velocity*, or rate of change of displacement, or (3) *acceleration*, or rate of change of velocity. For pure harmonic motion, the three characteristics of motion are related mathematically.

Displacement

The instantaneous displacement of a particle from its reference position under influence of harmonic motion can be described mathematically as:

$$s = S \sin\left(2\pi \frac{t}{T}\right) = S \sin(2\pi ft) = S \sin \omega t$$

where s = instantaneous displacement from reference position
S = maximum displacement
t = time
T = period of vibration
f = frequency of vibration
ω = angular frequency ($2\pi f$)

Of the possible vibration measurements, displacement probably is the easiest to understand and is significant in the study of deformation and bending of structures. However, only if the rate of motion, i.e. frequency of vibration, is low enough, can displacement be measured directly.

Velocity

In many practical problems, displacement is not the most important property of the vibration. For example, experience has shown that the velocity of the vibrating part is the best single criterion for use in preventive maintenance of rotating machinery.

Although peak-to-peak displacement measurements have been widely used for this purpose, it is necessary to establish a relationship between the limits for displacement and rotational speed for each machine.

Since the velocity of a moving particle is the change of displacement with respect to time, the particle's velocity can be described as:

$$v = \frac{ds}{dt} = \omega S \cos(\omega t) = V \cos(\omega t) = V \sin\left(\omega t + \frac{\pi}{2}\right)$$

where
v = instantaneous velocity
V = maximum velocity

Acceleration

In many cases of vibration, especially where mechanical failure is a consideration, actual forces set up in the vibrating parts are critical factors. Since the acceleration of a particle is proportional to these applied forces and since equal-and-opposite reactive forces result, particles in a vibrating structure exert forces on the total structure that are a function of the masses and accelerations of the vibrating parts. Thus, acceleration measurements are another means by which the motion of vibrating particles can be characterized.

The instantaneous acceleration, i.e., the time rate of change of velocity of a particle in pure harmonic motion, can be described as:

$$a = \frac{dv}{dt} = \frac{d^2s}{dt^2} = -\omega^2 S \sin(\omega t) = A \sin(\omega t + \pi)$$

where a = instantaneous acceleration and
A = maximum acceleration.

Other terms, such as "jerk," defined as the time rate of change of acceleration, are sometimes used to define vibration. At low frequencies, jerk is related to riding comfort of automobiles and elevators and is also important for determining load tie-down in airplanes, trains and trucks. However, from the equations and definitions presented in the preceding discussion, it is apparent that, regardless of the parameter being studied, i.e., displacement, velocity, acceleration or jerk, the form and period of the vibration are the same.

As noted above, the instantaneous magnitudes of the various parameters have been defined in terms of their peak values, that is, the maximum values obtained. This approach is quite useful in the consideration of pure harmonic vibration because it can be applied directly in the above equations. However, in the consideration of more complex vibrations it is desirable to use other descriptive quantities. One reason for this is that the peak values describe the vibration in terms of a quantity dependent only upon an instantaneous vibration magnitude, regardless of the previous history of the vibration. One descriptive quantity which does take the time history into account is the "average absolute value" defined as:

$$S\,(\text{average}) = \frac{1}{T} \int_0^T |s| \, dt$$

Although the above quantity does take the time history of the vibration into account over one period, it has very limited practical usefulness. The root mean square (rms) value is a much more useful descriptive quantity which also takes the time history of the vibration parameter into account. This quantity is defined as:

$$S_{rms} = \sqrt{\frac{1}{T} \int_0^T s^2(t) \, dt}$$

The importance of the rms value as a descriptive quantity lies mainly in its direct relationship to the energy content of the vibration. For example, in the case of pure harmonic motion the relationship between the various values is:

$$S_{rms} = \frac{\pi}{2\sqrt{2}} \quad S\,(_{average}) = \frac{1}{\sqrt{2}} S$$

Or, in a more general form:

$$S_{rms} = F_f \qquad S_{(average)} = \frac{1}{F_c} S$$

The factors, F_f and F_c, are called "form factor" and "crest factor," respectively, and give an indication of the wave shape of the vibration being studied. For pure harmonic motion

$$F_f = \frac{\pi}{2\sqrt{2}} = 1.11$$

and

$$F_c = \sqrt{2} = 1.414$$

There have been attempts to introduce the concept of "acceleration level" and "velocity level." These terms, similar in concept to the sound pressure level used in expressing noise levels, indicate the acceleration and velocity in decibels; that is, the logarithm of the ratio of the acceleration or velocity to a reference acceleration or velocity. Although several references have been proposed, the most common velocity reference value is 10^{-8} meters per second; i.e., 10^{-6} centimeters per second. The acceleration reference value is 10^{-5} meters per second squared or 10^{-3} centimeters per second squared. Suitable "standard reference values" for acceleration, velocity and displacement are still being studied.

The discussion and certainly the equations which have been presented to this point have dealt exclusively with the periodic vibrations, and more specifically with pure harmonic periodic vibrations. It must be pointed out that most of the vibrations encountered in industry and in fact, practically anywhere, are not pure harmonic motions, even though many of them certainly can be characterized as periodic. A typical nonharmonic periodic vibration is illustrated in Figure 26-7.

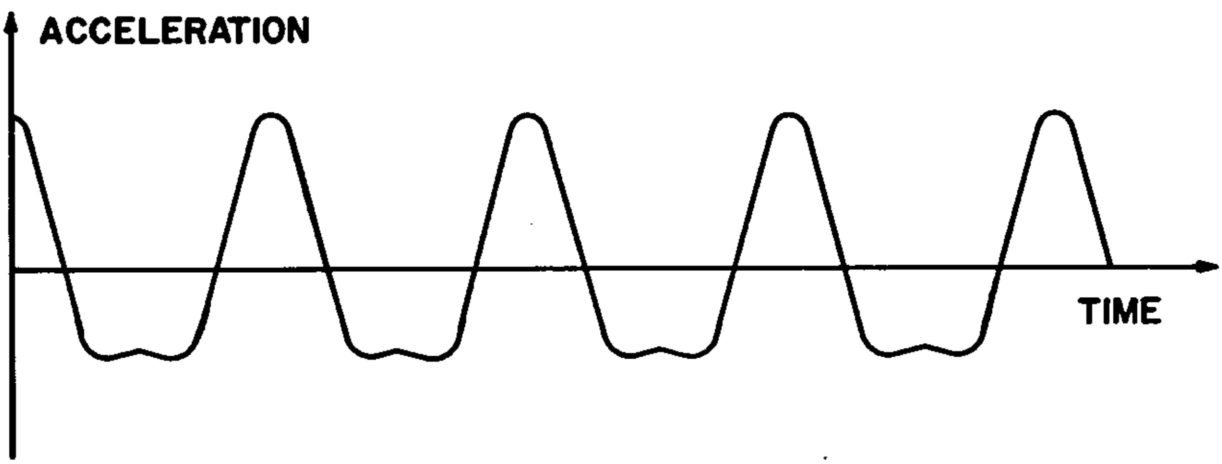

Figure 26-7. Example of a *Non-Harmonic Periodic Motion* (Piston Acceleration of a Combustion Engine).

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By examining this curve and determining the peak absolute and root mean square values of this vibration as well as the form factor and crest factor, it is obvious that the motion is not harmonic. However, on the basis of this information, it would be practically impossible to predict all of the various effects the vibration might produce in connected structural elements. Obviously, other methods of description must be used; one of the most powerful descriptive methods is that of frequency analysis, which is based on a mathematical theorem first formulated by Fourier, which states that "any periodic curve no matter how complex may be looked upon as a combination of a number of pure sinusoidal curves with harmonically related frequencies."

As the number of elements in the series increases, it becomes an increasingly better approximation to the actual curve. The various elements constitute the vibration frequency spectrum and in Figure 26-8 the non-harmonic periodic vibration illustrated in Figure 26-7 is re-illustrated, together with two important harmonic curves which represent its frequency spectrum. A more convenient method of representing this spectrum is shown in Figure 26-9. This characteristic of periodic vibrations is evident in examining Figure 26-10. Their spectra consist of discreet lines when represented in the so-called "frequency domain"; random vibrations show continuous frequency spectra when represented in similar fashion.

Random Vibrations

Random vibrations occur quite frequently in nature and may be defined as motion in which the vibrating particles undergo irregular motion cycles that *never* repeat themselves exactly. Theoretically then, obtaining a complete description of the vi-

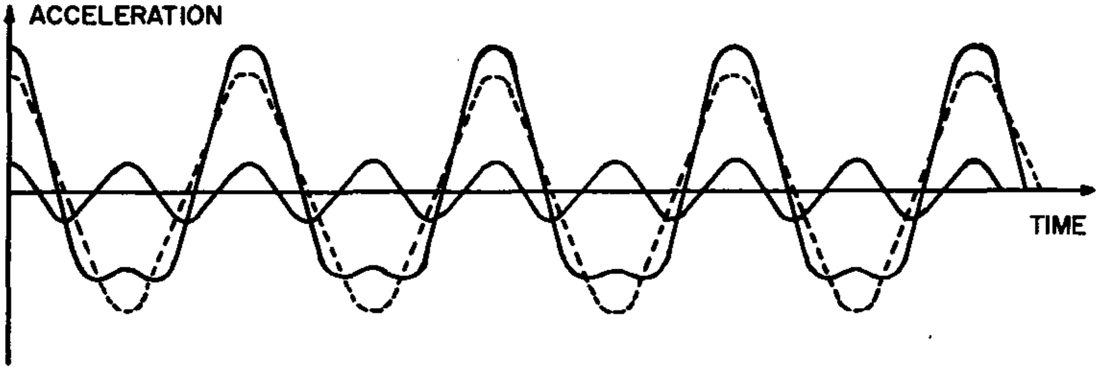

Figure 26-8. Illustration of How the Waveform Shown in Figure 26-7 Can Be "Broken Up" into a Sum of Harmonic Related Sinewaves.

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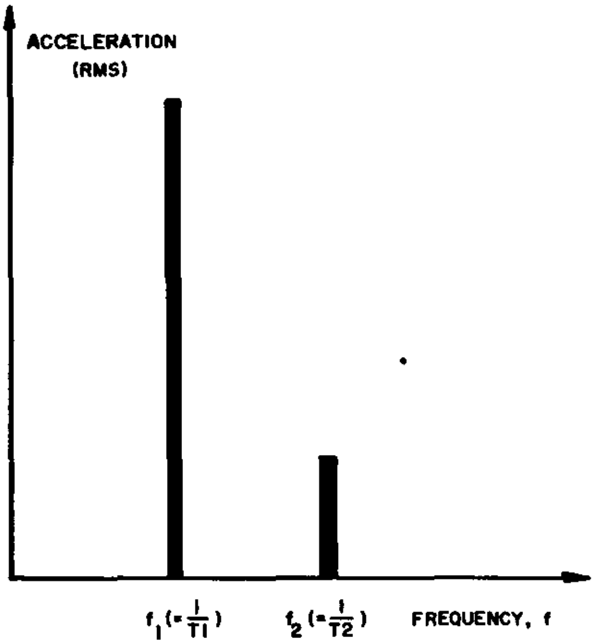

Figure 26-9. Examples of Periodic Signals and their Frequency Spectra.

a) Description in the time domain.

b) Description in the frequency domain.

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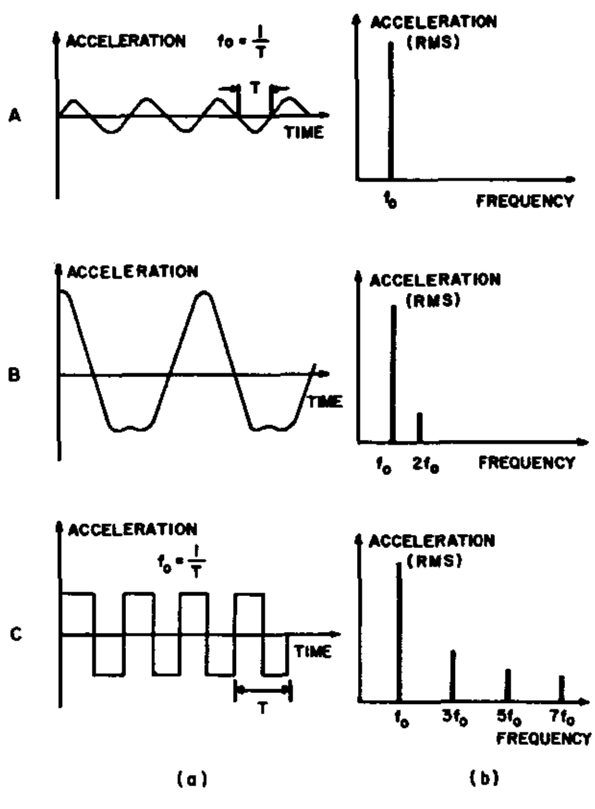

Figure 26-10. Examples of Periodic Signals and their Frequency Spectra.

a) Description in the time domain.

b) Description in the frequency domain.

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brations requires an infinitely long time record. This, of course, is an impossible requirement and finite time records would have to be used in practice. Even so, if the time record becomes too long, it also will become a very inconvenient means of describing the vibration, and other methods therefore have to be devised, and are commonly used. It is beyond the purpose and intent of this chapter to discuss the theoretical and mathematical models necessary to describe complex vibrations; this information is presented in several well-written scientific and engineering texts.

VIBRATION MEASUREMENTS

Basic Elements In a Measurement System

A wide variety of component systems, consisting of mechanical or a combination of mechanical, electrical and optical elements are available to measure vibration. The most common system uses a vibration pick-up to transform the mechanical motion into an electrical signal, an amplifier to enlarge the signal, an analyzer to measure the vibration in specific frequency ranges and a metering device calibrated in vibrational units.

Vibration Pick-ups

The vibration pick-up measures the displacement, the velocity, or the acceleration of the vibration. These parameters are all inter-related by differential operations so that it does not normally matter which variable is measured. If the result is desired in terms of velocity or displacement, electronic integrators are added at the output of an accelerometer, an instrument which measures the acceleration of a mass due to the vibration signal.

Accelerometers, the most common type of vibration pick-up, are normally smaller than velocity pick-ups and their useful frequency range is wider. An accelerometer is an electromechanical transducer which produces an output voltage signal proportional to the acceleration to which it is subjected.

The most common type of accelerometer is the piezoelectric type, such as the one shown in Figure 26-11, in which two piezoelectric discs produce a voltage on their surfaces due to the mechanical strain on the asymmetric crystals which make up the discs. The strain is in the form of vibrational inertia from a moving mass atop the discs. The output voltage is proportional to the acceleration, and thus to the vibration signal. The upper limit of the accelerometer's useful frequency range is determined by the resonant frequency of the mass and the stiffness of the whole accelerometer system. The lower limit of the frequency range varies with the cable length and the properties of the connected amplifiers. The accelerometer's sensitivity and the magnitude of the voltage developed across the output terminals depends upon both the properties of the materials used in the piezoelectric discs and the weight of the mass. The mechanical size of the accelerometer, therefore, determines the sensitivity of the system; the smaller the accelerometer, the lower the sensitivity. In contrast, a decrease in size results in an increase in frequency of the accelerometer resonance and thus, a wider useful range.[20]

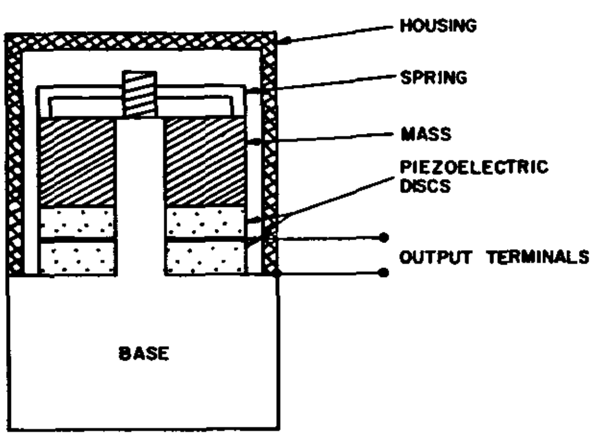

Figure 26-11. Sketch Showing the Basic Construction of the Bruel & Kjaer Compression Type Piezo-Electric Accelerometers (Single-Ended Version).

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Other factors to consider in the selection of a suitable accelerometer include:

1. The transverse sensitivity, which is the sensitivity to accelerations in a plane perpendicular to the plane of the discs.
2. The environmental conditions during the accelerometer's operation, primarily temperature, humidity, and varying ambient pressure.

Often two types of sensitivities are stated by the manufacturer — the voltage sensitivity and the charge sensitivity. The voltage sensitivity is important when the accelerometer is used in conjunction with voltage measuring electronics, while the charge sensitivity, an indication of the charge accumulated on the discs for a given acceleration, is important when the accelerometer is used with charge-measuring electronics.

Preamplifier

The preamplifier is introduced in the measurement circuit for two reasons:

1. To amplify the weak output signal from the accelerometer; and
2. To transform the high output impedence of the accelerometer to a lower, acceptable value.

It is possible to design the preamplifier in two ways, one in which the preamplifier output voltage is directly related to the input voltage, and one in which the output voltage is proportional to the input charge, with the preamplifier termed a voltage amplifier or charge amplifier, respectively.

The major differences between the two types of amplifiers rest in their performance characteristics. When a voltage amplifier is used, the overall system is very sensitive to changes in the cable lengths between the accelerometer and the preamplifier, whereas changes in cable length produce negligible effects on a charge amplifier. The input resistance of a voltage amplifier will also affect the low frequency response of a system. Voltage amplifiers are generally simpler than charge amplifiers, with fewer components and are, therefore, less expensive. In selecting the appropriate preamplifier for vibration work, these above-mentioned factors should all be considered.

Analyzers

The analyzer element in a vibration measuring arrangement determines what signal properties are being measured and what kind of data can be obtained in the form of numbers or curves.

The simplest analyzer consists of a linear amplifier and a detection device measuring some characteristic vibration signal value such as the peak, root mean square, or average value of the acceleration, velocity, or displacement. The phase response of a system must be taken into account, in addition to the frequency response in choosing the correct analyzer. A signal reaching the detection device may look completely different from the signal input to the linear amplifier, due to possible distortions during linear attenuation of a complex signal's various frequency components. Serious waveshape distortion of the signal may be avoided when the fundamental vibration frequency is higher than 10 times the low frequency limit of the measuring system, and the highest significant vibration frequency component has a frequency which is lower than 0.1 times the high frequency limit of the system.[20] Phase response can be disregarded for measurement of only RMS values.

In most practical cases it will be necessary to determine the frequency composition of the vibration signal, which is done with a frequency analyzer. Two types of frequency analyzers are commonly available, the constant bandwidth analyzer and the constant percentage bandwidth-type analyzer. If the vibrations are periodic, the constant bandwidth-type analyzer is preferred because the frequency components are harmonically related. If the signal is not quite stable and only the first few harmonics are significant, such as characteristic of a shock signal, a constant percentage bandwidth-type analyzer is suitable. If the vibration signal is random in nature, the type of analyzer will depend on the use to be made of the measurement as well as the frequency spectrum itself.

If the vibrations are periodic, the frequency spectra are presented in terms of the RMS value of the signal, defined earlier.

In random vibrations, the RMS-value fluctuates during measurement because the averaging time is limited to a practical time limit and the time of observation is greater than the averaging time of the instrument. A decrease in averaging time will cause considerable fluctuations in the RMS values.

Vibration Recorders

Many types of metering or recording devices are available to exhibit the analyzer output. These are divided into three classes: the strip-chart recorder which prints the output on preprinted calibrated recording paper; the vibration meter which indicates some characteristic value (RMS, peak, etc.) on a precalibrated scale; and an oscilloscope, which projects the wave form on a screen for analysis and measurement.

When a strip-chart recorder is used, the averaging time is determined by the writing speed of the recording pen, the input range potentiometer, and other internal properties of the recorder instead of observation time. The choice of averaging time affects the recording of a random vibration frequency spectrum.

The oscilloscope presents measurements as a function of time, and makes possible the study of instantaneous values of vibration. An oscilloscope with slow sweep rates, long-persistence screen, and a DC amplifier is recommended for most studies.

Accessories

The accessories used with vibration meters are determined by customer needs and usually can be built into any specified instrument. Integrators to interconvert displacement, velocity, and acceleration can be specified in the basic circuit design of measuring devices. Often, a series of ranges can be requested for a frequency analyzer, depending on the accuracy of the measurements required.

Additional analyzers including the third-octave-bandwidth, tenth-octave-bandwidtth, one-percent-bandwidth, and a wave analyzer are available. Different recorders, dependent on desired output, can be used in conjunction with any analyzer. For peak or high speed impacts, the signal can be recorded on tape and then replayed at varied speeds to give the optimum measurement on an electric output signal.

A calibrator is an important accessory for checking the overall operation of a vibration-measuring system. The common calibrator is nothing more than an accelerometer, driven by a small, electromechanical oscillator mounted within the mass, operated at a controlled frequency and RMS acceleration. Provisions can be made for additional calibration of transducers, insert voltage, and reciprocity.

Field Measurements

The arrangement of the vibration equipment for typical field use is shown in Figure 26-12. Sources of error in field measurements must be recognized by the investigator and avoided or minimized, at least. Common sources of error are incorrect mounting, incorrect calibration, connecting cable noises, and thermal effects. In the field, consideration must be given to the location and mounting of the accelerometer in order to record the original motion and resonant frequencies of the structure. The vibration transducer should load the structural member as little as possible, since any loading effects will invalidate the measurement results. In most practical situations, the mass loading effect is negligible but can be checked by the formula:

$$AR = AS\left(\frac{MS}{MS + MA}\right)$$

where AR = Response of structure with accelerometer

AS = Response of structure without accelerometer

MS = Weight of the structure member to which the accelerometer is attached

MA = Weight of the accelerometer

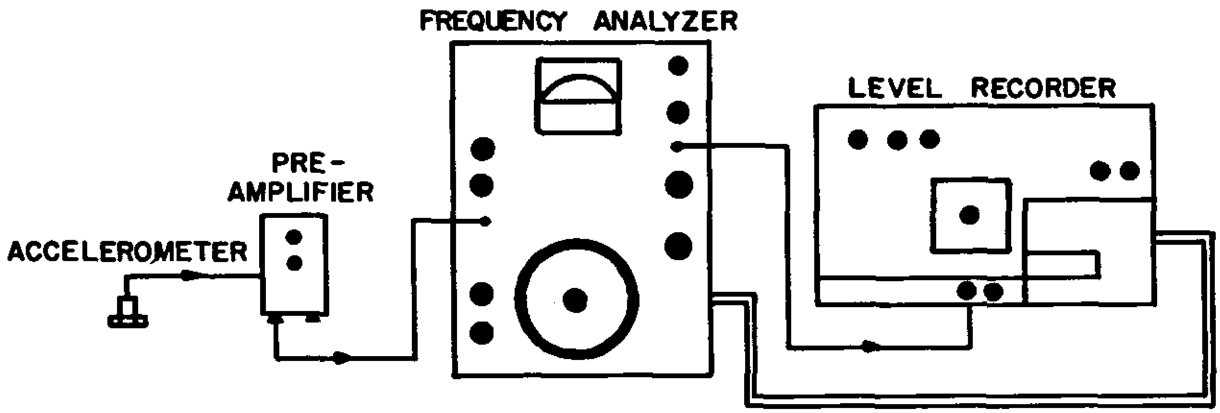

Figure 26-12. Arrangement of Equipment for Automatic Frequency Analysis.

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A number of methods can be used to mount the accelerometer, including mounting by means of a steel stud, an isolated stud and mica washer, a permanent magnet, wax or soft glue, or handheld with a probe. Use of a steel stud gives the best frequency response, approaching the actual calibration curve supplied with the accelerometer. When electrical isolation is required between the accelerometer and vibrator, a mica washer is used because of its hardness and transmission characteristics. A permanent magnet also gives electrical isolation, but magnetic mass, and temperature effects on the accelerometer can be introduced. Handheld probes are convenient but should not be used for frequencies higher than 1000 Hz where mass loading effects begin. Wax, because of its stiffness, gives a good frequency response but should not be used at high temperatures.

Another source of error, cable noise, originates either from mechanical motion of the cable or from ground loop-induced electrical hum and noise. Mechanical noises originate from local capacity and charge changes due to dynamic compression and tension of the cable, affecting low-frequency readings. A ground loop (see Figure 26-13) induces a current and small voltage drop which adds directly to the sometimes weak accelerometer signal. Insuring that grounding of the installation is made only at one point eliminates ground looping.

In ordinary vibration measurements, temperature effects need not be considered. However, measurements of very low frequency, very low amplitude vibration will be disturbed by even small temperature changes due to the variance of the accelerometer output at a rate determined by the time constant of the accelerometer preamplifier input circuit.

Calibration of a typical vibration measuring arrangement can be made directly from curves and figures supplied by the manufacturer. Special measuring arrangements can be calibrated using a vibration calibrator.

The following general outline points out the important considerations in a field measurement:

1. Determine placement of the vibration transducer with consideration for possible mass loading effects.
2. Estimate the types and levels of vibrations likely at the mounting point.
3. Select a suitable vibration transducer considering the mass loading effect, types of vibrations, temperature, humidity, acoustic and electric fields.
4. Determine what type of measurement would be most appropriate for the problem at hand.
5. Select suitable electronic equipment, considering frequency and phase characteristics, dynamic range and convenience.
6. Check and calibrate the overall system.
7. Make a sketch of the instrumentation system, including types and serial numbers.
8. Select the appropriate mounting method, checking vibration levels, frequency range, electrical insulation, ground loops, and temperatures.
9. Mount the accelerometer, carry out the measurements, and record the results. Record any octave band analysis of the vibrations, if necessary.
10. Note the setting of various instrument control knobs.

The apparent vibration level, similar to a background reading, should be recorded by mounting the accelerometer on a nonvibrating object in the measured system. Apparent vibrations should be less than one-third of the measured vibrations for accuracy in the actual vibration measurements.

Analysis should include measurements at different points on the machine or system to determine the areas of greatest vibration and their

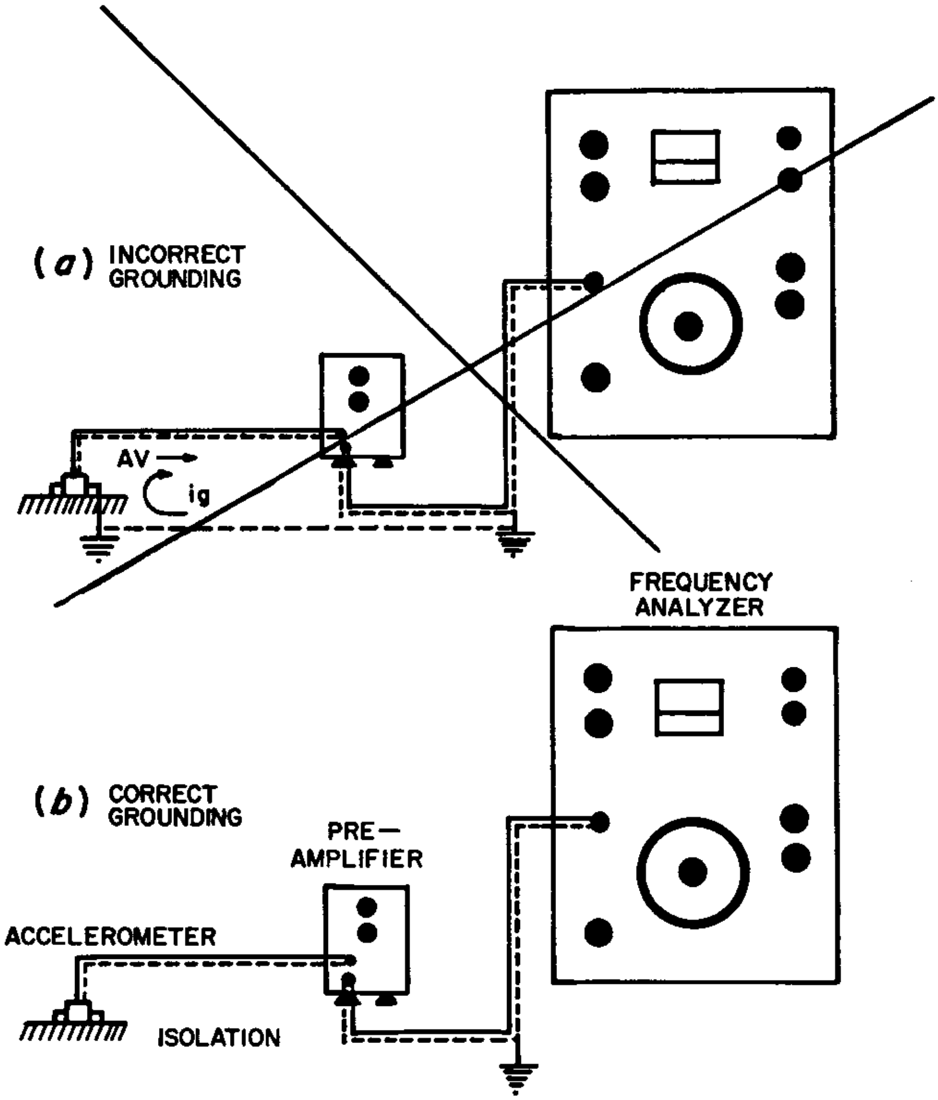

Figure 26-13. Illustration of Ground-Loop Phenomena.
a) This method of connection forms ground loop and should be avoided.
b) No ground-loop is formed. Recommended method of connection.

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respective frequency components. Experience is the best guide in pinpointing vibration sources in familiar machinery. Since machine vibrations produce noise, reduction of vibration often reduces noise problems as well.

CONTROL OF VIBRATION

Vibrations may be reduced by:
1. Isolating the disturbance from the radiating surface;
2. Reducing the response of the radiating surface; and
3. Reducing the mechanical disturbance causing the vibration.

Isolation

Vibration isolators effectively reduce vibration transmission when properly installed. The vibration of a machine on isolators is complex, since the machine can move along or rotate about its one vertical and two horizontal axes. The machine will have a resonant frequency for each of the six modes of vibration; thus, it is important that none of the six possible resonances occur at the frequency of the disturbance. To insure adequate isolation, the resonant frequency of the isolator should be less than half the disturbing frequency.

For the vertical mode, which is usually the most important, the resonant frequency of the isolator is related to its static deflection under the weight of the machine by the following formula:

$$f = \frac{3.13}{\sqrt{d}}$$

where d is the static deflection, inches
f is the frequency, Hertz

To find the stiffness required from the isolation mount (spring) when the desired frequency has been determined, the formula:

$$K \approx 0.04 \times MS \times F_o^2 \quad \text{kg/cm}$$

can be used.

K = Spring constant
MS = Machine weight
F_o = Resonant frequency of the machine and isolation mount system.

Since the degree of isolation increases as the resonant frequency is lowered, the equation indicates that isolation is improved as the static deflection is increased.

Transmission of vibration to an adjacent structure may also be reduced by making electrical connections less rigid. For piping, electrical connections, and ductwork flexible connectors may be used. When using enclosures, the housing should be anchored to the floor rather than the machine to eliminate transmission of the vibration to the enclosures.

Another important consideration is that the vibration isolators be placed correctly with respect to the motion of the center of gravity of the machine, with the center of gravity located as low as possible. In cases of instability, i.e., "rocking", the effective center of gravity may be lowered by first mounting the machine on a heavy mass and isolating the mass from the machine.

Reduction of Surface Response (Damping)

In cases where isolation of the vibrations is not suitable, or is difficult to arrange, the principle of vibration absorbers may be used. By attaching a resonance system to the vibrating structure (which counteracts the original vibrations), the structure vibrations can be eliminated. Through mathematical differential equations it can be shown that by tuning the absorber system resonant frequency, it is theoretically possible to eliminate the vibration of the machine.

Structural elements like beams and plates exhibit an infinite number of resonances. When subjected to vibrations of variable frequency, the application of separate dynamic absorbers to structural elements becomes impractical. Since there is usually little inherent damping of resonant vibrations in the structural elements themselves, external arrangements must be made to reduce vibrations.

External damping can be applied in several ways:

1. By means of interface damping (friction)
2. By application of a layer of material with high internal losses over the surface of the vibrating element
3. By designing the critical elements as "sandwich" structures.

Interface damping is obtained by letting two surfaces slide on each other under pressure. With no lubricating material the "dry" friction produces the damping effect, although this commonly causes fretting of the two surfaces. When an adhesive separator is used, a sandwich structure results, a concept which will be discussed later.

Mastic "deadeners" made of an asphalt base are commonly sprayed onto a structural element in layers, to provide external dampening. These deadeners are commonly made from high-polymer materials with high internal energy losses over certain frequency and temperature regions. To obtain optimum damping of the combination structural element and damping material, not only must the internal loss factor of the damping material be high, but so must its modulus of elasticity (the ratio of stress to strain).

An approximate formula (see Figure 26-14) governing the damping properties of a treated panel is given by:

$$N \approx 14 \left(\frac{N_2 E_2}{E_1}\right)\left(\frac{d_2}{d_1}\right)^2$$

N = Loss factor of the combination structure element + damping material
N_2 = Loss factor of the damping material
E_1 = Modulus of elasticity of the structural element
E_2 = Modulus of elasticity of the damping material
d_1 = Thickness of the structural element
d_2 = Thickness of the damping material layer.

The ratio $\left(\frac{d_2}{d_1}\right)^2$ is the most important factor, usually around 3:1 for best results.

The third method of damping is the use of sandwich structures; a thick or thin layer of viscoelastic material is placed between two equally thick plates, or a thin metal sheet is placed over the viscoelastic material which is covering the panel. The damping treatment will be more effective if applied to the area where vibration is greatest, but the actual amount of treatment and the area of coverage are best determined by experiment. As a rule of thumb, the amount (density × thickness) of material applied should equal that of the surface to which it is applied. Since this can lead to using large amounts of damping material, the method of covering the damping material with a sheet metal overlay is often used. Studies[21] show that to a certain extent, the thickness of a layer is not the most important factor; but in cases of sandwich layers, symmetry is most significant. Sandwich layers often are preferred due to a greater dampening factor (Figure 26-14) than with single layer coatings.

Reduction of Mechanical Disturbance

Mechanical disturbances that produce vibration can be reduced by reducing impacts, sliding or rolling friction, or unbalance.

In all cases, either mechanical energy is coupled into mechanical-vibratory energy, or energy in some other form is transformed into mechanical-vibratory energy. These other forms of energy include: varying electrical fields, varying hy-

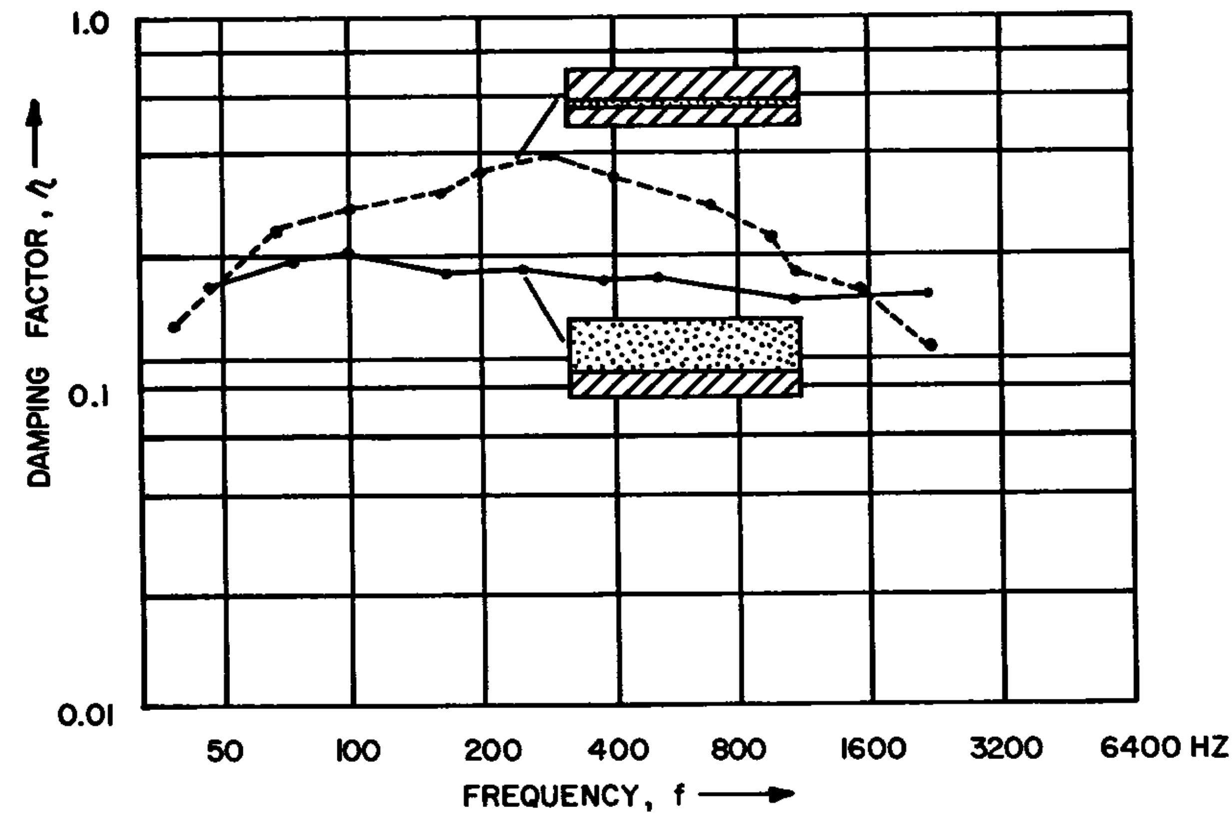

Figure 26-14. Results of Loss Factor Measurements on a Sandwich Structure with a Thin Visco-Elastic Layer, and on a Plate Supplied with One Layer Mastic Deadening ($d_2/d_1 \approx 2.5$). After Cremer and Heckl.[21]

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draulic forces, aerodynamic forces, acoustic excitation, and thermal changes.

Often mechanical vibration can be reduced by: (1) proper balancing of rotating machinery, (2) reducing response of equipment to a driving force, and (3) proper maintenance of machinery.

Limitations of Control Methods

In conjunction with the practical applications of isolators and dampers, certain limitations should be noted in the control of vibration:

1. Reduction in transmissibility can only take place by allowing the isolator to deflect by motion. Thus, certain space clearances must be provided for the isolated equipment.
2. If the resonant frequency of the isolation system is chosen incorrectly, the isolator may actually amplify the destructive characteristics. Select a spring mounting so that the natural frequency, F_0, of the spring-mass system is considerably (at least one-half) lower than the lowest frequency component in the force system produced by the machine.
3. If the isolator produces unexpected nonlinear characteristics a great number of extra response effects may take place.

The reduction in shock severity which may be obtained by the use of isolators results from the storage of the shock energy within the isolators and its subsequent release in a "smoother" form. Unfortunately, since a shock pulse may contain frequency components ranging from nil to near infinity it is not possible to avoid excitation of the isolator-mass system.

Damping is a costly method of reducing the response of a radiating surface, and is therefore generally avoided. Mechanical disturbances, primarily dependent upon an effective maintenance program, are rarely eliminated completely.

In conclusion, all vibration problems should be approached by determining first if a quick, simple, and "common sense" solution is available. If a simple answer is not obvious, the quantitative results of measurements become essential in the analysis and solution of the problem. As various control procedures are tried, vibration measurements can be used to show the progress being made and predict the correct steps in reducing vibrations further.

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