THE JOURNAL OF THE ACOUSTICAL SOCIETY OF AMERICA VOLUME 33, NUMBER 11 NOVEMBER, 1961

Calibration of Vibration Pickups at Large Amplitudes

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(Received July 18, 1961)

Axial resonances of long rods and tubes were used to generate motion for accurate calibration of vibration pickups over the frequency range from below 1 to above 20 kc at acceleration levels up to 12 000g. The resonators were driven by an electromagnetic shaker at low frequencies and by a piezoelectric ceramic stack shaker at high frequencies. Vibration amplitude was measured optically by means of a microscope using stroboscopic light and by means of the interference fringe disappearance technique. Adequate overlap between the two methods was achieved by going up to the 60th disappearance of the fringes. A simple, direct measurement of the phase angle between the pickup signal and the motion is described. Construction details of a small, light pickup which is unaffected by the high acceleration levels are given.

INTRODUCTION

A PROGRAM of vibration pickup calibration at high levels of sinusoidal acceleration arose from a need to determine if the pickup sensitivity is linear over a large range of amplitude and the extent to which a pickup can undergo sustained large acceleration forces without structural damage or change of sensitivity. Pickups which are unaffected by large accelerations are suitable for studies of mechanical fatigue and failure, studies of the vibrations of high speed rotating machinery and of projectiles, accelerated life and environmental tests, and studies of the effectiveness of vibration isolators and mounts. Their use for measurements in which the effects of sinusoidal excitation are compared with the effects of shock are more informative if the phase lag between the mechanical vibration and the output signal from the pickup is known. One measurement of phase lag is reported here to show how it can be made, but phase information was not required for our calibrations.

VIBRATION GENERATORS

The accelerations for most of the tests in the paper were generated by piezoelectric ceramic shakers constructed in our laboratory. Each consists of a driver element cemented between a heavy base of brass or steel and a top, usually steel, or sometimes aluminum. The driving element is a number of lead-zirconium-titanate discs cemented together with a conducting epoxy resin which acts both as cement and electrode. The behavior and construction details of these shakers will be described in a later paper.

Motion for a few of the lower frequencies was generated by a modification of an electromagnetic shaker which was described in a previous paper.[1]

A major factor limiting the accuracy of pickup calibration is the problem of generating uniaxial, undistorted sinusoidal motion. Much of the unwanted noise, distortion, and transverse motion of the shake table can be minimized through the insertion of a mechanical filter between the shaker and the pickup it drives.

In most of the calibrations for this study such a filter was formed by a slender rod or tube driven at one of its longitudinal resonances. The amplification of the motion by the resonant rod allowed a given amplitude to be reached with a reduction in the noise and distortion generated by the shaker and associated driving circuits. The amplification derived from the mechanical resonant system also reduced the brute force required from the power amplifiers.

A brief survey of possible resonant system types led to our selection of tubes and rods in longitudinal resonance for use in these studies. Although usable results have been reported,[2,3] our study showed that with any type of bending system it is difficult to generate large vibration amplitudes precisely along any preferred axis except at low frequencies. Modes above the fundamental are useless because, unlike longitudinal systems, bending systems have twisting and rotating effects difficult to control that increase with mode order. The problem of off-axis motion is less severe with longitudinal systems.[4–8] First, the bending modes in the frequency range of any longitudinal resonance, fundamental included, of a slender rod are of high order and attenuated. Second, there is seldom coincidence between the two mode types either with respect to frequency or node location. Therefore, a small transverse damping constraint at a longitudinal node usually suffices to eliminate transverse motion with little effect on longitudinal motion.

The shorter rods and tubes were usually attached upright to a solidly supported ceramic shaker and

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viewed in vertical vibration. Considerable care was required to suspend the rather flexible, longer units so that the rod was held with its axis unchanging and with longitudinal motion unhampered. Our choice was to orient shaker and rod horizontally, the shaker resting on a fixed support. About midway of the rod a rigid upright support held a horizontal beam over the rod free to pivot about a horizontal axis normal to the rod. At each end of the horizontal beam two smaller beams were attached, each also free to pivot about a horizontal axis normal to the rod. From the four ends of the smaller beams flexible cords dropped to support the rod. This method provided equally distributed support to four points along the rod. Attachment of the cords to points which were as near as possible to longitudinal nodes and transverse antinodes served further to constrain transverse motion. One of the smaller beams can be seen in upper left of Fig. 1.

Important considerations in choosing rods and tubes for resonators are uniformity of cross section and lack of curvature. In order to calibrate over a wide frequency range a set of resonators is necessary, although it is not required to have a separate resonant element for each test frequency. A given resonator can be used at many of its higher order harmonics. Tubes have lower resonant frequencies than rods of the same length and mechanical load and are used at the lower frequencies where space becomes important.

The rods were selected from hex aluminum alloy stock either $\frac{5}{8}$ or $\frac{1}{2}$ in. in diameter, one end machined for attachment to a shaker and the other machined flat for pickup attachment. The tubes were constructed from sections of stock aluminum alloy tubing 1 in. o.d., $\frac{1}{32}$ in. wall thickness having a shaker adapter on one end, and a pickup adapter with area enough to mount a small mirror for interferometric measurements next to the pickup on the other. The adapters were about 1 in. long cut from 1 in. hex aluminum and cemented to the squared-off tube with epoxy cement.

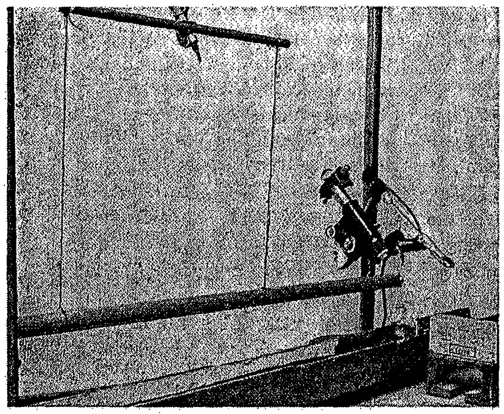

FIG. 1. Stroboscopic microscope measurement on horizontally supported resonator.

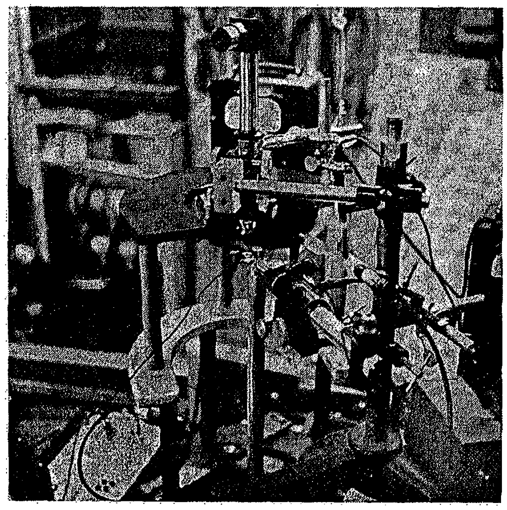

FIG. 2. Simultaneous interferometric and stroboscopic measurements on vertical resonator.

A number of tubes and rods were prepared with different lengths to constitute a set of resonators for calibrating at many points over a wide frequency range.

The power amplifiers used to drive the shakers are standard heavy duty hi-fi amplifiers. Since these amplifiers are designed to power low impedance loads, we use audio output transformers as step-up transformers to drive piezoelectric shakers. The shakers are essentially capacitors and become low impedance loads at the higher frequencies. Inductors of a value to present the amplifier an electrically tuned circuit at a given frequency are connected in parallel across the shaker terminals. In addition to permitting maximum driving voltage for a particular frequency, tuning serves to filter out distortion in the driving signal.

PICKUP ATTACHMENT

For high acceleration-level calibration considerable care is required in attaching test pickups. Failure to seat the pickup base firmly against the driving surface can result in chatter or loosening of the pickup under sustained vibration. Some pickups have screw mounting studs large enough for adequate torquing. Pickups with flat bottoms can be cemented to the driving surface. This seems to provide the best attachment.

PROCEDURE

In all our calibrations the sinusoidal acceleration level was determined by measuring the displacement amplitude of vibration. The frequency of vibration was determined from Lissajou comparisons with NBS standard frequencies or by electronic frequency counters. Acceleration amplitudes were computed from frequency and displacement.

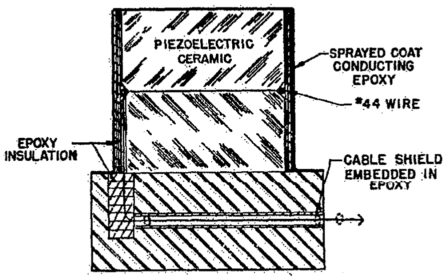

FIG. 3. Construction of pickup designed to withstand high acceleration levels.

Most displacement measurements were made by means of a microscope, with a micrometer eyepiece viewing a spot on the base of the pickup under stroboscopic illumination as it vibrated at constant amplitude. A description of this method, shown in Fig. 1, has been published.[1] For large vibration displacements the method of measuring motion directly with a microscope is well suited for the accurate determination of amplitudes. Also, it serves the equally useful function of determining precisely the direction of shaker motion. With a distortion check of the output waveform this method affords a complete description of large amplitude sinusoidal vibration.

The interference-fringe-disappearance method was used to measure the smaller displacements. A description of this method has also been given.[1] An overlap between the two calibration methods was secured by making an interferometer calibration past the 40th disappearance which corresponds to a displacement that the microscope method can measure accurately. To keep the setup physically small enough for interferometer use at a frequency low enough to measure displacements corresponding to 1g, a tube resonator was used. The interferometer microscope was focused on the fringe pattern formed above the small mirror cemented to the surface carrying the test pickup. The direct view strobe microscope was focused to measure the motion of a spot on the base of the pickup. Figure 2 shows how a simultaneous calibration was made. On a few occasions interferometric calibrations have been made up to the 60th disappearance. Due to the tediousness of calibrating at large orders of disappearance, the simultaneous check on sensitivity was not made at all frequencies after it was found that the two methods agreed.

PICKUPS CALIBRATED

High g calibrations have been made on commercial-type accelerometers and on pickups constructed in our laboratory. A homemade piezoelectric pickup was selected for most of this work since we were certain of its ability to stand up under a long series of high accelerations. This pickup is a cemented type similar in construction to our ceramic shakers.

Figure 3 is a diagram of the pickup which is 0.250 in. in diameter, 0.300 in. in height, and weighs 1.5 g. Its unmounted resonant frequency is about 150 kc. The capacitance of the pickup and 5 in. of permanently attached connector cable is 120 $\mu\mu f$. Sensitivity is about 3.5 mv/g. It consists of a circular cemented piezoelectric-ceramic sandwich cemented to a stainless steel base. The adhesive is conducting epoxy. The pickup cable passes through the base to which the shield is both electrically and mechanically anchored. The center conductor connects through a shallow groove in the lower disc to the conducting interface of the sandwich. A coating of plain epoxy insulates the center lead at all other points. The unit is shielded with a sprayed-on coating of conducting epoxy. No inertial mass other than the ceramic is used in this design.

TABLE I. Phase angle between pickup CB-1 signal and displacement.

Frequency (cps)	Phase angle by which signal lags displacement (deg)
2500	5–8
3000	10
3250	15
3500	40
3750	60

Resonant frequency—3900 cps

PHASE MEASUREMENT

Information on phase lag between pickup output and motion was not needed in this study but might be required in other applications. Two sets of phase lag measurements were made during these experiments. One set was made on the pickup which was used for most of the high acceleration level calibrations and showed no measurable lag in the frequency range of calibration. The other set of phase lag measurements was made on a larger pickup of the cemented ceramic type which had a large mass and weaker equivalent spring. The latter pickup was designed to have a comparatively low resonant frequency so that significant values of phase lag could be found at audio frequencies. These measurements are given in Table I.

The phase measurements were made during a strobe microscope calibration. Using the circuit shown in Fig. 4, strobe flashes, triggered by the signal used for

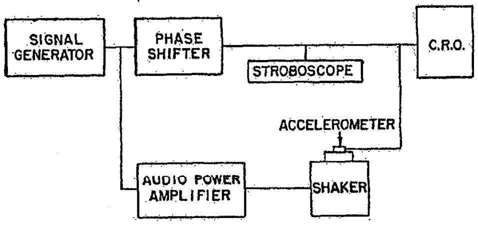

FIG. 4. Electronic circuit for stroboscopic measurement of displacement amplitude and phase.

Table II. Maximum acceleration level and resonator at a number of frequencies.

Calibration frequency 830	Maximum acceleration level peak g 2000	*C*	Resonator Shape Round tubing 1 in. o.d. $\frac{1}{32}$ in. wall	Material alum alloy 2024-T3	Length 52 in.
1400	5000	*E*	$\frac{5}{8}$ in. hex rod	2017-T4	35 in.
2600	6000	*H*	$\frac{5}{8}$ in. hex rod	2017-T4	18 in.
4300	8000	*E*	$\frac{5}{8}$ in. hex rod	2017-T4	35 in.
10 000	6000	*E*	$\frac{5}{8}$ in. hex rod	2017-T4	35 in.
13 000	6000	*E*	$\frac{5}{8}$ in. hex rod	2017-T4	35 in.
19 000	11 000	*F*	$\frac{5}{8}$ in. hex rod	2017-T4	18 in.
20 000	10 000	*B*	$\frac{1}{2}$ in. hex rod	2017-T4	20 in.
22 000	12 000	*G*	$\frac{5}{8}$ in. hex rod	2017-T4	27 in.

driving the shaker, were phased to illuminate the pickup at a displacement extreme as viewed by the microscope. Signals from these pulses were superimposed on the pickup output signal and displayed on an oscilloscope as in Fig. 5. The phase lag was the adjustment on the calibrated phase shifter in Fig. 4 required to place the strobe pips at the peaks of the pickup sine wave signal. Similar phase measurements have been made at higher frequencies and smaller amplitudes. The apparent motion of interference fringes formed by stroboscopic light was used to detect the extremes of displacement instead of the apparent motion of the pickup.

These measurements indicate that over the frequency range for which its response is reasonably flat, phase lag considerations are negligible for a piezoelectric-ceramic vibration pickup.

RESULTS

Figure 6 shows the range of accelerations through which a pickup was calibrated at one frequency. The pickup was mounted on a resonant tube in a way to permit the two types of measurements shown in Fig. 2.

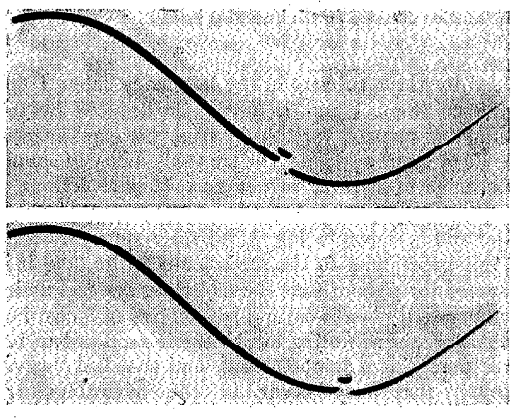

Fig. 5. Oscilloscope trace, strobe pulse superimposed on accelerometer output signal. Upper trace shows pip on pickup signal at time of extreme displacement of pickup. Lower trace shows pip shifted to extreme value of signal by calibrated phase shifter. The phase angle introduced by the phase shifter to move pip from position in upper trace to position in lower trace is phase lag of signal behind motion.

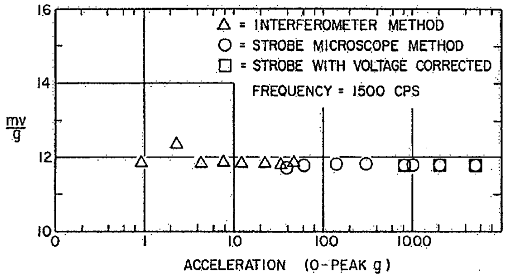

Fig. 6. Sensitivity vs acceleration of 2-9 accelerometer.

The higher acceleration points are labeled "corrected" because the pickup signal was too high for the meter circuit cathode follower, and signal attenuation was required ahead of the cathode follower. Table II shows the high acceleration levels obtained over the frequency range 800 to 20 000 cps.

In general, rods are used for higher frequencies and tubes at lower frequencies. It is to be noted that a given resonator is often usable at several of its harmonics, as well as in its fundamental mode.

SIDELIGHTS

It may be in order to mention a few sidelights of our experience in high g level calibrating. Sometimes a transverse resonance is mistaken for or lies close enough to be excited along with a longitudinal mode. The mistaken identity is usually detected acoustically in short order. In place of pure sine tones there is a build up of loud noises, and the slender rods whip about in a violent manner.

Sometimes attempts to raise the amplitude of a particular longitudinal resonance sets off a transverse mode many octaves below the driving frequency. The low frequency mode gradually builds up and takes over sufficiently to damp the longitudinal mode to a level where it no longer excites the low frequency mode, and the latter diminishes. The longitudinal resonance again builds up, and the whole sequence repeats every few seconds, as in the textbook description of coupled modes.

We have noticed that if the driving signal is ultrasonic and everything else is quiet, sliding the hand over antinodes produces mouse-like squeaks. The antinodes have a warm silky touch as though coated with oil.

The standing wave pattern of a rod tends to force contacting objects towards its nodes. If a wheel is moved along the rod from node to node, it will spin rapidly first in one direction and then another, always toward a node.

CONCLUSIONS

We have found it possible and convenient to calibrate vibration pickups at thousand g levels over a wide frequency range. It is also possible to establish amplitude linearity checks over a range of nearly 10.[4]

Many people, using other methods, have attained larger amplitudes of vibration than those reported here. Levels up to 100 kg were reported for the equipment described in reference 8 and similar equipment made by the Sandia Corporation, Livermore, California.[9] When amplitudes greater than those reported here were reached on our resonators we found that the motion was no longer undistorted and uniaxial. Some of the departure from good motion was due to the effect of the pickup and its mounting aided by the unbalancing effect of the cable. Some was caused by incipient nonlinearity in the rod and in the power amplifier. Thus the acceleration levels reported here are the highest our present equipment allows with proper regard for accuracy and precise control.

For any of our points the errors are estimated to be no greater than 2%. A program is presently underway to improve this figure. Many pickups are described as not being affected by high acceleration levels applied as shocks. We have found that some are damaged by lower levels applied as sustained vibration.

ACKNOWLEDGMENT

This work was sponsored by the Special Projects Office, Department of the Navy.

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THE JOURNAL OF THE ACOUSTICAL SOCIETY OF AMERICA VOLUME 33, NUMBER 11 NOVEMBER, 1961

Reduction of the Response to Vibration of Structures Possessing Finite Mechanical Impedance

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(Received July 3, 1961)

Expressions have been derived from which the isolation provided by a simple mounting system when supported by any nonrigid foundation may be determined, if the mechanical impedance of the foundation is known. As an example, the mechanical impedance of a damped beam at its midpoint has been employed to simulate the behavior of such a foundation. Knowledge of the dynamic mechanical properties of natural rubber and a high-damping synthetic rubber has enabled the behavior of antivibration mountings to be described realistically.

If the ratio of the mass of the mounted item to the mass of the foundation is large, the isolation afforded by a mounting is found to be much less than that predicted by its transmissibility curve, which relates to an ideally rigid foundation. It is shown, however, that the isolation provided by a mounting can be increased significantly at high frequencies if an additional mass is employed to load the foundation, the greatest isolation then being provided by a low-damping rubber such as natural rubber. Damping of the foundation is found to have little influence upon the over-all level of the isolation afforded by the mounting system.

1. INTRODUCTION

THIS paper describes a theoretical investigation concerned with the isolation of machinery vibration from structures possessing finite mechanical impedance. General equations are presented from which the response ratio[1] (Sec. 2) of simple mounting systems may be computed when the variation with frequency of the mechanical impedance of their foundation is known.

Particular attention has been devoted to representing realistically the dynamic mechanical properties of rubberlike materials employed as vibration isolators. The mechanical properties of natural rubber (vulcanized Hevea) and a synthetic rubber (Thiokol RD) have been considered to typify the mechanical properties of low- and high-damping rubbers, respectively. The dynamic shear modulus G_ω and damping factor δ_ω possessed by these rubbers in the frequency range 1 cps to 10 kc (Figs. 1 and 2) have been deduced by the method of reduced variables[2–4] from the experimental results of other workers.[3] The transmissibility of simple mountings of natural rubber and Thiokol RD at a temperature of 20° C has been computed from these data[4] and is shown in Fig. 3. All the results presented in this paper refer to the same temperature, and it is assumed throughout that the mounting systems under consideration possess natural frequencies of 5 cps. For simplicity, the vibrating machinery is supposed to behave purely as a lumped mass, and so-called "wave-effects" that may occur in the mountings are disregarded.

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