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PROTECTION AGAINST
SHOCK AND VIBRATION

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Protection Against Shock and Vibration*

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The low mass of subminiature electronic components probably accounts for the erroneous idea that these components do not experience severe and frequently damaging shock forces. "Trade-offs" between shock protection, vibration isolation and space requirements are particularly difficult in components which have moving parts. In these critical components, the maximum forces due to shock may exceed the minimum operating forces by 10^8. Analysis and measurements of hearing aid shock environments, and the protection of hearing aid transducers from damage due to these shocks are presented.

INTRODUCTION

THAT shock and vibration must be considered in design of military electronic equipment is well known.[1] Where the customer's comfort is recognized through product acceptability, consumer product engineers have concerned themselves with those aspects of shock and vibration pertaining to noise control and have assigned to packaging engineers the task of getting the product to the consumer intact. In miniaturized highly portable electronic equipment intended for personal use, design engineers have to assume responsibility for mechanical robustness in addition to functional performance. Examples are portable radios, portable tape recorders and hearing aids. The fact that the equipment is smaller and uses smaller components than its standard counterpart does not reduce its susceptibility to malfunction from excessive vibration and shock. As a matter of fact, in the case of hearing aids, as a result of significant technical advances enabling remarkable size reductions, vibration and shock isolation have become more important and present more difficult problems to solve. More compliant supports for transducers are used to achieve high gain without vibration feedback oscillation. User emphasis on small size and appearance has led the hearing aid designer to provide less space for the additional shock protection needed to protect components. Analysis of hearing aid component protection provides an excellent basis for review of the requirements of miniature component shock and vibration protection.

A common field failure mechanism in hearing aids is accidental drop. Components within electronic equipment which are not required to move can be made very shock resistant. Other components, notably, transducers, must move. A good hearing aid microphone reproduces sound pressures as low as 30 db re .0002 microbars. In a typical miniature microphone the force on the moving element from this pressure is on the order of 2×10^8 Newtons

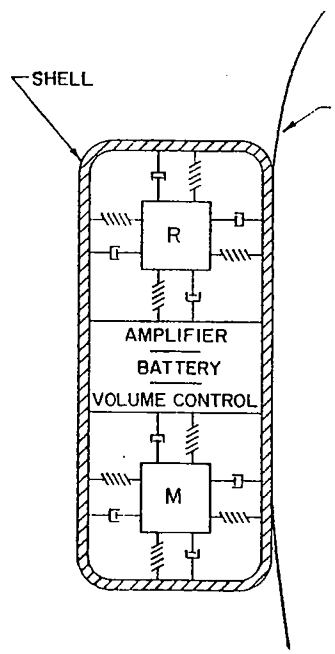

FIG. 1. Schematic of a head-worn hearing aid.

(2×10^{-3} dynes). Ideally, the movable parts should withstand the maximum shock force received under the worst conditions of use in a poorly designed isolator, namely, 2 Newtons (2×10^5 dynes). No transducer design has yet been found which provides requisite sensitivity and resistance to shock forces greater by a factor of 10^8. Supports around the transducer must reduce the shock forces to tolerable levels. To be effective, shock isolators in hearing aids must absorb substantial amounts of energy.[2,3] Small high-compliance vibration isolators used around transducers are not capable of absorbing the energy of a shock even for the small components used in hearing aids and other miniature equipment.

VIBRATION ISOLATION

From the foregoing remarks, it may be evident that vibration isolation in hearing aids is not for the purpose

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† This paper is based on work done for Knowles Electronics, Inc., Franklin Park, Illinois.

1. Most procurements of military equipment require acceptance under specifications of which the following are typical: *MIL-E-4970*: "Environmental Testing, Ground Support, Equipment, General Specification for"; *MIL-E-5272*: "Environmental Testing, Aeronautical and Associated Equipment (General Specifications for)."

2. M. D. Burkhard, *J. Acoust. Soc. Am.* 34, 1981 (1962).

3. H. S. Knowles, *Zeitschrift für Hörgeräte-Akustik* 4, 52 (March, 1965) (In English and German).

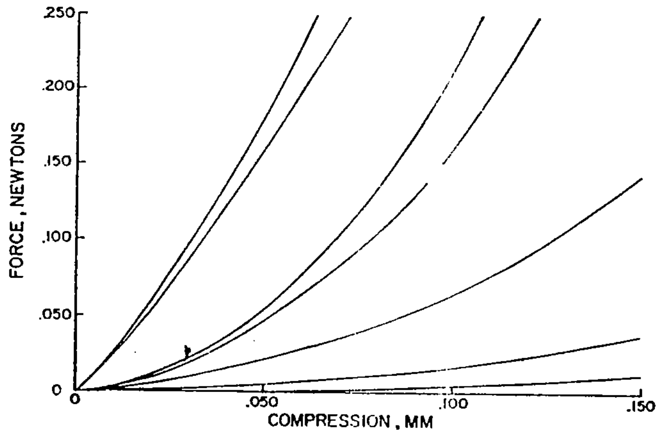


FIG. 2. Typical force-displacement curves for vibration-shock mounts used in hearing aids.

of preventing damage. Vibration isolation is employed in hearing aids to reduce feedback oscillations in small high-gain head-worn aids. (This contributes to their functionally acceptable performance.) Mechanical low-pass filtering is commonly used, one stage being placed between the receiver and the shell and another between the microphone and the shell, as illustrated in Fig. 1. Vibrational coupling between transducers frequently may be further reduced by orienting the microphone and receiver so that their axes of primary translational and rotational motion are perpendicular. Viscoelastic materials commonly used have a compliance to give a vibration resonance with a transducer in the vicinity of 100 to 400 Hz. The resulting support stiffness with a one-gram transducer is 0.4×10^3 to 1.6×10^3 N/m ($.4 \times 10^6$ to 1.6×10^6 dyne/cm). The smallest transducers now available have a mass of 0.7 gm, requiring proportionately smaller stiffness. Typical force *vs* compression curves for transducer mounting elements used for vibration isolation are shown in Fig. 2. Both foam and molded elastomers are represented. All of them have high compliance, i.e., low stiffness, for small amplitude vibrations. A large force due to vibration interaction between receiver and microphone in a hearing aid would be on the order of 12×10^{-3} N (12×10^2 dynes), for which typical isolator compression would be 0.0012 cm. Vibration isolators have dimensions at least an order of magnitude greater because of the limitations imposed by material properties, fabrication difficulties, and to assure linear motion and good high frequency filter properties. These characteristics represent vibration decoupling elements adequate for vibration feedback reduction.

SHOCK ISOLATION

The severity of the shock problem in the field failure of hearing aids, and in particular of their transducers, is known from several considerations:

1. The mean time to failure is much shorter for transducers used in hearing aids than for identical transducers used in other applications. (Identical transducers have been used in manned space flights for both satellite and ground station installations and other applications where reliability is critical.)

2. The types of failures found in most transducers returned for repair can only be reproduced in the laboratory by subjecting new transducers to high shock stress.

3. Controlled laboratory tests on 23 head-worn hearing aids show that very high shock stresses may occur when the aids are dropped on hard surfaces.

4. Field return rates for units evaluated to be more shock resistant, as measured in the laboratory, are significantly lower than for transducers identical in all respects except resistance to shock.

The concept of shock damage and its reduction may be divided into analysis of component susceptibility to damage, a description of the damage producing environment, design and application of shock or damage reducing mechanisms. This is presented qualitatively in Fig. 3, where Curve 1 represents the probability of damage to a transducer as a function of peak acceleration. Above some high value of peak acceleration due to shock, transducer damage and malfunction is certain. Typical transducers are most easily damaged by impacts directed toward the diaphragm, but random drop orientation is assumed. Transducer and other component manufacturers, in general, make the function of Curve 1 as small as practicable, i.e., low probability of failure at high shock levels, without sacrificing sensitivity, distortion and other essential performance. Curve 2 represents the probable distribution of shock peak accelerations that all hearing aids encounter in use during a period of time, e.g., one year. It might be obtained by observing the output of an accelerometer cemented to every hearing aid made. The shape of Curve 2 results from assuming a greater probability of drop from a few inches onto a table top or a few feet onto a carpeted floor during manipulation than from an user's ear onto a ceramic tile bathroom floor. Also implicit in the assumption of Curve 2 is a dependence of peak acceleration on drop height.

The probability of damage is the product of three functions: Curve 1 for the transducer itself; Curve 2 for the population of damaging exposures to the complete aid; and, a function which relates to the reduction of shock

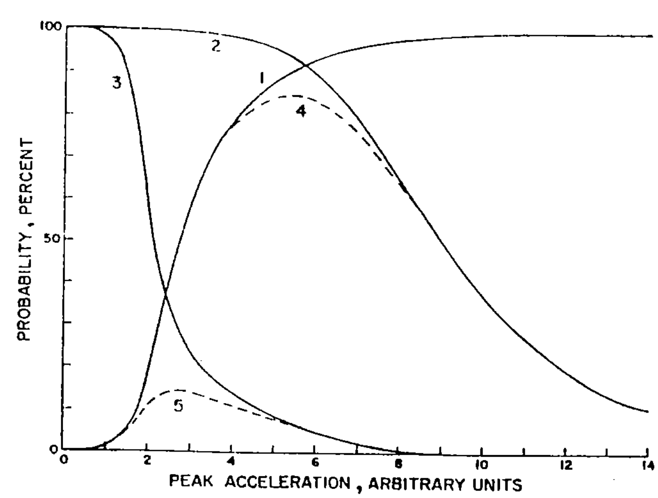


FIG. 3. Qualitative illustration of the reduction in probable transducer failures by use of shock isolation in hearing aids. *Curve 1*: transducer damage function. *Curve 2*: exposure to acceleration peaks. *Curve 3*: exposure modified by a shock-mounting system. *Curve 4*: probable transducer damage function with no shock protection. *Curve 5*: Probable transducer damage function with protection.

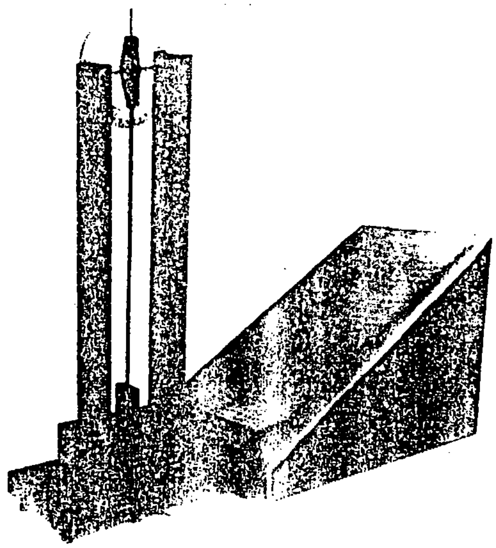


FIG. 4. Pendulum shock machine for the evaluation of hearing aid shock-protection mechanisms.

forces transmitted from the hearing aid shell to the transducer. The latter function depends on the acceleration pulse as well as the physical behavior of the structure surrounding the transducer (refer to Curve 3). Hearing aid designers have demonstrated a range of practices in adjustment of the position and magnitude of Curve 3, as will be evident from the distribution of acceleration peaks shown in Figs. 5 and 6. The probability of a damaged transducer in a hearing aid with little or no shock protection is shown by Curve 4 and the probability of a damaged transducer in a hearing aid possessing adequate shock isolation is shown by Curve 5. These curves are the products of 1 and 2 and 1 and 3, respectively. The curves indicate that the hearing aid with shock protection will have a lower probability of damaged transducers than an unprotected one.

Areas under Curves 4 and 5 represent a fraction of all hearing aid transducers installed in the respective hearing aid model. When they apply to a specified time interval, e.g., one year, they are proportional to the number of units returned due to shock damage from each hearing aid. This gives, therefore, a comparison of failure rates of transducers with and without shock isolation. The areas depicted here show a range of 11:1, typical of the range of transducer return rates among hearing aid models using a given transducer type. It should be noted that the postulated average reduction of amplitude in Fig. 3, as represented by Curves 2 and 3, is only 4:1. Improvement in field performance reliability, and decrease in return rates of hearing aids, can be accomplished by incorporating adequate shock protection designs.

The following paragraphs discuss and present observations from controlled experiments which elaborate and define these concepts for hearing aids.

Shock Resistant Transducers

Miniature microphones, as used in hearing aids, have a history of good resistance to damage from abusive environments. There is ample evidence of this from their wide acceptance for manned space flight operations and in airline communication installations. In one miniature microphone model, means were found to further improve the shock resistance. During a three-month period in 1964 a controlled

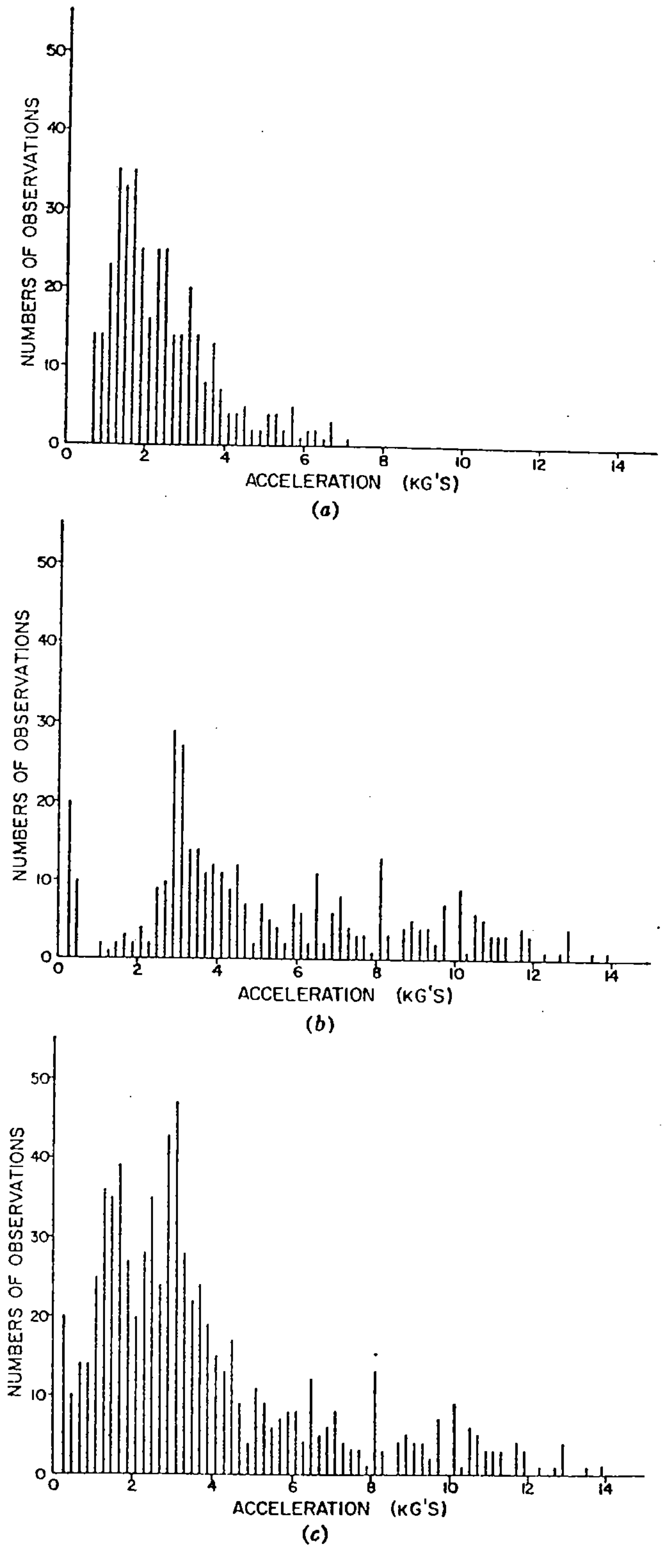


FIG. 5. Peak shock histograms; the data is shown for 30 drops on each hearing aid from 80 cm. *a*. For 13 hearing aids; *b*. For 10 hearing aids; *c*. Composite for 23 hearing aids, 750 observed shocks.

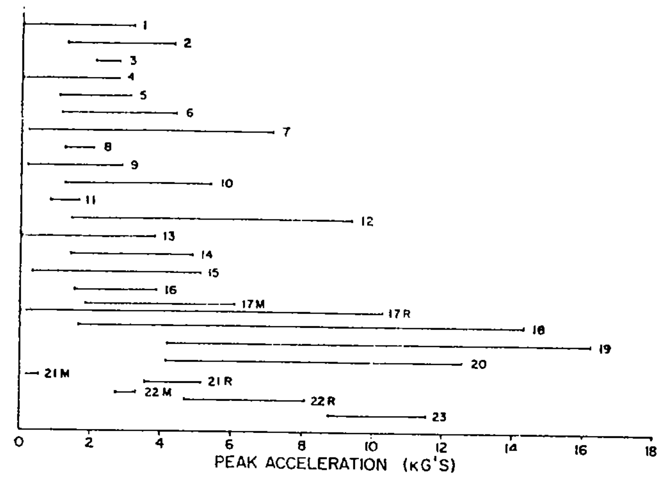


FIG. 6. Peak shock by an individual hearing aid. Each horizontal line includes 99.7 percent of the observations on that hearing aid (30 drops from 80 cm).

field experiment was conducted through Knowles Electronics. Microphones with a modified construction which laboratory experiments indicated to be more shock resistant than standard design were included with shipments of standard products. The mix of 9 standard to one modified was included in shipments of microphones to purchasers of the standard model. Units containing the modified construction had a failure rate of one-half the rate for standard microphones from this control group.

Laboratory evaluation of transducer robustness was done with a modified shock test machine specified for meter movement shock tests.[4] Our machine has a ½ lb. weight to which the transducers being tested are attached. The weight is dropped onto a spring having stiffness to give an 0.8 msec fundamental half sinewave acceleration pulse. A high-frequency acceleration component due to wave motion of the spring during contact of the weight and spring is superimposed on the basic acceleration pulse. As evaluated by this test, a doubled resistance to shock damage in the modified microphone was predicted for a 4000 G peak shock acceleration. High correlation between laboratory and field experience has provided additional evidence that mechanical shock is an important mode of field failure.

Laboratory Simulation of Shock Environment

When a hearing aid is dropped it may impact in any direction, but quantitative data from random drops free of bias are not possible if electrical connection is made from an accelerometer to indicating instruments. The alternative selected for hearing aid shock was to set the hearing aid in motion with a velocity step from a pendulum, shown in Fig. 4. Experiments with miniature transducers have shown that the most damaging shocks occur when the impact direction is perpendicular to the transducer diaphragm. The hearing aid is placed in a high damage-susceptibility orientation on a support platform and the pendulum is raised to the specified drop height and released. This method permits control of the impact direction relative to the transducer and mounting condition being evaluated. The pendulum mass is at least 100 times the typical hearing aid mass. Piezoelectric ceramic accelerometers having identical mass and geometry were substituted for the respective microphone or earphone. Accelerometer output is recorded on a storage oscilloscope or by photography of the acceleration transient on a cathode ray tube display. A net catches the hearing aid after impact.

Twenty-three eye-glass and head-worn hearing aids have been shocked with this apparatus, for evaluation of acceleration force transferred to internal microphones and receivers. The histograms of peak accelerations observed for 80 cm drops are shown in Fig. 5. Figure 5a shows the distribution for the first 13 hearing aids in the series. They were units generally larger in size with larger transducers than the succeeding ten aids included in Fig. 5b. The two histograms are combined in Fig. 5c to show distribution of approximately 750 peak acceleration observations. The range of acceleration by hearing aid is given in Fig. 6, also for a drop of 80 cm. The horizontal lines, one for each hearing aid, give the range of peak accelerations that would include 99.74% of the observations assuming the accelerations to be normally distributed. It is evident that there was considerable range in the effectiveness of shock iso-

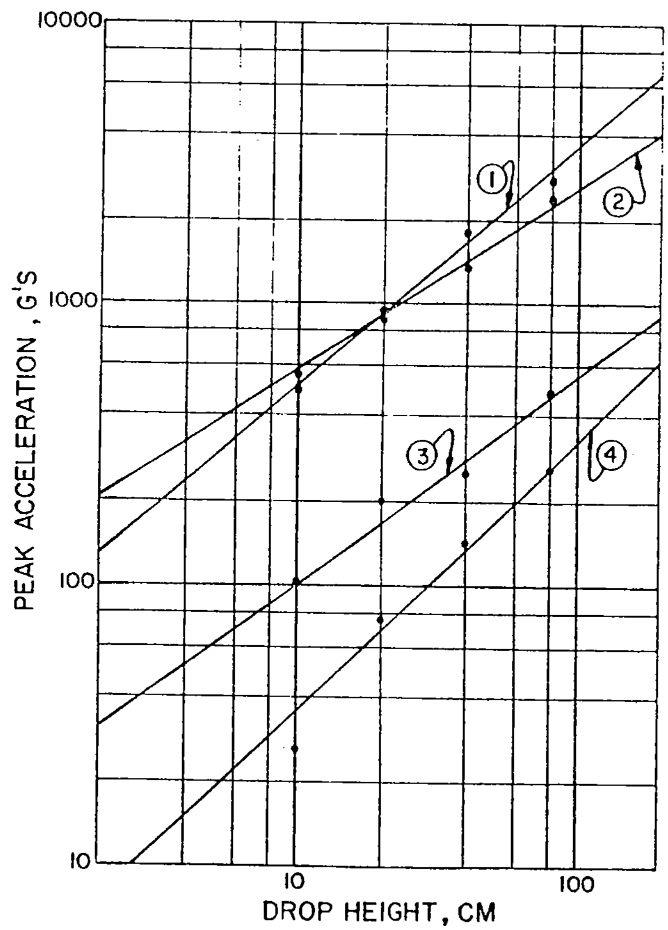


FIG. 7. Variation of peak acceleration with floor covering and drop height. *1.* Bare metal of pendulum; *2.* Asphalt-tile-covered pendulum; *3.* Carpet-covered pendulum; *4.* Carpet-and-pad-covered pendulum.

4. *Stock-testing Mechanism for Electrical Indicating Instruments,* American Standard C39.3-1948.

lation for hearing aid transducers. There appears a tendency for more recently designed hearing aids to allow higher transducer acceleration forces due to shock than older types. However, even in the small types, e.g., hearing aids Nos. 21 and 22, it is possible to maintain relatively lower shock peak accelerations by an adequately designed system. Very little difference is found between microphone and receiver mount shock protection. Hearing aids 17, 21 and 22, where microphone data is M and receiver data is R, are exceptions. In these aids, the mounting methods were dissimilar at the two positions.

The pendulum test for hearing aid shock has also been used to simulate the dropping of a hearing aid on other than hard surfaces. Figure 7 shows the shock observed for simulated drops on an asphalt tile floor, a carpeted floor and a combination carpet and foam pad floor. The test was performed for several drop heights. Substantially lower shocks were observed for the carpeted floor than for the equivalent hard floor and asphalt tile floor, as would be expected from practical experience. To perform these tests, the surface of the pendulum was covered with the respective floor covering materials. Although the pendulum has constant velocity, the rate of change of velocity, i.e., acceleration of the hearing aid, is lower with the resilient coverings on the face of the pendulum, so that relatively lower accelerations due to the impact are produced in the transducers. This suggests that one means of reducing the impact damage to hearing aid transducers and components would be to provide more compliance and energy storage mechanism in the shell and around the hearing aid. Apparently, dropping typical hearing aids on the carpet, even from a height of two meters, would not result in damage to most transducers, whereas dropping aids on a hard surface, simulated by the metal and metal backed asphalt tile, from one meter could be expected to yield a high incidence of damaged transducers.

The rate of change of peak acceleration with drop height is controlled by the force *vs* compression characteristics of the isolation system. As was evident in Fig. 2, typical transducer mounting provides a stiffness which increases with compression and/or applied force. Optimum use of any isolation system requires maximization of the area under a force *vs* compression curve for the isolation. The variables to be adjusted are the initial stiffness or rate of compression of the isolator, the total compression necessary to absorb the energy and the maximum force or acceleration to be allowed on the transducer. In general a low acceleration can be achieved if a large space is allowed for compression of the isolator; but, if space is limited, a larger force must be allowed.

DESIGN OF EFFECTIVE SHOCK PROTECTION

The following sections give a discussion of component isolation from the packaging point of view.[5] It is included here because it illustrates approaches to design of shock protective elements in hearing aids and other miniature equipment. Fortunately, the analysis is a little pessimistic relative to actual data on hearing aid shock, probably because it neglects shell compliance. The analysis does indicate relationships and approximate magnitudes applicable to shock protection design for transducers in hearing aids. Much more detail will be found in published literature on the subject of shock protection.[6]

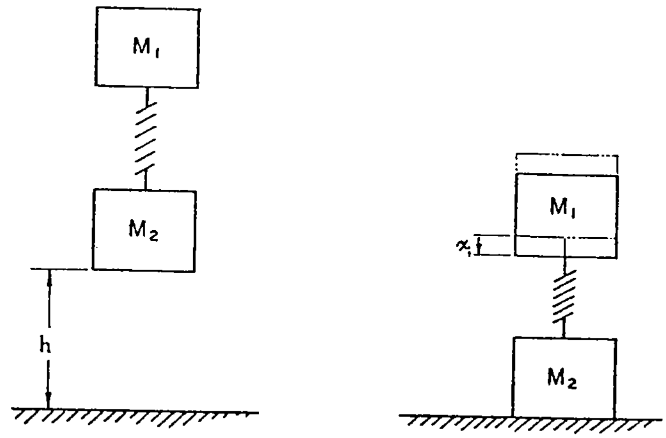


Fig. 8. Mass spring system of component isolation.

Theory

To describe the acceleration and motion of a component in a dropped hearing aid, Newton's second law of motion is applied. Damping in the isolator will be neglected. In Fig. 8, the component to be protected has mass m_1 and is attached to its surroundings by a spring having stiffness k_1. The surroundings have mass m_2 and may have additional stiffness which will be lumped with k_1 to simplify this analysis.[7] Also assume that $m_2 >> m_1$. The forces on mass m_1 will be gravitational, $m_1 g$, and the spring reaction P. If we consider only the motion of m_1 relative to m_2, the acceleration of m_1 is x''_1 so that the equation of motion is

$$m_1 x''_1 = m_1 g - P. \tag{1}$$

When the mass is falling from height, h, there is no reaction from the spring and $P = 0$. Then

$$x''_1 = g \tag{2}$$

is the equation of motion from the time the object starts to fall until instant of impact, neglecting air resistance. Integration of Eq. (2) gives the velocity, x'_1, and position, x_1 as a function of time, t, and h, during fall, which for m_1 initially at rest is

$$x'_1 = gt. \tag{3}$$

$$x_1 = \tfrac{1}{2} g t^2 - h. \tag{4}$$

Equation (1) may now be integrated and its constants of integration evaluated. The first integration gives the energy equation:

5. Raymond D. Mindlin, "Dynamics of Package Cushioning," *The Bell System Technical Journal* 24, 353-461 (1945).

6. Cyril M. Harris and Charles E. Crede, *Shock and Vibration Handbook* (McGraw-Hill Book Company, New York, 1961), in particular Chap. 31, Vol. 2.

7. This stiffness can be very important, however, to shock acceleration reduction when it enters as an elastic element between the hearing aid shell and impact surface. Examples are a flexible shell and the impact of the hearing aid on a carpet.

$$\tfrac{1}{2}m_1x_1'^2+\int_0^{x_1}Pdx=m_1g(h+x_1) \qquad (5)$$

where $\tfrac{1}{2}m_1x_1'^2$ = instantaneous kinetic energy of m_1, $m_1g(h+x_1)$ = instantaneous potential energy of m_1 at its initial height $h+x_1$. Pdx = energy stored in the spring at any instant.

Usually $h >> x_1$ so that Eq. (5) may be written

$$\tfrac{1}{2}m_1x_1'^2+\int_0^{x_1}Pdx=m_1gh. \qquad (6)$$

With equal approximation, Eq. (1) may be rewritten,

$$m_1x''_1+P=0. \qquad (7)$$

The important parameters in the shock protection problem are the maximum deflection, d_m, and the peak acceleration, G_m, of the component to be protected ($G_m = |x''/g|max$). If further the analysis is limited to the time when the supports are under compression, i.e., $P > 0$, and $x_1 > 0$, then from Eq. (6) it is seen that x_1 is a maximum when $x'_1 = 0$, hence

$$\int_0^{d_m}Pdx=m_1gh \qquad (8)$$

expressing the energy absorbed in the spring to be equal to the initial potential energy of the isolated component for optimum use of the elastic isolator.

$$G_m=P_m/m_1g. \qquad (9)$$

P_m is the maximum force value of P.

$$P_m=P(d_m). \qquad (10)$$

The general procedure for design of an isolator, then, is to calculate d_m by Eq. (8), P_m by Eq. (10) and G_m by Eq. (9). It is necessary to specify the dependence of force, P, on the compression x_1 of the spring. A familiar type support is a linear spring defined by force proportional to displacement,

$$P=kx. \qquad (11)$$

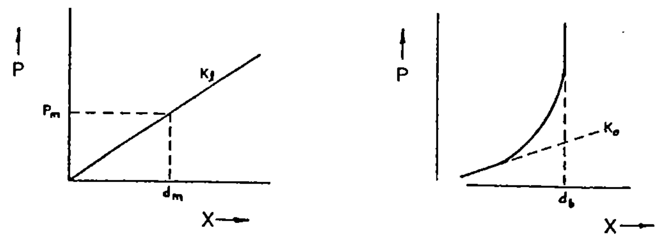
A nonlinear spring is more common in small component isolation in which the stiffness increases with displacement, as in Fig. 2. This may be in the form of a rather abrupt change of "bottoming" when a component on a linear spring comes against a mechanical stop. Usually it is a gradual transition from low stiffness to high stiffness as a viscoelastic material compressing under load. These are depicted as linear elasticity and tangent elasticity in Fig. 9.

Linear Elasticity

Consider first the case of linear elasticity, i.e., force P proportional to displacement x. For maximum displacement d_m, maximum force is specified.

$$P_m=k_ld_m. \qquad (12)$$

This is shown schematically in Fig. 9(a). Substitution of



(a) (b)

FIG. 9. Elasticity diagrams. *a*. Linear elasticity. *b*. Tangent elasticity.

Eq. (12) into Eq. (8) and integration gives, on rearrangement of terms,

$$k_l=2m_1gh/d_m^2. \qquad (13)$$

This is the spring constant which gives zero velocity for deflection d_m and limits the force to P_m. The resulting peak acceleration is

$$G_m=2h/d_m. \qquad (14)$$

A spring constant greater than k_l will produce accelerations greater than G_m.

A spring constant less than k_l, with specified d_m, also produces peak accelerations greater than G_m due to abrupt bottoming of the component against mechanical stops.

Tangent Elasticity

Tangent elasticity describes a mounting system in which the elastic constant increases with compression of mounting material. It is one of the ways for describing an infinite change of elasticity in a finite compression range. It might, for example, describe many types of rubber and compressed foam elements which are mounted directly between the transducer and the surrounding structure. This function also describes an isolation that might be made to have both vibration isolation and shock protection capabilities. The mathematical expression for tangent elasticity is

$$P=(2k_od_b/\pi)\tan(\pi x/2d_b). \qquad (15)$$

The force displacement relationship is indicated schematically in Fig. 9(b). Maximum excursion allowed for compression of the mounting material is d_b. There will be a range of approximately linear elasticity for small compressions having a slope of k_o. As before, the force function given by Eq. (14) is substituted in Eq. (8) and the integration performed,

$$\tan^2(\pi d_m/2d_b)=-1+\exp(\pi^2mgh/2k_od_b^2). \qquad (16)$$

Substitution of this result in Eq. (15) gives the maximum force for this type of elasticity,

$$P_m=(2k_od_b/\pi)[-1+\exp(\pi^2mgh/2k_od_b^2)]^{\frac{1}{2}}. \qquad (17)$$

We next ask what is the smallest value of G_m which is available in the range d_b of motion and for a given drop height? We have latitude to adjust the initial spring constant k_o. The adjustment is done by maximizing. Substitution of Eq. (17) into Eq. (9) gives the peak acceleration. Differentiating with respect to the initial stiffness k_o and setting equal to 0, we obtain the initial spring stiffness.

$$k_o=3.1\,mgh/d_b^2. \qquad (18)$$

For this value of k_o optimum peak acceleration is

$$G_m = 3.9h/d_b. \tag{19}$$

It is apparent that the acceleration experienced by an object mounted in a tangent elastic structure which is highly nonlinear becomes, under the best circumstances, almost double that obtained with a material which operates strictly as a linear spring throughout the range $d_b = d_m$. Maximum or limiting deflections for linear and tangent elasticity can be stated in terms of the small amplitude resonant frequency, since $k/m = \omega^2 = (2\pi f)^2$. Eqs. (13) and (18) become,

for tangent elasticity:

$$d_b = (\sqrt{3.1mgh})/2\pi f, \text{ and}$$

for linear elasticity:

$$d_m = (\sqrt{2mgh})/2\pi f.$$

Shock Isolator Example

It is instructive to examine the results predicted by Eqs. (13), (14), (18) and (19) for several conditions within the range of constants common in hearing aids. These are shown in Table 1 where linear and tangent elasticity are compared for the same maximum deflection, *1* and *2*; the same peak acceleration, *3* and *4*; and the same small amplitude resonance frequency and stiffness, *5* and *6*. Note that a linear spring for the same deflection limit has about one-half the peak acceleration of a tangent spring. Space in excess of 1 mm is desirable to achieve acceptable reduction of shock accelerations, when these are the only mechanisms employed. Since these are optimized conditions for a 100 cm drop they will not be optimized for either smaller or larger drop heights. A different set of optimum parameters is obtained for a 200 cm drop. Peak accelerations for higher drops using the isolator parameters in Table I will produce significantly larger shock peak accelerations. At lower drop heights, the peak accelerations will be less but not as low as if the isolation were optimized to that height. If an isolator is designed and determined to provide adequate protection to a transducer at a given height, it will provide protection for smaller heights.

TABLE I. Examples of optimum linear and tangent elasticity shock protection. Drop height is 100 cm. For lines *1* and *2*, a total space of 1 mm is assumed for a material which has a maximum deflection at 60% compression.

Elasticity	d_b or d_m, mm	G_m	K/m, sec^{-2}	f_o, Hz
1. Tangent	0.6	6500	8×10^7	1.4×10^3
2. Linear	0.6	3300	5.5×10^7	1.2×10^3
3. Tangent	1.9	2000	8×10^6	450
4. Linear	1.0	2000	2×10^7	710
5. Tangent	5.95	656	8.7×10^5	150
6. Linear	4.79	418	8.7×10^5	150

The numbers predicted are startling, but no more than the peak accelerations observed in hearing aids. Although small compliant elements are well adapted to vibration isolation of the transducers in a hearing aid, they are markedly ineffective as shock isolators. There must be means for absorbing large amounts of energy if shock protection is to be effective. When peak shocks are less than this theory predicts, it may be attributed to several factors: First, many viscoelastic materials exhibit much higher stiffness under high force transients than can be observed by quasi-static methods, and therefore absorb more energy than predicted. Second, most of the materials dissipate some of the shock energy as heat. Although the amount is usually small, it may be measurable. Third, most hearing aid shells flex under the impacts of shock and thereby absorb appreciable energy. This latter method of reduction of shock forces on internal components, particularly transducers, can be used much more effectively in small hearing aids than evidenced to date. Fourth, some isolator elements provide a crushing type of nonlinear elasticity not covered by either of those treated here. Premiums placed on space for cosmetic reasons seems to have dictated against these more recently.

CONCLUSION

Well designed hearing aid transducers will withstand shock peak accelerations in the range of 1000 to 2000 G. With forces of higher shocks, the moving parts that are sensitive to sound pressures are distorted beyond their limits of usefulness. For reliable field performance means must be included to absorb shock energy before it reaches the transducer. This requires space and careful analysis and design of not only the individual isolation elements but also the interaction with the rest of the structure including shell, amplifier, battery, etc. Two types of elasticity for application to the transducer isolation in hearing aids have been considered because they are easier to install in the small size demanded by hearing aid users. Other approaches are of course possible.

THE AUTHOR

Mahlon D. Burkhard was born in Seward, Nebraska in 1923. He received an A.B. in physics and mathematics from Nebraska Wesleyan University in 1946, an M.S. in physics from Pennsylvania State University in 1950, and has done additional graduate work in physics and mathematics.

In 1950, he joined the National Bureau of Standards, Acoustics Section. Moving to Chicago in 1957, he became supervisor of the Acoustic Design Section and later supervisor of the Acoustics Section of Armour Research Foundation. Since 1960, Mr. Burkhard has been with Industrial Research Products, Elk Grove Village, as Manager of Acoustical Research.

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