EFFECTS OF HIGH INTENSITY ACOUSTIC FIELDS ON CRYSTAL VIBRATION PICKUPS

by

Wilson Bradley, Jr.

ENDEVCO CORPORATION

I — Introduction

Considerable vibration in flight or static testing is induced by acoustical energy. It has been shown in numerous cases that random acoustical energy at 120 db can induce vibrations of the order of 50 g or higher in structural members and it can be assumed that higher acoustic levels may increase the vibration levels although not necessarily linearly. It has also been shown that crystal accelerometers have, at the most, outputs of only a few g at 150 db noise levels so that if the usual high level vibrations are to be measured in acoustic fields, good signal to noise ratios and accuracies are obtainable. There are other cases, however, where low level vibrations on structures that are not subject to acoustic excitation must be made in high level acoustic fields. In these cases considerable attention must be paid to the acoustic response of the measuring system to assure good signal to noise levels and accuracies. It is the purpose of this paper to discuss the important parameters to consider with experimental results plotted to confirm theoretical predictions.

II — Fundamental Considerations:

A. Internal Transducer Design

First consider the internal construction of a crystal transducer which can be represented as shown in Figure 1 regardless of actual configuration where:

P_e = Acoustic Pressure in db outside transducer case.
P_i = Acoustic Pressure in db inside transducer case.
A_e = Effective external area over which acoustic pressure is effective.
A_i = Effective internal area over which acoustic pressure is effective.
F_n = Resultant force experienced by crystal sensing element due to acoustic pressures.
E_n = Voltage out of transducer due to F_n.
M = Effective mass of inertial mass acting on crystal sensing element.
K_x = Combined spring constant of crystals (and other structures) that support the inertial mass.
K_e = Combined spring constant of outer case of transducer.
K_i = Spring constant coupling crystal or inertial mass to outer case.
F_a = Resultant force experienced by crystal sensing element due to acceleration.
E_a = Voltage out of transducer due to F_a.
f = frequency in cps.
f_n = resonant frequency in cps of transducer.
a = acceleration in g's.

NOTE: For the sake of this analysis the system is considered to be single degree of freedom in the z axis with x or y components of the above parameters contributing only secondarily. The actual test data was taken with omni-directional acoustic pressures and the response of the system tends to bear out the assumption of single degree of freedom.

The nature of the crystal sensing element is that it produces a charge $\left[\text{and hence a voltage } (E) = \dfrac{\text{charge } (q)}{\text{capacity } (C)}\right]$ directly proportional to applied force along the z axis.

Therefore:

$$(1) \quad E_a \cong F_a = Ma \times \frac{1}{1 - \left(\frac{f}{f_n}\right)^2}$$

The last term here represents the frequency amplification factor introduced by a crystal transducer as a result of the fact that it is virtually undamped. For all accurate vibration measurements $\frac{f}{f_n}$ is kept below 0.2, resulting in errors at the highest frequency of only a few per cent. If this procedure eliminating frequency dependence is followed, then:

$$(2) \quad E_a \cong F_a = Ma$$

and for any given acceleration value (a)

$$(3) \quad E_a/a \cong F_a/a = M$$

The voltage due to acoustic pressure is:

$$(4) \quad E_n \cong F_n = P_i \times A_i \times \frac{1}{1 - \left(\frac{f}{f_n}\right)^2}$$

and for any given acoustical pressure P_i

$$(5) \quad E_n/P_i \cong F_n/P_i = A_i \times \frac{1}{1 - \left(\frac{f}{f_n}\right)^2}$$

The frequency dependent term cannot be automatically dropped here since acoustic energy frequencies are usually broad band to at least 10,000 cps requiring f_n to be 50,000 cps or above before the term becomes negligible.

A Technical Paper prepared for the 26th Shock and Vibration Symposium

The primary parameter of interest in any measurement is the best possible signal to noise ratio which in this case is:

$$(6)\quad \frac{E_a/a}{E_n/P_i} \cong \frac{M}{A_i \times \dfrac{1}{1-\left(\dfrac{f}{f_n}\right)^2}}$$

for any given acceleration level and internal acoustical pressure level.

The first thing to note is that the signal to noise ratio will be affected adversely as f approaches f_n. Figure 2 shows this effect graphically for acoustic pressures containing frequencies up to 10,000 cps. It indicates that if acoustic pressures up to 10,000 cps are present, transducers with greater than 30,000 cps natural frequencies are required to achieve optimum signal to noise ratios. For lower natural frequency pickups low pass filtering can be employed, but at considerable loss in accuracy due to the introduction of other sources of error, particularly when measuring complex vibration or shock phenomena.
The second thing to note is that signal to noise ratios are optimum for higher inertial mass and small cross sectional areas. The ratio of K_x to M is defined however by the first consideration that:

$$(7)\quad f_n = \frac{1}{2\pi}\sqrt{\frac{K_x}{M}} \geqslant 30{,}000$$

so that as M is raised K_c must be raised also to maintain a high natural frequency. This suggests that using the crystal sensing element in compression will give optimum results because for a given natural frequency a higher mass loading can be employed than for crystals in bending or shear modes.

B. Effects of Transducer Case Design

The above discussion relates to response of *internal* mechanism to acoustic pressure P_i. The design of the transducer case will determine the ratio of internal pressure to external pressure, (P_i/P_e). Consider the various design approaches as shown in Figure 3.
The ratio of internal force or pressure P_i experienced by the crystal element to the external pressure P_e is determined by how much of the pressure is absorbed by the case and how much absorbed by the crystal element. For each design the approximate ratios are as follows:

Compression:

$$(8)\quad \frac{P_i}{P_e} = \frac{K_x}{K_x + K_c}$$

Since the crystal constant K_x is usually equal or greater than K_c.

$$(9)\quad \frac{P_i}{P_e} \geqslant 1/2$$

Isolated Compression:

$$(10)\quad \frac{P_i}{P_e} = \frac{K_s}{K_s + K_c}$$

where K_s is the combination spring constant of K_i and K_x and equals: $\dfrac{K_i \cdot K_x}{K_x + K_i}$

By appropriate design:

$$(11)\quad K_i \text{ can } \leqslant \frac{1}{20} K_x \text{ so } K_s \geqslant \frac{1}{21} K_x$$

then if K_x and K_c are approximately equal

$$(12)\quad \frac{P_i}{P_e} \approx \frac{1}{20}$$

with the resultant advantage over the non-isolated compression design, that for a given external acoustic pressure, the internal pressures will be only about one-tenth as large, resulting in lower noise levels.

Isolated Bending:

For this design Equation (10) holds true also except that K_s is now frequency dependent being equal to $\dfrac{K_i \cdot K_x}{K_x + K_i}$ where K_i is the effective spring constant of the internal gaseous medium measured as varying between 0 at very low frequency pressures to approximately 1/20 K_x at frequencies over a few hundred cycles. So it can be said that:

$$(13)\quad \frac{P_i}{P_e} \approx \frac{1}{20} \text{ for most acoustical frequencies.}$$

III — Test Results

All test data shown here* were taken in an Altec-Lansing noise chamber as shown in Figure 4. Equipment included a magnetic tape source of wide band (150 cps to 9600 cps) noise, single frequency oscillators, power amplifiers, filters and an Altec-Lansing microphone calibrated by the National Bureau of Standards as a standard measure of acoustic noise. Specimens were suspended on their cables in the noise chamber. A low frequency mount is mandatory; otherwise the transducer will experience vibration, induced by acoustic pressures, and it will be impossible to determine the effects of the acoustic environment alone. Confirming tests showed that an aluminum plate one inch thick experiences as much as 100 g's of vibration at 140 db, random sound pressure. Checks were made proving the omnidirectional characteristics of the noise. Over 50 vibration accelerometers were tested including experimental units. All duplicate models showed less than 5% scatter so the following results can be considered as typical.

*Other tests were made in plane wave chambers with data of the same magnitude resulting but that was not predictable or repeatable. Frequency errors in the chamber were proven and other non-uniform information was attributed to the effect of standing waves so that this method of testing was abandoned.

Narrow Band Test:

All accelerometers with natural frequencies of 35 KC or higher were checked for noise output at discrete single octave bands up to 10 KC at a constant db level of 150 db. Results are shown in Figure 5 plotted at the mid-frequency point of each band. These results confirm that:

a) The noise output from compression and isolated compression designs is essentially independent of frequency.

b) The noise output from isolated-bending designs is frequency dependent as predicted.

Random Frequency Test:

All accelerometers with natural frequencies of 35 KC or higher were checked for their acoustic response to random-white noise at levels up to 156 db. The noise was of equal amplitude from 150 cps to 9600 cps chosen to simulate as closely as possible the broad band frequencies that have been measured in the near field of a rocket engine. The results of all commercial production models are as shown in Figure 6* and in general show that all designs have output directly proportional to pressure so that for every 20 db increase in noise level the signal output increases by a factor of 10.

Other experimental units confirmed that acoustic response can be significantly reduced by designing units specifically for the environment, in accordance with previous design considerations.

*Isolated compression design plotted is Endevco Model 2213.

External Acoustic Attenuation Tests:

The best production units (isolated-compression) were further tested using external attenuation techniques such as:

a) Unit with rubber cap over the body.

b) Unit with Styrafoam over the body.

c) Unit with sound absorbent material sprayed over the case.

In general only insignificant reductions in acoustic responses were achieved with broad band noise. This was as predicted since surface dimensions would have to be large compared to sound wave lengths before significant attenuation could be achieved.

Summary:

All crystal transducers tested showed noise outputs due to acoustical pressures to be very low in relation to vibration levels usually encountered under these environments. If, however, low level vibrations are to be measured in high acoustic noise fields the following points should be observed:

1) Transducer resonant frequencies should be at least three times the highest *acoustical* energy present. See Figure 2.

2) Transducer inertial mass (and therefore probably total mass) should be as high as practical for device under test with transducer cross-sectional areas kept to a minimum. See Equation 6.

3) When necessary, noise levels can be further reduced, not by external acoustic attenuation but but by specialized design techniques.

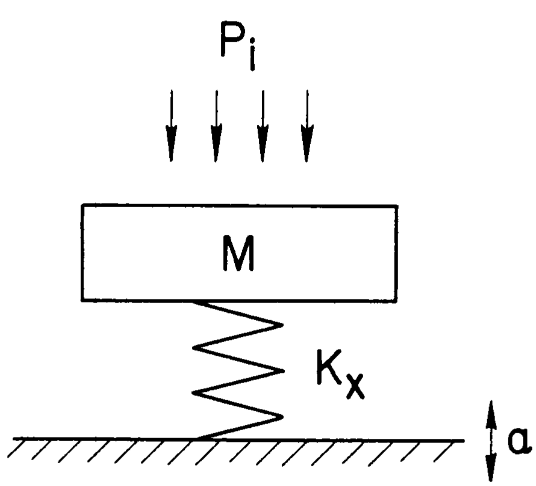

Figure 1

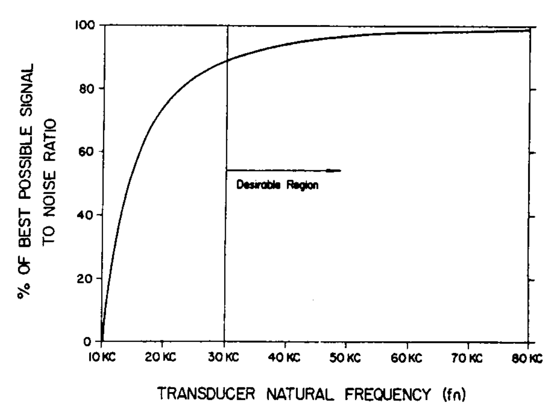

Figure 2

REV. 2/6/59

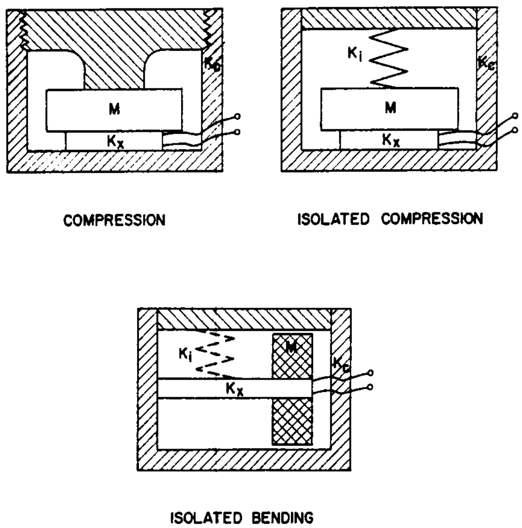

Figure 3

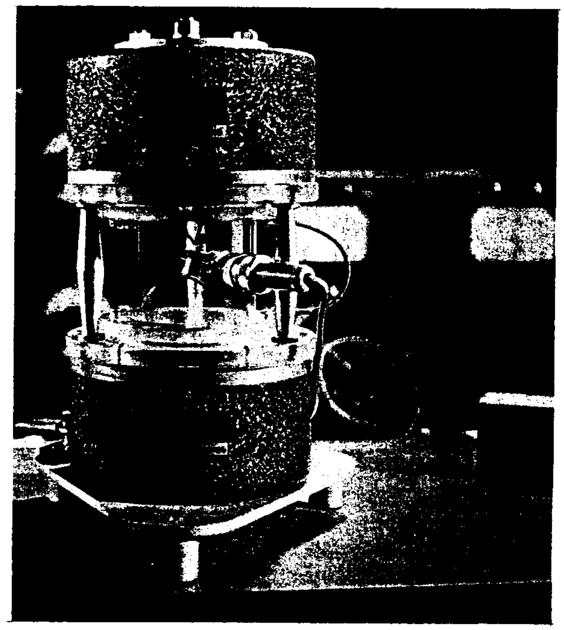

Figure 4

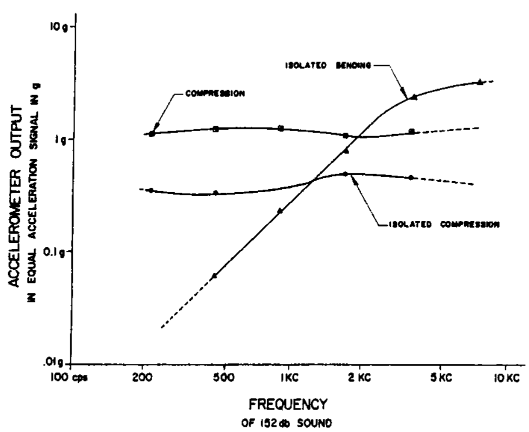

Figure 5

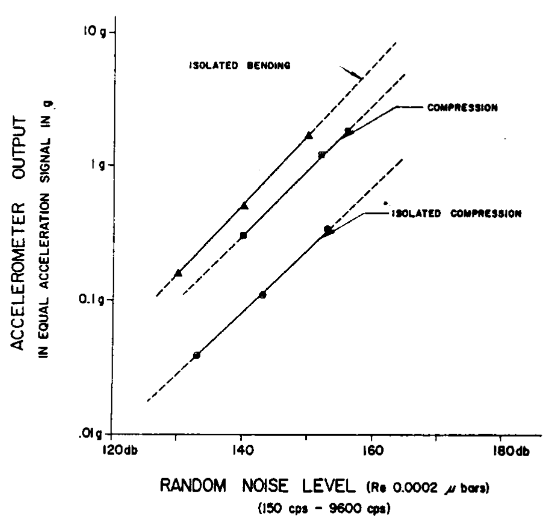

Figure 6