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MATERIALS AND TECHNIQUES FOR DAMPING VIBRATING PANELS

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The technology of vibration damping has to do with methods for the conversion of the energy of mechanical vibration in solids into the harmless form of heat. The subject is too broad to allow of treating all of its aspects here, so attention is directed to the specific problem of damping materials applied to the surfaces of vibrating panels, thereby neglecting such subjects as the damping inherent to structural materials and inherent to the technique of fabrication, and also neglecting damping treatments for structural shapes and piping, etc., which are of a different character from extended panels. This specialization of topic is doubly justified in a noise symposium: first, because vibrating panel surfaces constitute the most efficient radiators of noise and are, therefore, most in need of treatment; and second, because other techniques of damping structures are generally less susceptible to quantitative treatment, and the other sources of damping are generally orders of magnitude too small to accomplish respectable noise reductions that are still consistent with requirements for structural integrity. It invariably turns out, whenever you are designing a system of machinery or noise-reduction enclosures, that you get the most noise-reduction per dollar by applying the damping treatment directly to those panels that are responsible for radiating the offending noise, and that at this point in the development of damping technology, it is still the best engineering decision to select your structural materials from considerations of strength, weight, cost, etc., and then derive the necessary damping separately by application of surface treatments that have been devised specifically for the sole purpose of high damping capacity at low cost.

The need for good panel-damping treatments in the whole noise-reduction picture is quite obvious. The techniques of vibration isolation are not perfect, so that vibratory excitation from noise sources is bound to be communicated to some extent to panel surfaces which are capable of radiating broadband noise. Furthermore, the full transmission loss of partitions for isolating airborne sound can never be realized without adequate panel damping. Hence, the need for damping must be recognized by the machine designer, who couples tanks and ducting, etc. into his machinery layouts; by the architect, who erects partitions to ensure some acoustical privacy; and by the noise-reduction expert, who designs machinery enclosures to isolate airborne and structureborne sound at its source.

But, it is equally important to recognize the limitations of damping in the noise-reduction picture. Brute-force damping is not a cure-all, both by reason of technical limitation and uncompromising economics. There are certain situations where damping is of enormous and indispensable benefit toward noise reduction, and others where it is a pitiful waste of money and effort. This is what motivates us to the discussion of the <u>design</u> of vibration-damping treatments.

Steady-state resonance and free vibration after impact are the key situations where the use of damping is likely to be of great benefit. At resonance, i.e. whenever the frequency of the exciting force happens to coincide with one of the natural frequencies of a panel, the effect of increased damping is shown in Figure 1. Here, the noise radiated by a panel has been plotted against the frequency of its exciting vibration. A light damping treatment is seen to reduce the radiated noise level by more than 20 db near one of the natural frequencies of the panel at 138 cps, but the reductions due to damping are not worth the effort at frequencies far removed from resonance. Hence, the noise-reduction applicability of damping depends upon how difficult it might be to avoid this situation of steady-state resonance in practical machinery and panel designs. But the next figure will give some insight into why it is virtually impossible to design resonance out of the noise-reduction problem.

Figure 2 shows the frequency and relative responsiveness of some of the resonances excited in an ordinary automobile door panel under constant excitation. Suppose you were assigned the task of redesigning this panel, without resorting to damping, so that none of its prominent natural frequencies would go into resonance with the engine-noise vibrations. If you succeeded in the impossible task of avoiding resonance with the harmonics of the firing frequency at any one speed, what about a change in engine speed? What about resonant amplifications of the bands of white noise that fall near natural frequencies? What about the afterring due to excitation of all natural frequencies by the impact of a competing salesman's knuckle test and the shock of door slamming and road bumps. What about the increased average level of noise due to excitation of all natural frequencies by the recurrent impulses of rain and pebbles and the recurrent shocks of road noise? And, worse yet, what about the awful coincidence of "pseudo-resonance" when the rate of recurrence of the impulse coincides with a submultiple of some natural frequency so as to re-excite the afterring each time in phase to build up out of all proportion with the magnitude of the exciting force. Vibration-damping treatment is an answer to each of these questions, whereas any amount of redesign of the mass-stiffness relationships will invariably only retune the noise problem unless the machine and panel are so trivially simple as to be academic.

This kind of reasoning poses the damping problem as a very common one, with the effects of resonance contributing to noise production as the rule rather than the exception, but nothing has been said yet about the relative importance of resonance in establishing the total noise level in any particular machinery noise application. These latter considerations are the ones that motivate damping-treatment design through the establishment of the point of diminishing returns in the noise reduction that can be accomplished by damping in specific applications. For example, there would be little advantage to damping the resonant noise contribution of the vibrating panels beyond the limit set by some other non-resonant noise source. More damping on a door panel is hardly likely to reduce windage noise in an automobile, so that if windage remains there is a practical upper limit on the damping capacity needed for the door treatment. Design considerations of this type arise in most noise-reduction problems: the question of how much damping is needed on a panel is not much different than how much absorption is needed in a room. A certain amount is good, but too much is a waste of money.

Treatment design of any kind requires a quantitative test whereby the relative effectiveness of different materials can be ranked on a practical scale. The question of which material and how much material are answered by deduction from arbitrary test results; e.g., which ceiling treatment and how much ceiling treatment are required for noise reduction in a specific room are deduced from absorption coefficients determined in a reverberation room which itself may not look even remotely like the room that will ultimately be treated. Similarly panel-damping treatments are selected on the basis of test results established with an arbitrary panel that may not look even remotely like the panel to be treated. Indeed, automotive sheet-metal treatments are selected in accordance with tests performed with 1/4"-thick steel plating. The reason for this arises from an analysis of the laboratory requirements for a valid test of damping capacity over a practical application range, and is no more dictated by the average metal-thickness encountered in practice than is the size of a laboratory reverberation room influenced by the average length of the school corridors that might have to be treated with ceiling tile. Much confusion can be avoided by clearly recognizing this point of testing philosophy from the outset.

A schematic diagram of the test apparatus devised by the late Paul H. Geiger for ranking automotive deadeners is shown in Figure 3. For our purposes here, we need not dwell on the details of the experimental procedure nor on the detailed results of the numerous tests that have been performed on a wide variety of materials. In these respects, reference is made to Chapter IV of Dr. Geiger's Noise-Reduction Manual. We need only concern ourselves at this point with the fact that this instrumentation provides us with a number, the vibration decay rate in decibels per second (db/sec), which indicates the damping capacity of a treatment by virtue of its effect on an arbitrarily selected free vibrating system; viz, a 1/4" x 20" x 20" cold-rolled steel panel, weighing about 30 pounds, freely suspended at nodal points to resonate at 160 cycles per second. The analytical validation of the test procedure culminates in the idealized relationship shown in the corner of the figure, where the vibration decay rate as measured, D, is stated to be directly proportional to a damping constant associated with the treatment, c, whenever the natural frequency, w_o, and the critical damping constant, c_c, are standardized by fixing on a specific panel weight and shape, as well as a standardized suspension. The higher the decay rate, the better the damping. The practical validation of this type of test procedure arises from the extensive correlations that have been established between the test decay rates of treatments and their noise-reduction effectiveness by several different criteria, such as db-reduction at steady-state resonance, tinniness judgements by juries of observers, and vehicular road-noise measurements.

At this point, then, we are in a position to classify different vibration-damping treatments by their test decay rates, and thereby deduce some very approximate, but very useful, selection rules for the design of specific treatment applications. But, since the effectiveness of most damping materials depends markedly on their method and weight of application, and since some are notoriously temperature sensitive, the general classification that is about to appear must itself be very approximate in order to have enough scope to be instructive. Hence, Figure 4 ranks various vibration-damping treatments in accordance with the following remarkably qualified parameter: the vibration decay rate in db/sec at 160 cps at 70 degrees Fahrenheit for overall application by "standard" techniques to one side of the standard 1/4-inch panel. Just blink at the arbitrariness of the criterion for the moment, if you can, and notice particularly the wide spread in results. Several orders of magnitude of effectiveness are indicated along the logarithmic ranking scale. Each horizontal dash represents the range of effectiveness of some class of material for the range of application weight indicated in pounds per square foot. All but the top dash apply to continuous overall application of the material. The so-called frequency-selective deadener represents a special case to be mentioned later.

From this tabulation we see certain trends in the data. The effectiveness of mastic deadeners is quite variable as to product, and a strong function of application weight. The effectiveness of impregnated felts depends profoundly on the number of ply, its indentations, and its surface loading. The effectiveness of fibrous blankets depends on method of attachment and surface loading, light blankets correlating with treatment weight and heavy blankets with treatment thickness. These are interesting observations, but how are they to be interpreted in the light of a specific application problem? To answer this we must appeal to a little low-brow deduction from the equation shown in Figure 3. Recall that there the decay rate of an idealized panel in free vibration was shown to depend both on its natural frequency of vibration and its percentage of critical damping. We can show also that the dependence on critical damping constant can be taken into full account at a given frequency by regarding the decay rate to be inversely proportional to the panel mass in vibration. Hence, if we loosely associate the decay rate in db/sec with noise-reduction effectiveness (a perfectly legitimate association in the spirit of our present thinking, because the db/sec decay rate of a free vibration is directly proportional to the db-suppression of steady state resonance at the natural frequency), we have deduced two operational rules for the use of damping treatments. First, for panels of equal mass, more damping will be required for given suppression of lower-frequency resonances; i.e., treatments of higher test decay rate are required at lower frequencies. Second, for resonances at the same frequency, more damping will be required for given suppression of resonances in panels of greater mass; i.e., treatments of higher test decay rate are required for heavier panels. Now we have design criteria for the use of our data, admittedly deduced and stated here as rough-and-ready rules, but only for lack of space as more extended analysis and experimentation would show in the practical range of noise-reduction parameters.

Although the amount of damping required in any specific panel application can only be determined with complete assurance of best economy by an experimental establishment of a point of diminishing returns in noise reduction, a rough-and-ready figure for "adequate" panel damping in most applications has been found to be about two per cent of critical damping, which corresponds to a test decay rate of 150 db/sec at 160 cps. Referring back now to our tabulation of data (Figure 4) we can engage ourselves in some hypothetical treatment design based on this rough adequacy criterion and our selection rules. If we are dealing with 1/4" steel plating which is found to be resonating at 160 cps, we can read off the suitable treatments directly at 150 db/sec; i.e., a certain multilayer felt cemented on, and any of several weights of septum-loaded fibrous blankets. If we are dealing with 20-gauge sheet steel found to be resonating around 80 cps, as in the case of modern automobile panels, we reason as follows according to our rules: the mass adjustment depends only on thickness because the material is steel as for the test panel, 20-gauge steel is about five "octaves" thinner than the 1/4" test panel thereby allowing a reduction of decay rate by $(2)^5$ but the frequency of resonance is an octave lower than test frequency thereby requiring a doubling of decay rate, so that suitable treatments lie along the 10 db/sec line; i.e. $150/(2)^4$. This qualifies the mastic deadeners, light asphalted felts and light fibrous blankets, in nice accord with current practice. And so goes the matter of treatment design for overall treatments for panels of various weight and varying natural frequency.

There is no space to go into the interesting complications of practical treatment design where considerations of cost and weight and temperature sensitivity make the task really challenging, except to mention briefly a combination of parameters that requires very high decay rates and yet does not permit of high treatment weights; e.g., the damping of low-frequency resonances in aircraft fuselage panels. These applications introduce need for localization of damping material at those points where they will do the most good; i.e. at panel antinodes where vibration is most intense so that maximum motion is communicated to the damping material which then produces the most energy dissipation per pound of treatment. A new variant of of spot treatment is the frequency-selective configuration of damping material mentioned earlier. Its properties are summarized in Figure 5, where damping capacity is now plotted against frequency to compare two configurations of the same damping material. The peaking family of curves shows the damping capacity of ten square inches of a material in the frequency-selective spot configuration with different tuning masses attached, whereas the flat curve shows the damping capacity of 400 square inches of the same material applied in the standard fashion. The great improvements in low-frequency performance are very striking, especially when the economy of weight is also taken into account.

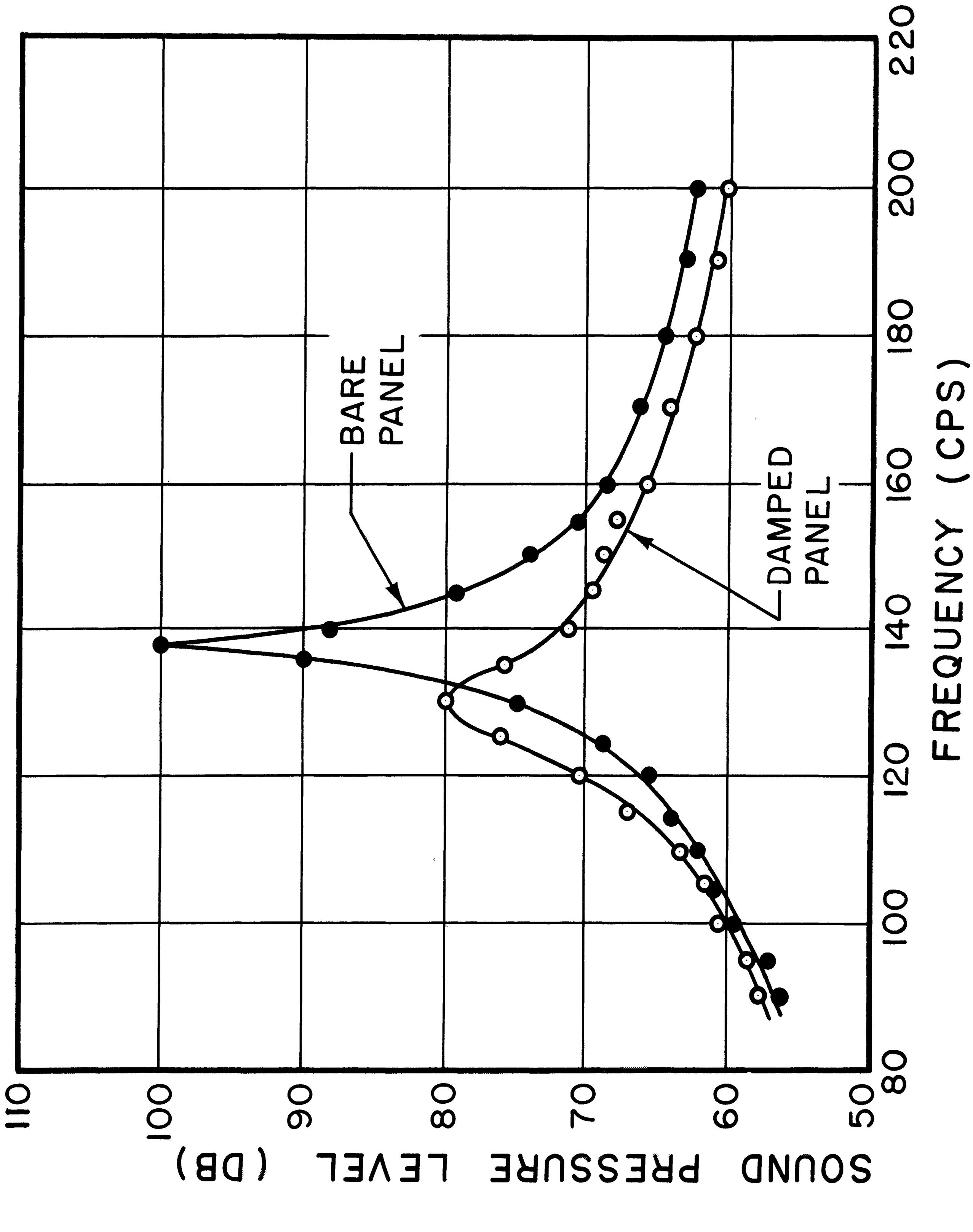

BARE PANEL
DAMPED PANEL
FREQUENCY (CPS)
SOUND PRESSURE LEVEL (DB)
80
100
120
140
160
180
200
220
50
60
70
80
90
100
110

FIGURE 1

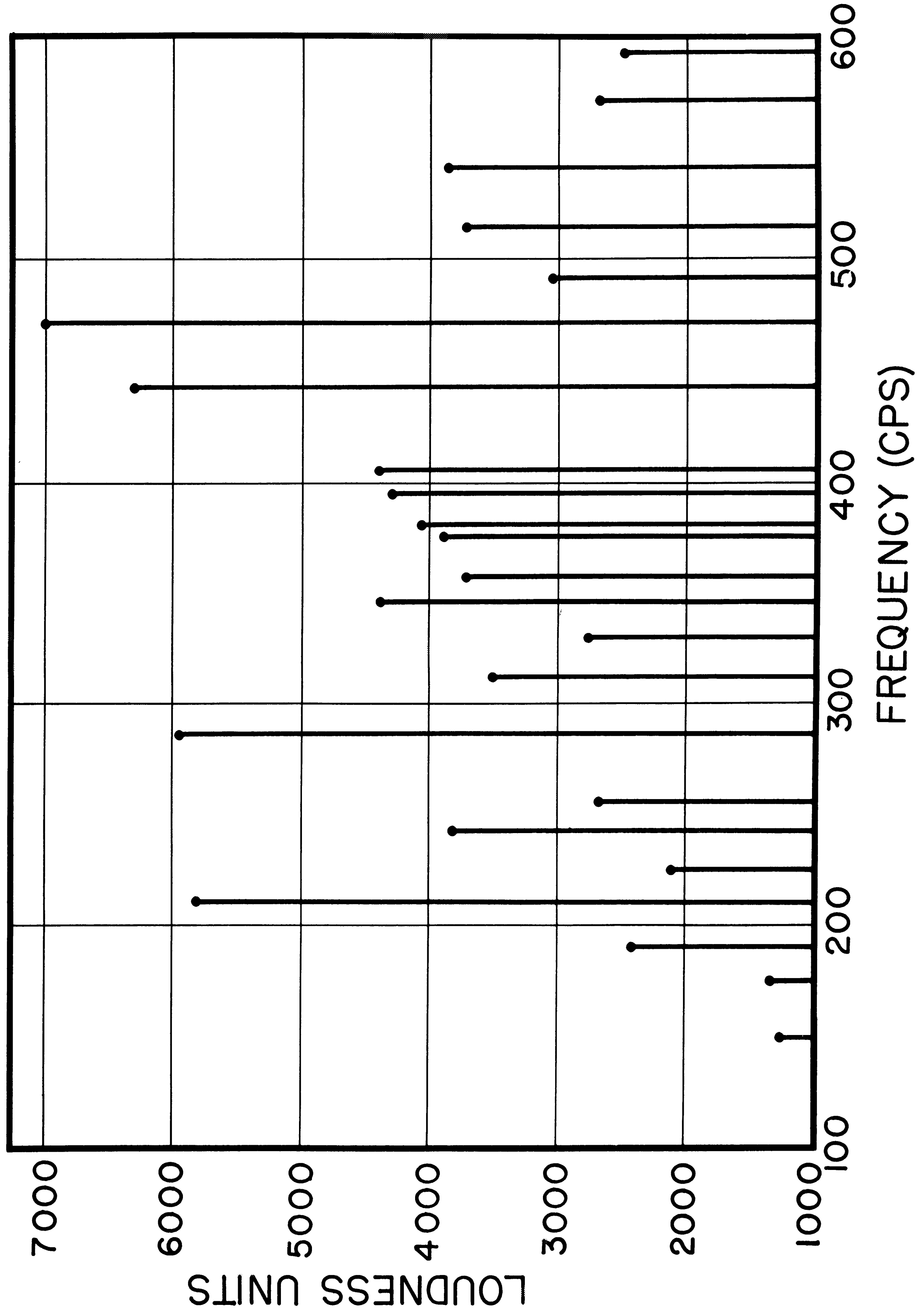
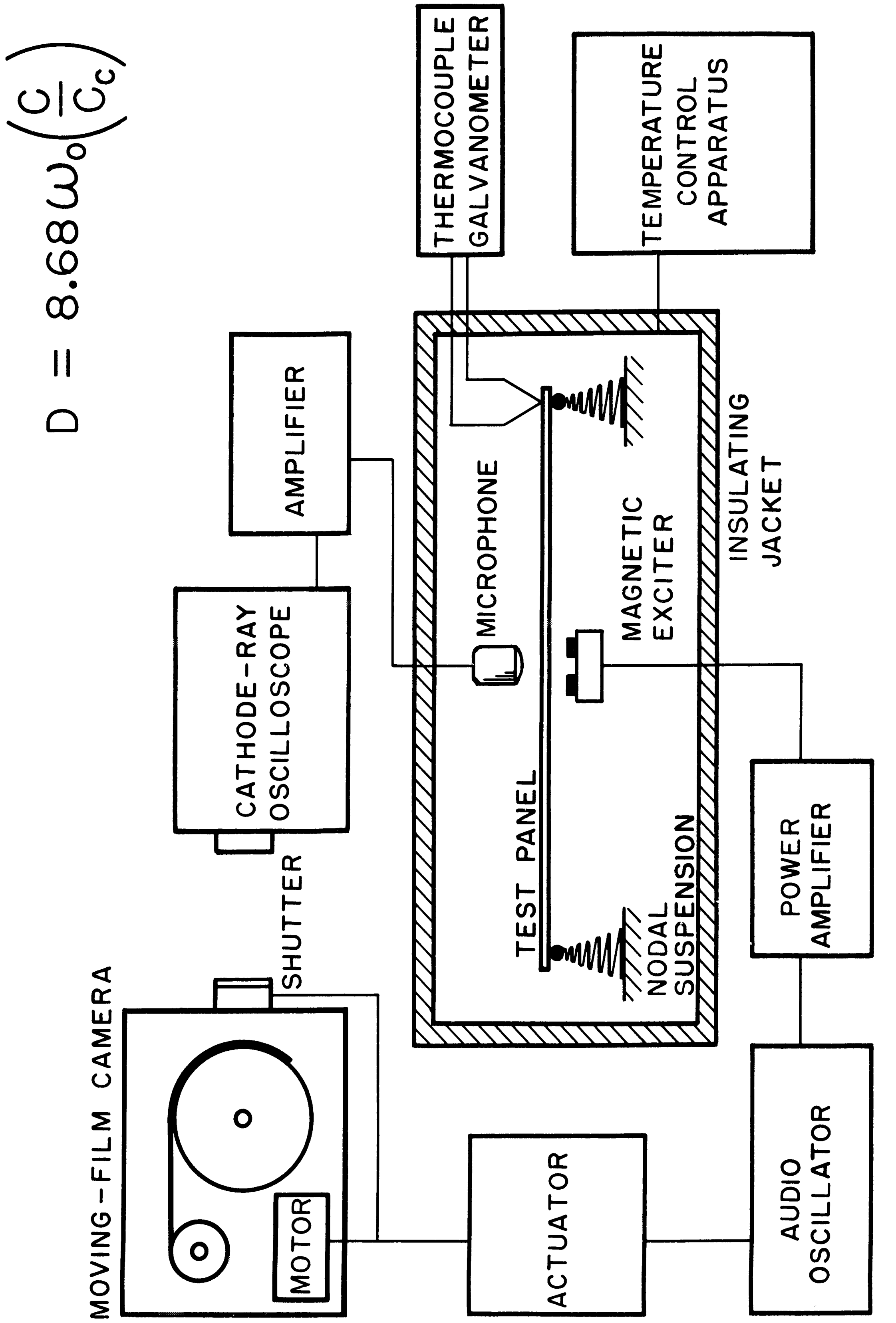
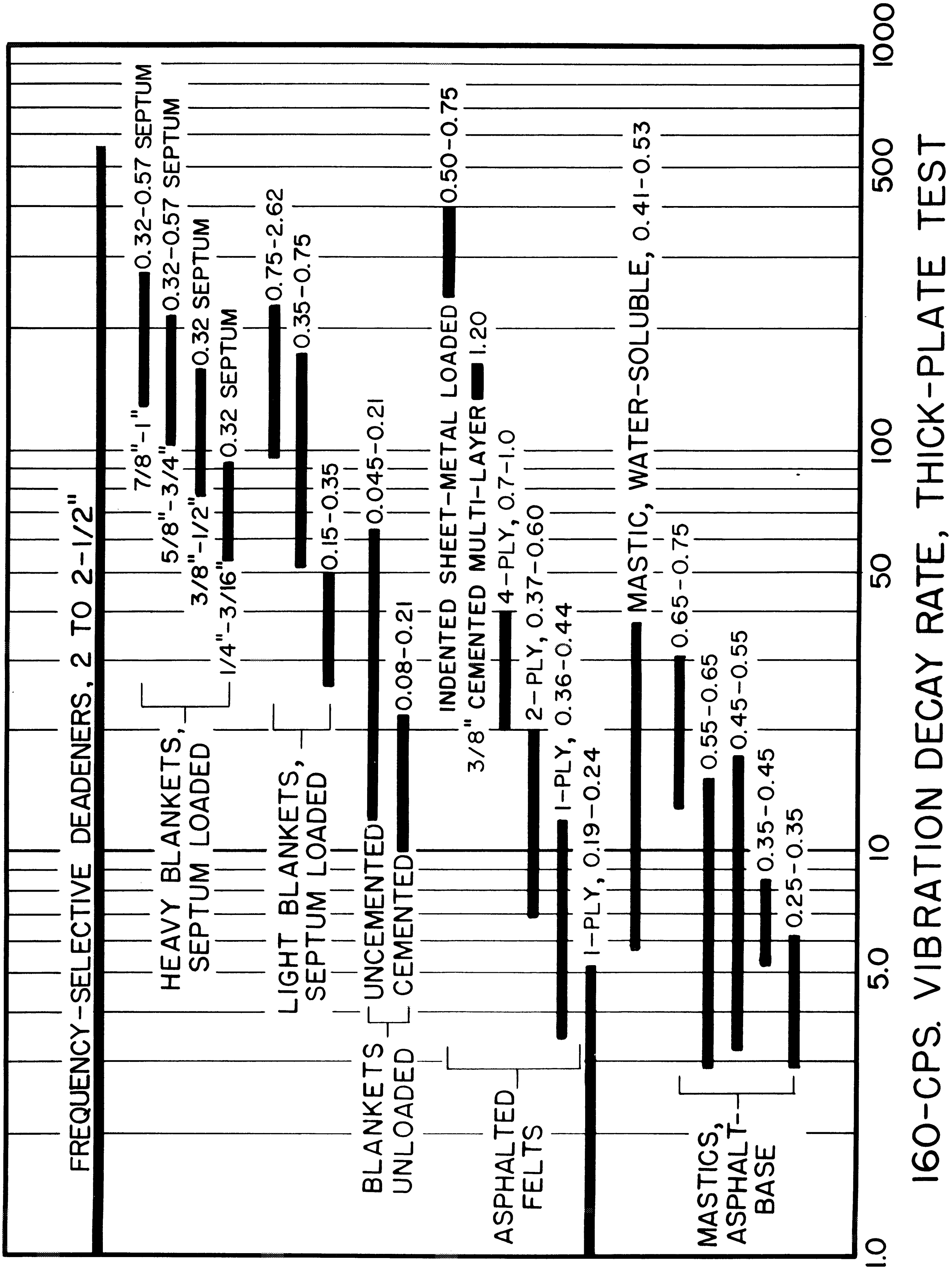

LOUDNESS UNITS
7000
6000
5000
4000
3000
2000
1000
FREQUENCY (CPS)
100
200
300
400
500
600

FIGURE 2



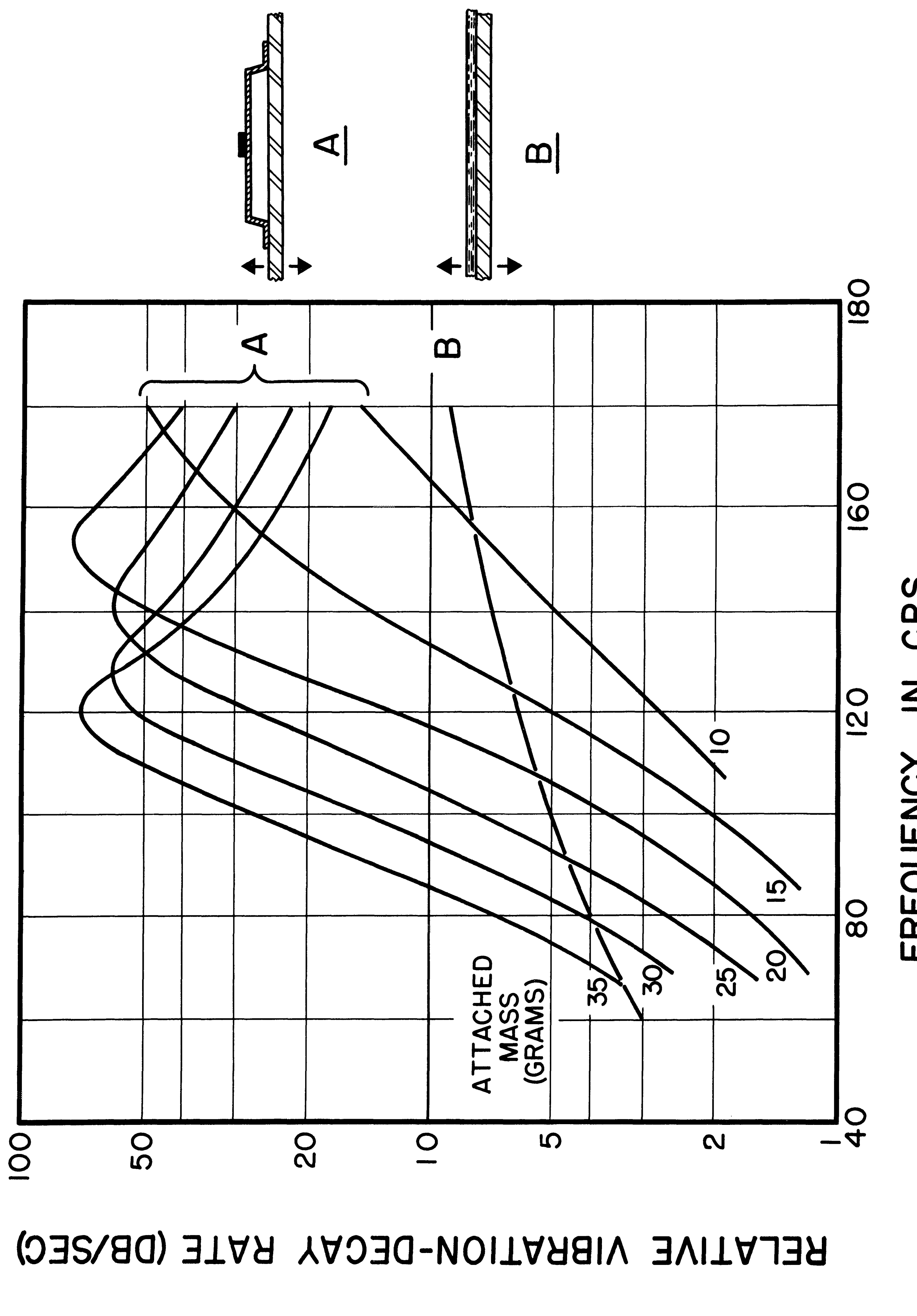
D = 8.68 ω₀ (C/Cc)
THERMOCOUPLE GALVANOMETER
TEMPERATURE CONTROL APPARATUS
AMPLIFIER
MICROPHONE
MAGNETIC EXCITER
INSULATING JACKET
CATHODE-RAY OSCILLOSCOPE
TEST PANEL
NODAL SUSPENSION
SHUTTER
POWER AMPLIFIER
MOVING-FILM CAMERA
MOTOR
ACTUATOR
AUDIO OSCILLATOR

FIGURE 3



160-CPS. VIBRATION DECAY RATE, THICK-PLATE TEST
1.0
5.0
10
50
100
500
1000
FREQUENCY-SELECTIVE DEADENERS, 2 TO 2-1/2"
HEAVY BLANKETS, SEPTUM LOADED
7/8"-1" 0.32-0.57 SEPTUM
5/8"-3/4" 0.32-0.57 SEPTUM
3/8"-1/2" 0.32 SEPTUM
1/4"-3/16" 0.32 SEPTUM
LIGHT BLANKETS, SEPTUM LOADED
0.75-2.62
0.35-0.75
0.15-0.35
BLANKETS UNLOADED
UNCEMENTED 0.045-0.21
CEMENTED 0.08-0.21
INDENTED SHEET-METAL LOADED 0.50-0.75
3/8" CEMENTED MULTI-LAYER 1.20
ASPHALTED FELTS
4-PLY, 0.7-1.0
2-PLY, 0.37-0.60
1-PLY, 0.36-0.44
1-PLY, 0.19-0.24
MASTIC, WATER-SOLUBLE, 0.41-0.53
0.65-0.75
MASTICS, ASPHALT-BASE
0.55-0.65
0.45-0.55
0.35-0.45
0.25-0.35

Figure 4



RELATIVE VIBRATION-DECAY RATE (DB/SEC)
FREQUENCY IN CPS
100
50
20
10
5
2
1
40
80
120
160
180
ATTACHED MASS (GRAMS)
35
30
25
20
15
10
A
B
A
B

FIGURE 5