How to Brace a Speaker Cabinet—

Vibration Reduction in Loudspeaker Enclosures

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Presenting the reasons for bracing a bass-reflex loudspeaker cabinet, and showing how to do it with the assurance of improved performance when the job is completed.

Benjamin Franklin once observed that "Empty barrels make the loudest noise." In this case he was referring obliquely to the common phenomenon of uninformed vociferation, not describing the performance of a loudspeaker enclosure. G. A. Briggs, in the second chapter of "Sound Reproduction," was commenting on the latter when he wrote: "The indications are that the effect of cabinet resonance has been underestimated in the past." He observed that the tone-quality of reproduced sound was greatly improved when the loudspeaker cabinet was constructed of materials having a high density. In a paper presented before the IRE PGA[1], Frank McIntosh pointed out that "boomy" sounds are caused by acoustic radiation due to decaying vibration of the panels in a poorly braced cabinet.

Briggs offers one solution to this problem—make the panels massive and they won't vibrate. The principle involved is that of relative momenta. Consider the effect of a moving mass of air striking a panel. Referring to *Fig.* 1, if a unit volume of air having a mass M_1, and an instantaneous maximum velocity V_1 strikes a unit volume of panel having a mass M_2, initially at rest, both masses will have a resulting velocity V_2. This relationship may be written:

$$M_1V_1^2 - M_2 0^2 = (M_1 + M_2)V_2^2 \quad (1)$$

Note that for the optimum condition, V_2 approaching zero (panel does not move) it is necessary to have the ratio of M_2/M_1 as large as possible. Since M_1, the mass of air is fixed at roughly 1/13 lb./cu. ft., M_2 remains the only variable. By varying M_2, it is possible to obtain the ratios shown in Table I for several different materials. It appears obvious from a glance at this table that a small improvement in V_2 (hence a reduction in vibration) may be had only at the expense of a large increase in weight. For example, a panel made of concrete would tend to vibrate (other things such at the modulus of elasticity being equal) 1/4.5 times as much as one made of wood, but would weigh 20 times as much. Thus a typical bass reflex cabinet weighing 50 pounds constructed of wood, would weigh 1000 made of concrete—probably too much for the average living room floor to support. Even if this were permissible, such an enclosure would be virtually immovable and would present the baffling (no pun intended) problem mentioned in a recent editorial.[2]

TABLE I

Physical Properties of Common Construction Materials

Material	Mass lbs./ft.³	Thick. In.	M_2 lbs./ft.²	M_2/M_1	V_1/V_2
Dry packed sand	105	1	8.75	114	10.7
Brick	125	3	31.2	406	20.2
Concrete	150	3	37.5	488	22.1
Plaster	—	1	8.0	104	10.2
White Pine	26	⅞	1.9	24.7	4.9
White Oak	46	⅞	3.36	43.8	6.6
2-⅞ wood panels with 1" sand between			12.55	163.5	12.8

Fortunately for the cabinet designer there is another solution to the problem of reducing panel vibration. Instead of relying on weight alone to accomplish the desired results, he can make the panels stiff, and join them rigidly together.

Stiffness of Panels

Several factors determine the stiffness of a panel. The chief factor of course is the geometry of the panel. In most cases it is very difficult to analyze the behavior of a vibrating plate, especially if one attempts to relate various design parameters to a resulting acoustic output. It is entirely beyond the scope of this discussion to examine these theoretical considerations in minute detail. Furthermore, it can be shown that a much simpler method of analysis provides the essential information necessary to make very substantial improvements in cabinet construction.

For all practical purposes it is reasonably sufficient to consider a panel as made up of an infinite number of small beams arranged side by side as shown in *Fig.* 2. Notice that in this type of analysis, the beams are represented as extending across the shorter dimension of the panel. Assuming the edges of the panel are supported, it is logical to suppose that the beam exhibiting the most severe deflection when subject to a load, will be one near the center of the panel such as beam A. Now, without attempting to determine an exact coefficient for the stiffness factor, it can be shown that the maximum deflection of beam A is dependent on a few easily determined variables. Actually, since the beams are integral parts of a homogeneous plate, the deflection will be somewhat less than that of a single unattached beam.

The equation for the deflection of a rigidly supported, uniformly loaded beam may be written as

$$Z = \frac{W L^3}{384 E I} \quad (2)$$

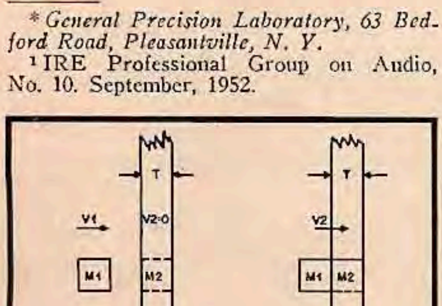

Fig. 1. Kinetic energy effect of Unit Volume of air M_1, impinging on Unit Area of panel, M_2.

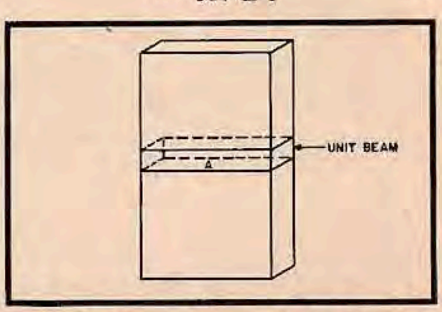

Fig. 2. Typical panel as used in the discussion.

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[1] IRE Professional Group on Audio, No. 10. September, 1952.

[2] Editor's Report, Audio Engineering, March, 1953.

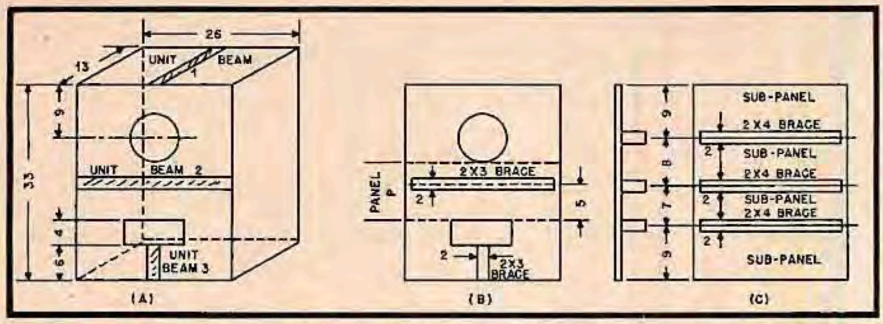

Fig. 3. Typical bass-reflex enclosure, with dimensions as used in the sample problem. All stiffening beams are mounted so that their maximum cross-section dimension is perpendicular to the plane of the panel.

in which Z equals the maximum deflection, W equals the load on the beam (maximum instantaneous value), and I equals the moment of inertia of the cross-section of the beam. If this equation is compared with that for a non-rigidly supported beam, in which case

$$Z = \frac{5\,W\,L^3}{384\,E\,I} \qquad (3)$$

it will be observed that the deflection is five times the magnitude of the former. Of course in actual practice these extremes are almost never encountered; no panel however loosely secured would exhibit a vibration five times as severe as one firmly attached to an immovable support. Nevertheless, this simple comparison emphasizes the need for rigid support of the panels.

The maximum deflection of such a beam may also be reduced by decreasing its length as far as is practicable. On the other hand, little benefit is derived from attempting to vary the value of E; reference to appropriate tables reveals that the modulus of elasticity of commonly used lumber varies from about 1.00×10^6 to 1.6×10^6.

The moment of inertia of these beams equals $bd^3/12$, where b is the width of the beam and d is its thickness. (See Table II) It is interesting to note here that doubling the thickness of a panel (and hence the thickness of a unit-width beam) reduces the deflection by a factor of eight; tripling the thickness reduces deflection by a factor of 27; and so on. Before demonstrating a typical solution to a cabinet design problem, it will be found helpful to introduce one further beam equation into the discussion. In the case where a panel contains an opening (speaker mounting hole or reflex port), the beams which have one termination at an opening are classified as cantilevers. The deflection for this type of beam is written as:

TABLE 2

Moment of Inertia about the Center of Gravity for Common Lumber Sizes (max. Value)

Nominal Size In.	Dressed or Finished Size In.	Moment of Inertia In.[4]
1 × 1	⅞ × ⅞	.049
1 × 2	⅞ × 1⅝	.31
1 × 3	⅞ × 2⅝	1.32
2 × 3	1⅝ × 2⅝	2.45
1 × 4	⅞ × 3⅝	3.44
2 × 4	1⅝ × 3⅝	6.45

$$Z = \frac{W\,L^3}{8\,E\,I} \qquad (4)$$

For convenience, the terms in the equations which remain more-or-less constant in any one design, are W and E, and they may be lumped into one constant, K. The equators are then rewritten as:

Simple beam—no end supports

$$Z = \frac{5\,K\,L^3}{384\,I} \qquad (5)$$

Simple beam—fixed end supports

$$Z = \frac{K\,L^3}{384\,I} \qquad (6)$$

Cantilever

$$Z = \frac{K\,L^3}{8\,I} \qquad (7)$$

Figure 3 and the sample problem show the suggested method of analysis

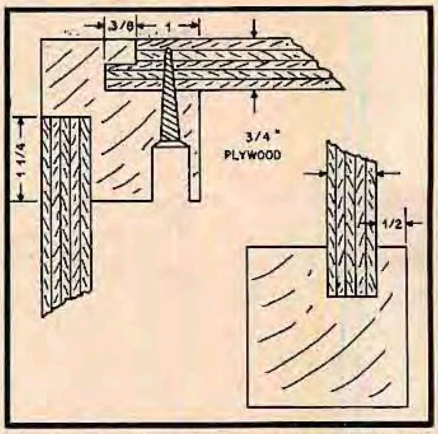

Fig. 4. Examples of cornerpost construction.

as applied to a typical cabinet. Cabinets of different shapes will present stiffness problems somewhat different from the sample illustrated. In this case, the analysis is used to determine the size, shape, and number of braces required to improve the performance of a cabinet originally constructed without full consideration of the factors just discussed. Naturally the need for extensive bracing is reduced if the enclosure is properly designed from the start.

Construction

There is more to the problem of designing a rigid speaker cabinet than mere choice of adequate panel thickness and arrangement of bracing, however. As was observed from the discussion of the beam equations, above, the method of supporting the panels is highly important. Since most panels are made of plywood, it is necessary to insure that all joints are constructed to achieve the maximum possible rigidity. A simple but effective method for accomplishing this is to use corner posts to which the panels are securely anchored. A few typical joints of this type are illustrated in *Fig.* 4. It will be found that such methods also contribute to the over-all appearance of the finished structure.

At first glance, the prospective builder may be somewhat dismayed by the suggestion that special tools are needed to prepare the joints for assembly. This type of work is best done using a joiner or router, although the patient craftsman may use a plane with very good results. In many instances the whole problem can be greatly simplified by having a local mill cut each piece to size and prepare the joining surfaces from drawings furnished by the designer. The task, then, is merely one of assembling the finished parts.

During this assembly operation it is best to screw and glue all joints together so that the finished cabinet will be tight and solid. The front panel of the enclosure can also be permanently anchored if the loudspeaker is mounted as shown in *Fig.* 5. It is fairly well recognized that speaker performance is greatly improved by mounting the speaker in this manner. As illustrated, an alternate method using two superimposed panels obviates the necessity for special routing. In either case captive nuts attached to the back of the panel receive the speaker mounting bolts from the front. Many loudspeaker manufacturers will, upon request, supply suitable gaskets for this type of mounting.

Conclusion

It has been the purpose of this discussion to analyze the problem of spurious acoustic output caused by excessive panel vibrations in a loudspeaker enclosure, and to suggest an approach which will result in a definite reduction in the effect and a vast improvement in the over-all performance. As might be

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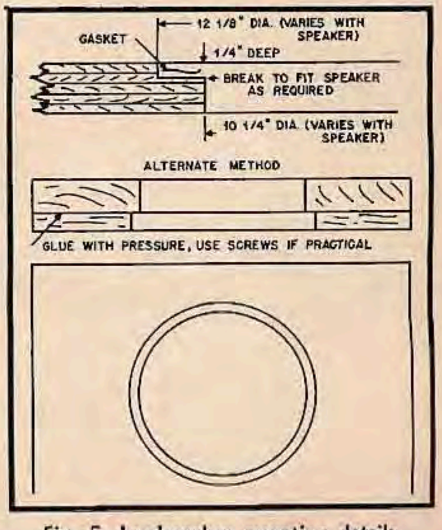

Fig. 5. Loudspeaker mounting details.

VIBRATION REDUCTION

(*from page 25*)

expected, the vibrations dealt with are those which occur at low frequencies. The various equations do not take resonance into account but explain the phenomenon below resonance where dynamic and static deflections are nearly the same. If one remembers that the natural resonant frequency of a solid varies inversely with its weight and directly with its stiffness, it is obvious to conclude that constructing panels as rigidly as possible helps to raise the resonant frequency to a region where damping is more easily accomplished by the use of padding and sound absorbing materials. This is easily demonstrated with the test set up illustrated in *Fig.* 6. Connected in this manner, the oscilloscope indicates the power factor of the load. At resonance, the power factor is unity and a straight line appears on the scope. Power factors at non-resonant frequencies show up as loops of various widths. If the oscillator is adjusted until a panel resonance is detected, it may be observed that the application of hand pressure to the vibrating panel will cause the straight line to open up into a loop, and by returning the oscillator it will be found that the resonant frequency occurs at a higher fre-

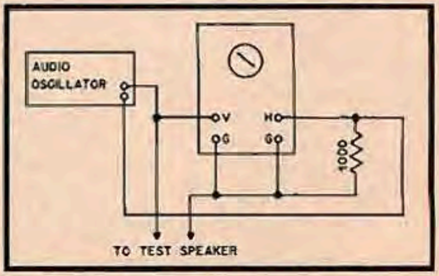

Fig. 6. Test circuit used for checking impedance and resonant frequency of speaker in enclosure.

quency. During this test, of course, one must not pick an oscillator frequency which corresponds with the natural acoustical resonance of the cabinet. In this case applying pressure to the panels will have little effect.

SAMPLE PROBLEM

Consider the bass reflex cabinet shown at (A) in *Figure* 3. Assume that it is constructed of 1-in. plywood (⅞-in. dressed), and that all corners are joined in such a manner that the panels are rigidly supported.

The top panel will exhibit the smallest maximum deflections because it is the smallest panel. The deflection of unit beam 1 is given by equation (6).

$$Z_{max} = \frac{K\ L^3}{384\ I}$$

In this case *I*, the moment of inertia, from Table II is .049 and

$$Z_{max} = \frac{K\ 13^3}{384 \times .049} = 117\ K$$

Since unit beams chosen from the bottom or sides would have identical dimensions, the maximum deflection of these panels would be the same as the top panel.

The maximum deflection of the front panel is determined by considering unit beam 2. In this case

$$Z_{max} = \frac{K\ 26^3}{384 \times .049} = 936\ K$$

The deflection of the back panel is similar.

Because of its shape, the area of the front panel below the port (6×26 in.) exhibits a deflection which is characteristic of unit beam 3, which is a cantilever. The deflection is given by

$$Z_{max} = \frac{K\ L^3}{8\ I} = \frac{K\ 6^3}{8 \times .049} = 551\ K$$

In determining the bracing required to reduce front and back panel vibration to a value consistent with the top panel, consider the load as the area defined by panel "P" in (B). Its width is roughly 10 inches. Since a unit beam has a width of 1 inch, and the deflection is 936/117 or roughly 8 times as severe as for unit beam 1, the required stiffness of the brace beam is 8×10 or 80 times that of unit beam 2. From Table II it may be seen that the best brace would be a 1×4. The stiffness factor equals 3.49/.049, or 71, which is close enough to the desired value. Actually it may be more desirable to use a 2×3 as shown at (B) with a stiffness factor of 245/.049, or 50, in some instances where a brace 1×4 would adversely affect the acoustics of the cabinet. Since such effects are generally not serious near the back of the cabinet, it is recommended that 2×4's be used. Here the stiffness factor is 6.45/.049, or 132. If three braces are used as shown at (C), each will support approximately one fourth of the total load. This makes the stiffness factor equal to $8 \times \frac{33K}{4}$ or 66 K. If the above calculations are applied successively to the braced panel shown at (C), two deflections are obtained.

Since both sub-panels and braces vibrate in unison, the final deflection is obtained by adding the two and the sum will be found to equal 117 K.

Because of the short length of unit beam 3, sub-panel Q (A of *Fig.* 3) requires less bracing than sub-panel P. Just to be on the safe side however, a 2 × 3 should be installed as shown.

The reader will note that all the braces used add up to a total volume of ½ cu. ft. which is about 1/12 of the cabinet volume.