Vented-Box Loudspeaker Systems Part II: Large-Signal Analysis

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The power capacity of a vented-box loudspeaker system is shown to be directly related to the system frequency response and to the volume of air that can be displaced by the system driver. The vent area must be made large enough to prevent noise generation or excessive losses; the required area is shown to be quantitatively related to enclosure tuning and to driver displacement volume. Mutual coupling between driver and vent is found to be of negligible importance in most cases.

The basic performance characteristics of a vented-box system may be determined from knowledge of a number of fundamental system parameters. These parameters can be evaluated from relatively simple measurements. The vented-box system is shown to possess two important performance advantages compared with the closed-box system.

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6. DISPLACEMENT-LIMITED POWER RATINGS

Diaphragm Displacement

The vented-box system displacement function given by Eq. (14) is a low-pass filter function which has a notch at f_B contributed by the numerator and an ultimate cutoff slope of 12 dB per octave at high frequencies. The behavior of this function is examined at the end of Appendix 1.

The normalized diaphragm displacement magnitude $|X(j\omega)|$ is plotted in Fig. 17 for a few common alignments. For convenience, the frequency scale is normalized to f_B. Note that the effect of moving from the C4 alignments toward the QB3 alignments (i.e., increasing α) is to reduce the diaphragm displacement near and above f_B relative to the displacement at zero frequency, and that the principal effect of enclosure losses is to increase the displacement near f_B, i.e., reduce the sharpness of the notch.

Acoustic Power Rating

Assuming linear large-signal diaphragm displacement, the steady-state displacement-limited acoustic power rating P_{AR} of a loudspeaker system, from [12, eq. (42)], is

$$P_{AR} = \frac{4\pi^3\rho_0}{c} \cdot \frac{f_S{}^4 V_D{}^2}{k_x{}^2 |X(j\omega)|_{\max}{}^2} \qquad (36)$$

where $|X(j\omega)|_{\max}$ is the maximum magnitude attained by the displacement function and V_D is the peak displacement volume of the driver diaphragm, given by

$$V_D = S_D x_{\max} \qquad (37)$$

x_{max} being the peak linear displacement of the driver diaphragm, usually set by the amount of voice-coil overhang.

For the vented-box system, Eq. (15) gives $k_x = 1$. The displacement-limited acoustic power rating of the vented-box system then becomes

$$P_{AR(VB)} = \frac{4\pi^3\rho_0}{c} \cdot \frac{f_S{}^4 V_D{}^2}{|X(j\omega)|_{max}{}^2}. \tag{38}$$

For SI units, the value of $4\pi^3\rho_0/c$ is 0.424.

Power-Rating Constant

Eq. (38) may be written in the form

$$P_{AR(VB)} = k_P f_3{}^4 V_D{}^2 \tag{39}$$

where k_P is a power-rating constant given by

$$k_P = \frac{4\pi^3\rho_0}{c} \cdot \frac{1}{(f_3/f_S)^4 |X(j\omega)|_{max}{}^2}. \tag{40}$$

The value of f_3/f_S is already established for any alignment in the C4–B4–QB3 range. But from Fig. 17, $|X(j\omega)|$ has *two* maxima. The first occurs outside the system passband; this has a value of unity and is located at zero frequency for the QB3, B4, and moderate C4 alignments but slightly exceeds unity and is located below f_B for the extreme C4 alignments. The second maximum occurs within the system passband, above f_B, and is always smaller than the first.

There are thus two possible values for k_P, one if the system driving signal is allowed to have large-amplitude components at frequencies well below cutoff, and another, which is substantially larger, if the signal is restricted so that all significant spectral components are within the system passband.

Fig. 18 is a plot of the values of k_P for each of the above driving conditions as a function of the alignment parameters k and B for systems with lossless enclosures. The crosses in Fig. 18 indicate the values of k_P for a few selected alignments with $Q_L = 5$. The effect of this relatively severe amount of enclosure loss on k_P is negligible for the QB3 alignments but gradually increases as the extreme C4 alignments are approached. For these alignments, k_P is slightly reduced for the passband-drive case but slightly increased for the wideband-drive case.

Program Acoustic Power Rating

In most program applications, a portion of the driving signal spectrum lies below the system passband. The lower value of k_P given by Fig. 18 is then in general conservative, while the higher value is comparatively optimistic. A truly realistic value of k_P for program material can be evaluated only if the actual spectral power distribution of the particular driving signal is known. Thiele for example has obtained comparative power handling data for a number of system alignments (including amplifier-assisted alignments) based on a particular random-noise driving signal [20].

In most cases, provided that the program spectrum is principally within the system passband, a satisfactory program rating is obtained by setting k_P equal to 3.0, regardless of the alignment used. This is indicated by the broken line in Fig. 18. This compromise value for k_P is arrived at by considering, for the entire range of alignments, the passband and wideband values of k_P, the ratio of maximum displacements for passband- and wideband-drive conditions, and the degree to which the driving signal spectrum may extend below system cutoff before the displacement exceeds the passband maximum (see Fig. 17).

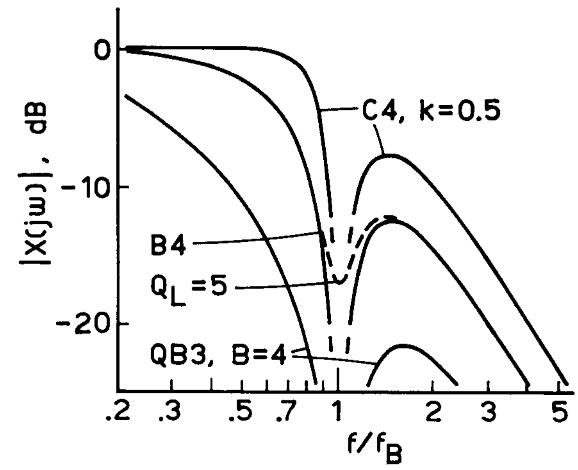

Fig. 17. Normalized diaphragm displacement of vented-box system driver as a function of normalized frequency for several typical alignments (from simulator).

With this value of k_P, Eq. (39) becomes

$$P_{AR(VB)} = 3.0 f_3{}^4 V_D{}^2. \tag{41}$$

This relationship is generally applicable to all vented-box alignments for which the system passband includes the major components of the program signal spectrum. Whenever the signal and alignment properties are accurately known, a more exact relationship may be obtained with the help of Fig. 18 or by using Eq. (38) directly.

Power Output, Cutoff Frequency, and Displacement Volume

Eq. (41) is illustrated in Fig. 19. P_{AR} is expressed in both watts (left scale) and equivalent sound pressure

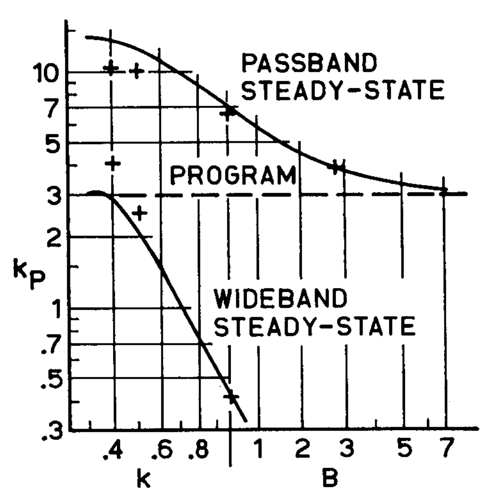

Fig. 18. Power rating constant k_P for vented-box loudspeaker system as a function of response shape. Solid lines are for lossless systems; crosses represent systems with $Q_L = 5$.

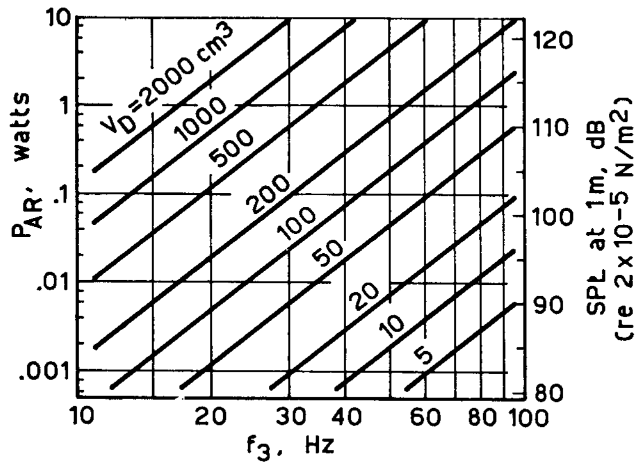

Fig. 19. Relationship between cutoff frequency, driver displacement volume, and rated acoustic power for a vented-box loudspeaker system operated on program material.

level (SPL) at 1 meter [3, p. 14] for 2π-steradian free-field radiation conditions (right scale). This is plotted as a function of f_3 for various values of V_D (note 1 $cm^3 = 10^{-6}\ m^3$). The SPL at 1 meter given on the right-hand scale is a rough indication of the SPL produced in the reverberant field of an average listening room for a radiated acoustic power given by the left-hand scale [3, p. 318]. For particular listening environments such as large halls, the reference just cited gives methods for computing the acoustic power required to obtain a specified SPL.

Fig. 19 represents the approximate physical large-signal limitation of vented-box system design. It may be used to determine the maximum performance tradeoffs (P_{AR} versus f_3) for a given voice-coil/suspension design or to find the minimum value of V_D which is required to meet a given specification of f_3 and P_{AR}.

Power ratings calculated from Eq. (41) or Fig. 19 apply only for "typical" program material which does not drive the system hard at frequencies below cutoff. For other circumstances the applicable rating may be higher or lower. Even where the condition of passband drive is met with regard to the intended program material, the vented-box system is clearly vulnerable to extraneous signals such as turntable rumble and subsonic control tones. These normally inaudible signals may produce audible harmonics or cause noticeable modulation distortion [21]. In cases where such signals are particularly troublesome and cannot otherwise be eliminated, the use of a closed-box design or one of the higher order amplifier-assisted vented-box alignments described by Thiele [10], [20] may provide relief.

Electrical Power Rating

The displacement-limited electrical power rating P_{ER} of the vented-box system is obtained by dividing the acoustic power rating Eq. (38) by the system reference efficiency Eq. (25). Thus,

$$P_{ER(VB)} = \frac{P_{AR(VB)}}{\eta_0} = \pi\rho_0 c^2 \frac{f_S Q_{ES}}{V_{AS}} \cdot \frac{V_D^2}{|X(j\omega)|_{\max}^2}. \quad (42)$$

This rating is subject to the same adjustments for program material as used above. Its dependence on the performance factors already discussed is easily observed from the form obtained by dividing Eq. (39) by Eq. (26):

$$P_{ER} = \frac{k_P}{k_\eta} f_3 \frac{V_D^2}{V_B}. \quad (43)$$

In practice, the values of P_{AR} and η_0 are much more important; these would normally be specified or calculated first. P_{ER} is then obtained directly from these numbers as indicated by Eq. (42). P_{ER} describes only the amount of nominal power which may be absorbed from an amplifier if thermal design of the voice-coil permits. It gives no indication of acoustic performance unless reference efficiency is known.

Enclosure and driver losses reduce η_0 without much effect on P_{AR} and thus lead to a higher value of P_{ER}. Driver displacement nonlinearity for large signals also has the effect of reducing efficiency at high levels, i.e., increasing the electrical input required to actually reach the driver displacement limit. In both cases, the extra input power is only dissipated as heat.

7. PARAMETER MEASUREMENT

The direct dependence of system performance characteristics on system parameters provides a simple means of assessing or predicting loudspeaker system performance from a knowledge of these parameters. The important small-signal parameters can be found with satisfactory accuracy from measurement of the voice-coil impedance of the system and its driver.

The voice-coil impedance function of the vented-box system is given by Eq. (16). A plot of the steady-state magnitude $|Z_{VC}(j\omega)|$ of this function against frequency has the shape illustrated in Fig. 20; the measured impedance curve of a practical vented-box system has this same characteristic shape.

The impedance magnitude plot of Fig. 20 has a minimum at a frequency near f_B (labeled f_M) where the impedance magnitude is somewhat greater than R_E. The additional resistance is contributed primarily by enclosure losses and is designated R_{BM} on the plot axis. There are two maxima in the impedance plot, located at frequencies below and above f_M. These are labeled f_L and f_H. At these frequencies, the magnitudes of the impedance maxima depend on both driver losses and enclosure circuit losses and are seldom equal.

Where only normal enclosure losses are present, the basic system parameters and the total enclosures loss Q_B may be found with satisfactory accuracy using the method developed by Thiele in [10]. The indicated value of Q_B may then be used to check the measurement approximations. Thiele's method is based on an initial assumption of negligible enclosure losses and may be summarized as follows. The relationships are derived in Appendix 2.

1) Measure the three frequencies f_L, f_M, and f_H where the impedance magnitude is maximum or minimum. The accurate identification of these frequencies may be aided by measuring the impedance phase; if this passes through zero at the appropriate maximum or minimum, the frequency of zero phase (which may be located with high precision) may be taken as the center of the maximum or minimum. However, if zero phase is not closely coincident with maximum or minimum magnitude, as may occur for moderate to high enclosure losses, the frequency of actual maximum or minimum impedance mag-

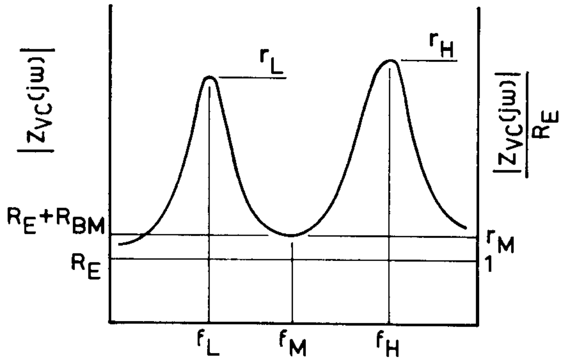

Fig. 20. Voice-coil impedance magnitude of vented-box loudspeaker system as a function of frequency.

nitude must be located as carefully as possible. Experience with many systems and experiments with the analog circuit simulator have shown that where the frequencies of zero phase and maximum or minimum magnitude do not coincide, the latter always provide more accurate values of the system parameters. Bypass any crossover networks for this measurement, and keep the measuring signal small enough so that both voltage and current signals are undistorted sinusoids. For the following calculations, assume that $f_B = f_M$.[2]

2) Calculate f_{SB}, the resonance frequency of the driver for the air-load mass presented by the enclosure, from the relationship

$$f_{SB} = \frac{f_L f_H}{f_B}. \tag{44}$$

3) Calculate the compliance ratio α from the relationship

$$\alpha = \frac{(f_H + f_B)(f_H - f_B)(f_B + f_L)(f_B - f_L)}{f_H^2 f_L^2}. \tag{45}$$

If the enclosure contains little or no lining material, the driver compliance equivalent volume V_{AS} may be calculated in terms of the enclosure net volume V_B. The relationship is, from Eqs. (9), (10), and (33),

$$V_{AS} = \alpha V_B. \tag{46}$$

4) Calculate the tuning ratio h from

$$h = f_B / f_{SB}. \tag{47}$$

5) Remove the driver from the enclosure, measure the driver parameters f_S, Q_{MS}, and Q_{ES} by the method of [12, Appendix],[3] and correct the driver Q values if necessary to correspond to the driver resonance frequency in the enclosure. This is done by multiplying the measured values of Q_{MS} and Q_{ES} by the ratio f_S/f_{SB}, where f_S is the resonance frequency for which Q_{MS} and Q_{ES} have been measured and f_{SB} is the resonance frequency in the enclosure found from Eq. (44). Usually if the driver parameters are measured on a test baffle of suitable size, the two resonance frequencies are almost identical and the correction is not required.

6) Calculate Q_{TS} from

$$Q_{TS} = \frac{Q_{ES} Q_{MS}}{Q_{ES} + Q_{MS}}. \tag{31}$$

7) Measure the minimum system impedance magnitude $R_E + R_{BM}$ at f_M and calculate

$$r_M = \frac{R_E + R_{BM}}{R_E}. \tag{48}$$

Then, using the corrected values of Q_{ES} and Q_{MS} obtained above, determine the total enclosure loss Q_B from the relationship

$$Q_B = \frac{h}{\alpha}\left[\frac{1}{Q_{ES}(r_M - 1)} - \frac{1}{Q_{MS}}\right]. \tag{49}$$

The term $1/Q_{MS}$ can usually be neglected.

8) The accuracy of the approximation $f_B \approx f_M$ on which the above method is based may be checked by calculating the approximate error introduced by the enclosure losses. Assuming that leakage losses are dominant in effect and that f_M is the measured frequency of zero phase, the error correction factor is

$$\frac{f_B}{f_M} = \sqrt{\frac{\alpha Q_B^2 - h^2}{\alpha Q_B^2 - 1}}. \tag{50}$$

This factor is usually quite close to unity. If it is significantly different from unity, it may be used to correct the value of f_B used in the above calculations to obtain better accuracy in the calculated parameter values.

The estimation or measurement of driver large-signal parameters is discussed in [22, Sec. 6].

With values determined for all important system parameters, system performance may be determined from the relationships given in earlier sections. The system frequency response may be calculated manually or using a digital computer but is most easily obtained by introducing the system parameters to an analog circuit simulator. The design of a simple simulator suitable for this purpose will be published in the future.

8. VENT REQUIREMENTS

The vent of a vented-box system must provide the necessary small-signal enclosure resonance frequency f_B; it must also provide the maximum required large-signal volume velocity without excessive losses or generation of spurious noises.

The second requirement can be satisfied by adjusting the vent area to a value which prevents the vent air velocity from exceeding a specified limit. An experimentally determined limit which avoids excessive noise generation is about 5% of the velocity of sound, provided that the inside of the vent is smooth and that the edges are rounded off with a reasonable radius. This velocity

[2] In [32, Appendix 4] Benson shows that if a large voice-coil inductance (or crossover inductance) is present, the measured value of f_M is lower than the true value of f_B, while f_L and f_H are negligibly affected. A much better approximation to f_B is obtained by carefully blocking the vent aperture and measuring the resonance frequency f_C of the resulting closed-box system [22]. Then, from [32, eq. (A4-6)], $f_B = (f_L^2 + f_H^2 - f_C^2)^{1/2}$. Because this relationship is true, f_C can be used directly in place of f_B in Eq. (45) to determine the system compliance ratio.

[3] Again, if the driver voice-coil inductance is large, Benson [32, Appendix 2] shows that the accuracy of determination of the Q values is improved if f_S in [12, eq. (17)] is replaced by the expression $\sqrt{f_1 f_2}$.

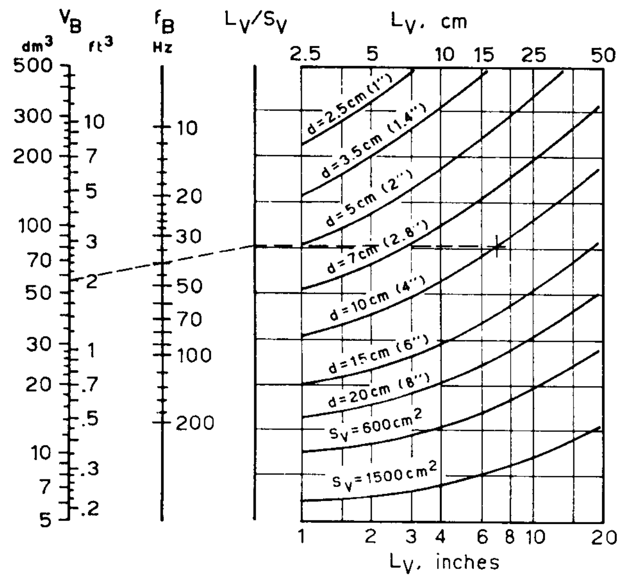

Fig. 21. Nomogram and chart for design of ducted vents.

limitation generally ensures acceptable losses as well, provided that the vent is not unduly obstructed.

The alignment, response, and power rating data of this paper combine to yield a relationship between vent area and maximum vent velocity for any given system. For program power ratings this relationship reduces to a simple approximate formula for vent area which limits the peak vent velocity, at maximum rated power input and at the frequency of maximum vent velocity, to 4½% of the velocity of sound. This formula, which is accurate within ±10% for the entire C4–B4–QB3 range of alignments, is

$$S_V \geqq 0.8\, f_B V_D \quad (51)$$

or

$$d_V \geqq (f_B V_D)^{1/2} \quad (52)$$

where S_V is the area of the vent in m^2 or d_V is the diameter of a circular vent in meters; V_D must be expressed in m^3 and f_B in Hz. Because the noise generated depends on factors other than velocity (e.g., edge roughness), and because the annoyance caused by vent noise is subjective, this formula should be regarded as a general guide only, not as a rigid rule.

Once the area of the vent is determined, the length must be adjusted to satisfy the first requirement, i.e., correct enclosure tuning. There are many popular formulas and nomograms for doing this. Using Thiele's formulas [10, eqs. (60)–(65)], the nomogram and chart of Fig. 21 were constructed to simplify the calculation process for ducted vents.

To use Fig. 21, lay a straight-edge through the enclosure volume on the V_B line and the desired resonance frequency on the f_B line and find the intersection with the L_V/S_V line. This is illustrated on the figure with lightly dashed lines for V_B = 57 dm^3 (2 ft^3) and f_B = 40 Hz. Next, move horizontally to the right from this intersection point until a curve is reached on the chart which corresponds to the required minimum size determined from Eqs. (51) or (52). The intersection of the horizontal projection with this curve indicates on the horizontal scale the required duct length L_V for a vent of the prescribed size. For the example illustrated, if the minimum duct diameter is 100 mm (4 inches), the required length is about 175 mm (7 inches). End corrections for one open end and one flanged end are included in the construction of the chart. For intermediate vent areas the chart may be interpolated graphically.

For some proposed systems a satisfactory vent design cannot be found. This is particularly the case for small enclosures when a low value of f_B is desired. Also, tubular vents for which the length is much greater than the diameter tend to act as half-wave resonant pipes, and any noise generated at the edge is selectively amplified. In these cases it is better to use a drone cone or passive radiator in place of the vent [2], [23]. Systems of this type will be discussed in a later paper.

9. DIAPHRAGM–VENT MUTUAL COUPLING

Mutual Coupling Magnitude

The acoustical analogous circuit of a lossless vented-box system, modified to include mutual coupling [2], [6], is presented in Fig. 22. The mutual coupling components are inside the dashed lines. (The mutual coupling resistance [2] is equal to the radiation load resistance and is therefore neglected [4], [12].)

The acoustic mutual coupling mass M_{AM} has a maximum magnitude when the diaphragm–vent spacing is a minimum. A practical minimum spacing between the centers of diaphragm and vent is about $1.5a$, where a is the diaphragm radius. Using this value, and assuming radiation conditions of a 2π-steradian free field, the maximum value of M_{AM} is about $0.13/a$ [2]. This value is reduced for a 4π-steradian free-field load [6].

For a 12-inch driver with an effective diaphragm radius of 0.12 m, the mechanical equivalent M_{MM} of the acoustic mass M_{AM} has a maximum value of 2.2g. The mechanical diaphragm mass M_{MD} for 12-inch drivers varies from about 20g for older types used in large enclosures to more than 100g for newer types designed for use in compact enclosures. Thus the mutual coupling mass may have a magnitude of from 2 to 8% of the total moving mass of the driver when all of the diaphragm air-load mass is accounted for [3, pp. 216-217].

The effect of these values of mutual coupling mass was investigated using the analog circuit simulator. A "lossless" system aligned for a B4 response was compared to the same circuit with the driver and vent masses reduced by the amount of the mutual coupling

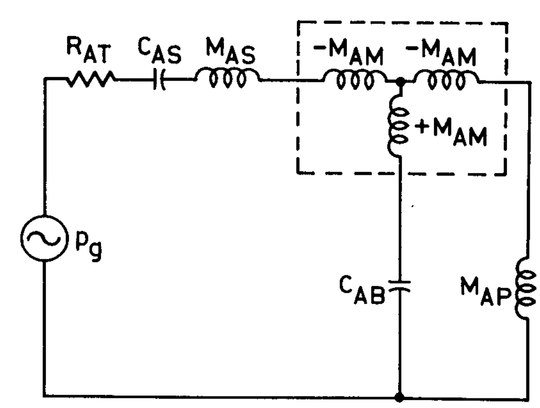

Fig. 22. Acoustical analogous circuit of lossless vented-box loudspeaker system modified to include effects of diaphragm–vent mutual coupling.

mass and the same amount of mass then introduced into the enclosure branch in agreement with Fig. 22.

Effect on Response

The effect of 2% mutual coupling mass on the frequency response could not be observed. The effect of 4% mutual coupling mass could be observed but was hardly worth taking into account. With 8% mutual coupling mass, the cutoff frequency was lowered by about 5% and the corner of the response curve became sharper as described by Locanthi. Similar effects were observed for other alignments.

It would appear that in most cases the effect of mutual coupling on system response is negligible. Only when a driver with a light diaphragm is mounted very close to the vent is the effect on response significant. It then amounts to a slight alignment shift with a very small decrease in cutoff frequency.

Effect on Measurement

Mutual coupling alters the location of the frequencies f_L and f_H of Fig. 20 but does not affect the location of f_M [2]. The shift in f_L and f_H toward each other upsets the calculation of the compliance ratio from Eq. (45), giving a value lower than the true value.

This suggests that if it is desired to measure the true compliance ratio of a system for which the magnitude of mutual coupling is very high, the vent should be blocked and the compliance ratio measured by the closed-box method described in [22]. However, if the parameters of a system are being measured only to evaluate the response of the system, the presence of mutual coupling may be ignored. Experiments on the analog circuit simulator show that the response of a system having the false calculated value of α and no mutual coupling is essentially identical to that of the actual system with its mutual coupling.

10. DISCUSSION

Features of Vented-Box Loudspeaker Systems

The vented-box loudspeaker system acts as a fourth-order high-pass filter. This basic fact determines the available range of amplitude, phase, and transient response characteristics. By suitable choice of parameters, the response may be varied from that of an extreme C4 alignment with passband ripple and very abrupt cutoff to that of an extreme QB3 alignment for which the response is effectively third order. The cost of the gentler cutoff slope and improved transient response of the QB3 alignment is a reduced value of the system efficiency factor $k_{\eta(G)}$, although this reduction is relatively small for real systems with typical enclosure losses. A further sacrifice in the value of this efficiency factor permits the use of SC4 alignments for which the transient response may approach that of a second-order system.

Perhaps the most important feature of the vented-box loudspeaker system is the very modest diaphragm excursion required at frequencies near the enclosure resonance frequency f_B. This feature is responsible for the relatively high displacement-limited power capacity of the system; it also helps to maintain low values of nonlinear distortion and modulation distortion [21].

The "misalignment" curves of Figs. 7 and 8 indicate the necessity for careful alignment of the vented-box system. The plurality of variables makes it very difficult to obtain optimum adjustment by trial-and-error methods, although simulators or computers may be used to speed up the process.

Comparison of Vented-Box and Closed-Box Systems

Most direct-radiator loudspeaker systems use or are based on either the closed-box or vented-box principle. It is therefore of interest to compare these two fundamental systems, and to observe the advantages and disadvantages of each.

One obvious difference is that the vented-box system is more complex, i.e., has more variables requiring adjustment, than the closed-box system. This difference means that satisfactory designs are relatively easier to obtain with the closed-box system and probably accounts for much of the popularity of this system.

The performance relationships derived in this paper for the vented-box system and in [22] for the closed-box system make possible a number of interesting quantitative comparisons which follow.

Response

The response of the vented-box system can typically be adjusted from fourth-order Chebyshev to quasi-third-order maximally flat; that of the closed-box system can be adjusted from second-order Chebyshev to an over-damped second-order condition approaching first-order behavior. This means the closed-box system is nominally capable of better transient response, but Thiele [10, Sec. 13] suggests the differences among correctly adjusted systems of both types are likely to be inaudible.

Efficiency

A comparison of Fig. 16 or Eq. (35) with [22, Fig. 7 or eq. (28)] reveals that the vented-box system has a maximum theoretical value of k_η which is 2.9 dB greater than that of the closed-box system. Both systems suffer to a similar degree from the combined effects of driver and enclosure losses, and both must sacrifice efficiency to make use of alignments which have better transient response than the maximum-efficiency alignment (see Fig. 15 and [22, Fig. 8]).

Typical values of k_η for practical designs still favor the vented-box system by about 3 dB. The larger efficiency constant may be used to obtain higher efficiency for the same size and cutoff frequency, a smaller enclosure size for the same efficiency and cutoff frequency, a lower cutoff frequency for the same size and efficiency, or any proportional combination of these [22, Sec. 4].

Power Capacity

The reduced diaphragm excursion of the vented-box system near the enclosure resonance frequency gives the vented-box system a higher power rating constant k_P than a comparable closed-box system. Comparing Eq. (41) with [22, eq. (35)], the advantage in favor of the vented-box system for average program material is a factor of 3.5, or 5½ dB; for particular applications it may be larger.

However, except for the extreme C4 alignments, **this** advantage is limited to the passband; at frequencies well below cutoff, the vented-box system has a higher relative displacement sensitivity and is therefore more vulnerable to turntable rumble and other subsonic signals.

Driver Requirements

For a given specification of enclosure size and system cutoff frequency, the driver of a vented-box system requires a lighter diaphragm and greater electromagnetic coupling in the magnet-voice-coil assembly compared to the same size driver used in a closed-box system (cf. example of Section 12, Part III, with that of [22, Sec. 10]). These differences are physically consistent with the higher efficiency of the vented-box system. However, for equivalent acoustic power rating, the peak displacement volume V_D and therefore the peak diaphragm displacement x_{max} is substantially smaller for the vented-box driver. Because x_{max} determines required voice-coil overhang, total amount of magnetic material required for the vented-box driver is not necessarily greater.

The closed-box system driver must have high compliance relative to the enclosure if maximum efficiency is to be achieved. While high driver compliance may be beneficial to the vented-box design in terms of transient response, it is not necessary. In fact, a maximum efficiency constant is obtained for the vented-box system with a relatively low value of compliance ratio, and maximum displacement-limited power capacity is obtained with very low values.

Enclosure Size

It is stated above that the larger value of k_η for the vented-box system may be used to obtain a size advantage, i.e., the enclosure may be smaller than that of a closed-box system having the same efficiency and cutoff frequency. Then, despite the smaller enclosure size, if the drivers have equal peak displacement volume, the larger value of k_P for the vented-box system must give a higher acoustic power rating.

This is theoretically correct, but it is practically possible only so long as V_B remains very much larger than the maximum volume displacement required. The maximum air-volume displacement from the enclosure of a vented-box system is larger than V_D because of the contribution of the vent; if this *total* volume displacement exceeds a small percentage of V_B, the compression of air within the enclosure becomes nonlinear to such a degree that the system must produce distortion regardless of the driver linearity [3, p. 274].

In most practical loudspeaker system designs, V_D is indeed very much smaller than V_B, and power capacity is not limited by enclosure size. However, if extreme miniaturization is attempted or if a driver is specifically designed to obtain a very large value of V_D, this limitation may become relevant.

It is important to realize that two direct-radiator loudspeaker systems operated at the same frequency and *acoustic* power level have the same total output volume velocity and displacement regardless of the type of system [12, eq. (2)]. Thus for both closed-box and vented-box systems, adequate enclosure volume is essential to the production of high acoustic output power with low distortion at low frequencies. Some size reduction is possible for closed-box systems if motional feedback is used to control distortion [24], but this technique can be difficult to apply successfully [25].

Typical System Performance

A sampling of commercial vented-box loudspeaker systems was tested in late 1969 by measuring the system parameters as described in Section 7 and programming these into the analog simulator to obtain the system response. For a few systems, the response obtained in this way was checked by indirect measurement [26].

Most of the samples tested fitted into the same two categories previously described for closed-box systems [22, Sec. 8]: systems with a volume of 40 dm^3 (1.5 ft^3) or more, a cutoff frequency of 50 Hz or lower, and relatively flat response; and smaller systems with a cutoff frequency above 50 Hz and several decibels of peaking in the response above cutoff. There was, however, a greater tendency for these two categories to overlap.

While most of the systems were probably designed by traditional trial-and-error methods, the general objectives of system manufacturers appear remarkably consistent. The larger systems fulfill the traditional requirements for high-fidelity reproduction, while the smaller systems suit the apparent requirements of the mass marketplace.

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