

Performance of Enclosures for Low-Resonance High-Compliance Loudspeakers*

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Summary—This paper analyzes the low-frequency behavior of the vented loudspeaker enclosure and the pressure-tight closed box. Inherent interrelationships of the speaker-amplifier system Q , efficiency and response balance are discussed. A method is described for design of small enclosures with very low-resonance high-compliance loudspeakers.

The author concludes that the vented enclosure can have greater acoustic output for a given amount of distortion, lower total harmonic, intermodulation, and transient distortion than a completely closed box of similar size.

LIST OF SYMBOLS

A_d = effective speaker diaphragm area—meters².
 A_p = area of vent—meters².
 B = air gap flux density—webers/meter².
 C_{MS} = total mechanical compliance of the suspension—meters/Newton.
 C_{MR} = total mechanical compliance of the enclosure volume—meters/Newton.
 E_g = amplifier open circuit voltage—volts.
 f, f_L, f_H, f_r = input frequency, lower critical frequency, upper critical frequency, enclosure resonant frequency—cps.
 $F = BLE_g / (R_g + R_e)$ = driving force.
 $g = \omega / \omega_s$ = forced frequency ratio.
 $h = \omega_p / \omega_s$ = tuned frequency ratio.
 l_d = duct length—meters.
 L = voice coil conductor length—meters.
 M_M = total speaker moving system mass—kilograms.
 M_{MD} = diaphragm and voice coil mass—kilograms.
 M_{MR} = total air load mass—kilograms.
 M_P = air mass of vent—kilograms.
 ω_L = lower critical frequency—radians/second.
 ω_M = upper critical frequency—radians/second.
 ω_0 = system resonant frequency—radians/second.
 $\omega, \omega_s, \omega_p$ = input frequency, speaker resonant frequency, enclosure resonant frequency—radians/second.
 P, P_s, P_p = complex rms sound pressure, complex rms sound pressure from speaker diaphragm, complex rms sound pressure from vent—microbar.
 $Q_s = \omega_s M_M / R_M$ = speaker mesh Q .
 $Q_p = \omega_p M_P / R_P$ = port mesh Q .
 R_M = total mechanical resistance in speaker mesh—MKS mechanical ohms.

R_{MP} = total vent resistance—MKS mechanical ohms.

R_{MS} = mechanical resistance of suspension—MKS mechanical ohms.

R_P = total mechanical resistance in vent mesh—MKS mechanical ohms.

ρ = density of air—kilogram/meter³.

R_g = amplifier internal resistance—ohms.

r = average distance of observation point from diaphragm and/or port—meters.

R_e = voice coil resistance—ohms.

S_s = speaker suspension stiffness—Newtons/meter.

S_B = stiffness of enclosure volume—Newtons/meter.

t = vent wall thickness—meters.

U, U_s, U_p = complex rms volume velocity, complex rms volume velocity of speaker diaphragm, complex rms volume velocity of vent—meter³/second.

v_s = speaker diaphragm velocity—meters/second.

v_p = vent velocity—meters/second.

x, x_s, x_p = amplitude, diaphragm amplitude, vent amplitude—meters.

$x_{ST} = F / S_s$ = static deflection.

V_B = volume of enclosure—meters³.

V_D = volume of duct—meters³.

INTRODUCTION

THE subject of enclosure design for direct-radiator loudspeakers has received much attention in recent years. There is no other subject so controversial, perhaps as a result of a general lack of understanding of the basic principles of operation of enclosures. The current trend is toward smaller enclosures, lower speaker resonances, and better performance claims. Trade journals tell of "all new enclosures," "revolutionary concepts," and "totally new principles of acoustics," when in reality there is a close identity with enclosure systems described long ago in well-known classics on acoustics. Actually there has been no basically new type of enclosure developed in this decade, although much worthwhile effort has been devoted to refinement and improvement of existing basic types.

The objectives of this paper are to select one of the currently popular enclosures, analyze its low-frequency behavior, discuss its limitations, and find a means of improving its performance.

The pressure-tight closed box using a high-compli-

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ance low-frequency speaker was selected for two reasons: 1) the design is based on sound engineering principles, and 2) the ever increasing popularity of this enclosure may alone have justified the expenditure of the time.

THE PRESSURE-TIGHT CLOSED BOX

In order to proceed intelligently, it is necessary to define certain design criteria for low-frequency reproduction. A slight problem arises here because all loud-speaker and enclosure designs must at the final stages of development be based on subjective judgments by people as to what constitutes "good quality." Because of this, it is not possible to be too specific in defining the design criteria. A reasonable amount of research into this problem disclosed that most listeners preferred a flat response to frequencies as low as about 40 cps. They also preferred speaker systems with low harmonic distortion and little or no transient distortion. Low-efficiency speaker systems were generally frowned upon because in most cases this meant forced obsolescence of existing amplifiers.

It will be shown later that it will be necessary to sacrifice over-all efficiency in order to extend the low-frequency response. It now remains to determine the maximum allowable efficiency loss. That a maximum limit on efficiency loss must be established becomes apparent when one considers three factors: 1) the amplifier economics, 2) the necessity for adequate reserve power to handle peaks without overload, and 3) the deterioration of amplifier characteristics with time. This last factor is tied in closely with the second.

Observations have shown that VU meter readings of one watt or so are about as high as reached in normal home listening. It should be possible, therefore, to justify a loss of 4 to 5 db of efficiency before noticing distortion on peaks from a good 10 to 12 watt amplifier. A loss of 10 db would be too excessive for a 10 to 12 watt amplifier as there would then be no reserve to handle peaks.

From the evidence just stated it is possible to set up the design criteria.

- 1) The response must be flat to as low a frequency as possible.
- 2) The total harmonic distortion must be as low as possible.
- 3) There must be no transient hangover.
- 4) The efficiency must be great enough to permit the use of a 10 to 12 watt amplifier.

The first design criterion, the extension of response to as low a frequency as possible, ordinarily would mean that the value of enclosure compliance would be made as large as possible so that the suspension compliance determined the resonant frequency. Because the small box precludes this approach, the speaker compliance is made as large as practical. The enclosure compliance then determines the resonant frequency. Although it is not possible to obtain a resonant frequency much lower

than about 60 cps in this manner, the distortion characteristics are somewhat improved as a result of the good linearity of the enclosure compliance.

The resonant frequency can be lowered an additional 10 to 15 per cent by completely filling the box with loosely packed fiberglass, kapok, or cellufoam.¹ The resonant frequency decreases because the compressions become isothermal. This means that the velocity of sound decreases from about 344 to 291 m/sec. A 1 to 2 db loss in efficiency results, however, because the resistive component of box impedance is increased.

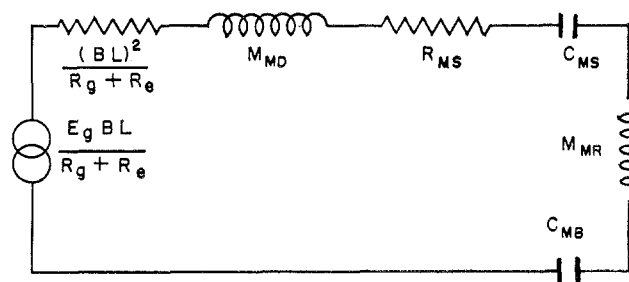


Fig. 1.

The mechanical equivalent circuit, Fig. 1, and equations of motion of the closed box-speaker system are well known. The steady-state solution yields the equation describing amplitude which when multiplied by the angular frequency and effective speaker area becomes the equation for volume velocity. Once volume velocity is known, it is a simple matter to obtain the sound pressure output.

The plot of amplitude vs frequency, Fig. 2, reveals that the amplitude increases many times in the region of resonance and below when Q becomes greater than 0.5. Suspension nonlinearities play an important part in determining the amount of total harmonic distortion at large amplitudes even though the box compliance supplies the major part of the restoring force. The result of a nonlinear suspension is the production of odd order harmonics with the third harmonic being predominant. Because the amplitude of a direct radiator speaker is inversely proportional to frequency squared below the region of ultimate radiation resistance, greatest distortion will occur at the low frequencies.²

Fig. 3 is a plot of the sound pressure response of a closed box-speaker system. Note that a Q of 0.5, corresponding to critical damping and best transient response, does not give the flattest output down to the lowest frequency possible. For flat response, Q should approximately equal unity. This conflicts with the first and third design criteria.

The only factor determining the transient response

¹ L. L. Beranek, "Acoustics," McGraw-Hill Book Co., Inc., New York, N. Y., 1st ed., p. 220; 1950.

² H. F. Olson, "Elements of Acoustical Engineering," D. Van Nostrand Co., Inc., New York, N. Y., 3rd ed., p. 186; 1958.

EXCURSION OF DIRECT RADIATOR AS A FUNCTION OF Q
CLOSED BOX, INFINITE BAFFLE OR FREE AIR OPERATION ASSUMED

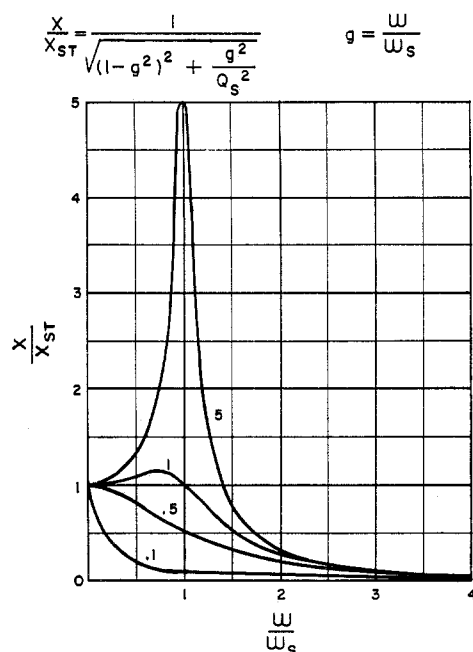


Fig. 2.

SOUND PRESSURE LEVEL RESPONSE OF DIRECT RADIATOR
LOUDSPEAKER, INFINITE BAFFLE OR CLOSED BOX ASSUMED

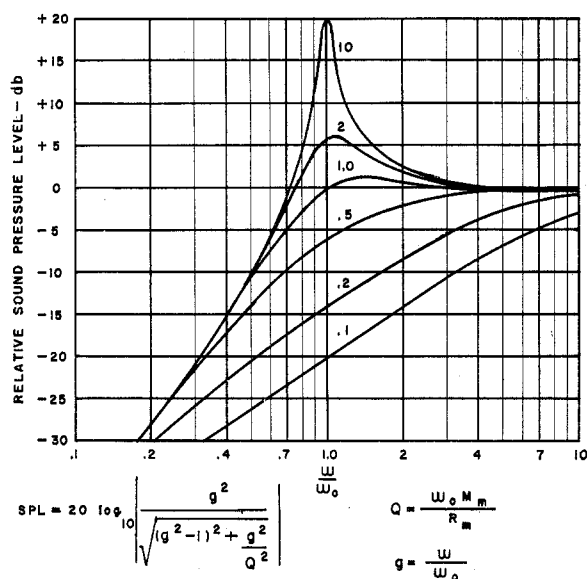


Fig. 3.

and low-frequency performance is the amount of damping. The damping can be changed by proper choice of amplifier damping factor or by adjustment of the flux density in the magnetic circuit. The popular belief that a large value of air stiffness in a small closed box increases speaker damping is erroneous. Damping is a function of resistances in the system.

The factor exerting the greatest influence on Q (damping) and low-frequency output is the product of magnetic field strength and voice coil conductor length

commonly known as the BL product. Decreasing the BL product will actually increase efficiency in the region of resonance but an efficiency loss results at frequencies above resonance. It now becomes apparent that the pressure-tight closed box cannot fulfill adequately all four design criteria. It will be necessary to compromise transient response and over-all efficiency because increases in Q are usually achieved by an increase in moving system mass, a decrease in BL product or both.

THE VENTED ENCLOSURE

A review of other known speaker enclosures suggests that the vented enclosure when used with a high-compliance speaker could produce a sound pressure response at least as good as that of the completely closed box and with certain advantages.

A search of the literature pertaining to vented enclosure design reveals that although equivalent circuits and equations for calculating Helmholtz resonance and location of the three critical frequencies were thoroughly developed, apparently no method has been published for calculating the response shape. Beranek's excellent chapter on enclosures³ describes a method for obtaining the relative sound pressure level at the three critical frequencies. But the critical frequencies become widely separated when the box is small; and because the speaker system operates as a simple doublet at the lowest critical frequency, it is no longer possible to describe adequately the response shape because useful output occurs at only two of the three points, the middle and upper critical frequencies.

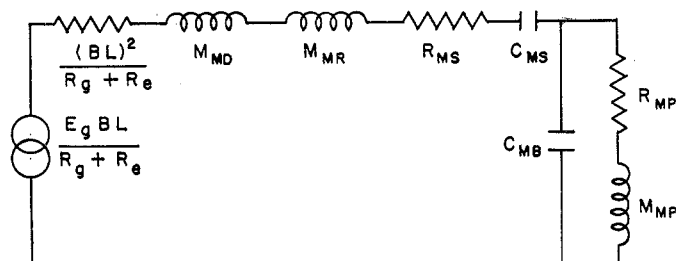


Fig. 4.

The mechanical equivalent circuit of the vented enclosure, Fig. 4, can be solved with the methods used for the closed box case. An exact solution becomes extremely complex because of the mutual coupling between the speaker and vent and the presence of resistance in both meshes. The vent resistance, however, can be ignored because the Q of the vent mesh is usually 20 to 40 times greater than the speaker mesh Q . The effects of mutual coupling can also be ignored since gains are insignificant when the two piston areas are very much different.⁴

³ Beranek, *op. cit.*, pp. 241-258.

⁴ S. J. Klapman, "Interaction impedance of a system of circular pistons," *J. Acoust. Soc. Amer.*, vol. 2, pp. 289-295; January, 1940.

The error resulting from these simplifications becomes significant only for large enclosures and then only in the region of speaker resonance when the vent Q is less than 10.

A vent in the closed box adds a second degree of freedom to the system causing a redistribution or repartition of the resonant frequency and damping of the speaker. The original speaker resonance is replaced by two new resonances, one near the closed box resonance and one substantially below the speaker resonance.

Damping at these two resonances is greater than for the closed box case.⁵

The vent acts as an acoustical mass which resonates with the compliance of the enclosure volume at a particular frequency. The equivalent circuit indicates that if the enclosure is tuned to the speaker resonance, the impedance becomes a maximum and is resistive at this frequency so that speaker amplitude should be greatly reduced. Maximizing the equation for sound pressure output with respect to enclosure resonance reveals that enclosure resonance must indeed be equal to speaker resonance for maximum over-all output to the lowest frequency possible. This condition of tuning is assumed for the remainder of this paper although it may be desirable to tune the enclosure to a higher frequency if increased output is desired in the region of the upper critical frequency. The sound pressure output of the system now becomes a function of two variables, the speaker Q and the ratio of enclosure stiffness to speaker suspension stiffness. This ratio can be thought of as a coefficient of coupling between the speaker and port meshes.

Low-frequency output is increased and the low-frequency cutoff is lowered when the stiffness ratio is decreased. Although output varies directly with speaker Q as in the closed box case, it will be shown that a Q less than 0.5 will give a flat response whereas the closed box requires a Q of approximately unity.

Fig. 5, a plot of the ratio of vent velocity to speaker diaphragm velocity, shows that for normal vent Q 's (greater than 10) the acoustic power radiated from the vent predominates over that radiated from the speaker diaphragm for about one-half of an octave above and below speaker resonance. It is observed, however, that total radiation decreases rapidly below speaker resonance. This occurs because at frequencies below speaker resonance the volume velocity in the vent becomes out of phase with the speaker diaphragm volume velocity. Fig. 6 is a plot of the relative phase angle between vent and speaker diaphragm volume velocity. Although vent and diaphragm radiation are in quadrature at speaker resonance for all values of vent Q , the transition to out-

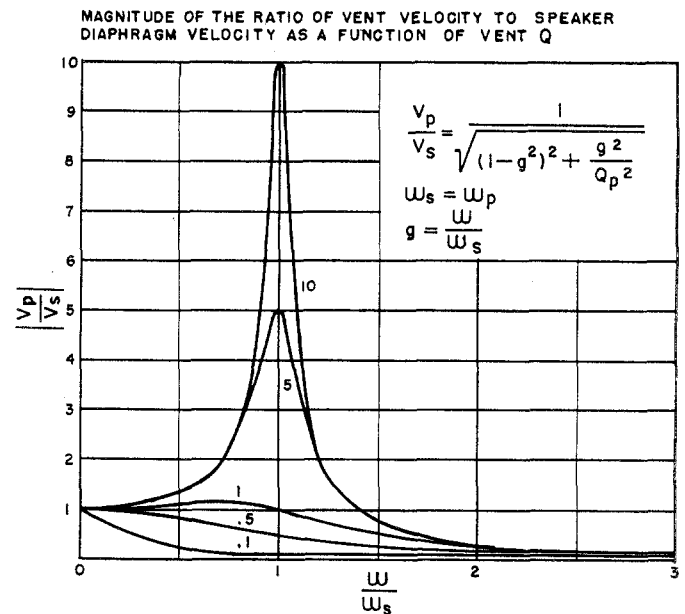


Fig. 5.

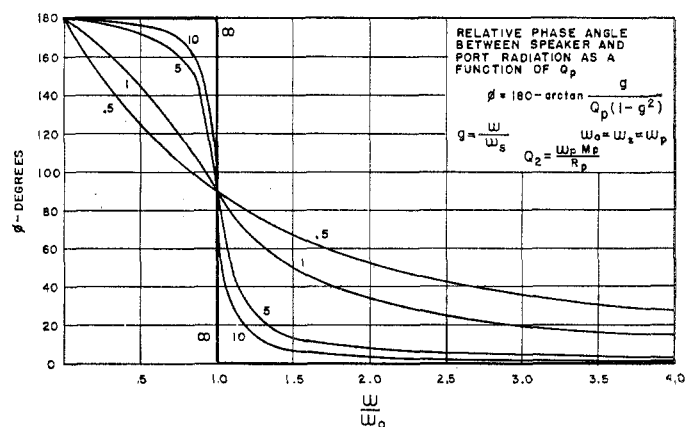


Fig. 6.

of-phase or in-phase operation is very rapid when the Q is greater than 5.

The gradual phase shift occurring when the vent Q is less than 0.5 may lead to the erroneous conclusion that low values of vent Q can lower the low-frequency cutoff. An extension of low-frequency cutoff cannot occur because vent power output diminishes rapidly as Q becomes equal to or less than unity (see Fig. 5). The performance then approaches the closed box performance. This is predicted by the equations of motion which reduce to those of a closed box when either the vent Q or area are allowed to approach zero.

The diaphragm amplitude becomes a minimum at speaker resonance as opposed to the maximum which occurs in the closed box. Fig. 7 shows the amplitude variations of a speaker diaphragm in a vented enclosure for a stiffness ratio of 7 as a function of speaker Q . A

⁵ J. B. Crandall, "Theory of Vibrating Systems and Sound," D. Van Nostrand Co., Inc., New York, N. Y., 2nd ed., pp. 62-63; 1927.

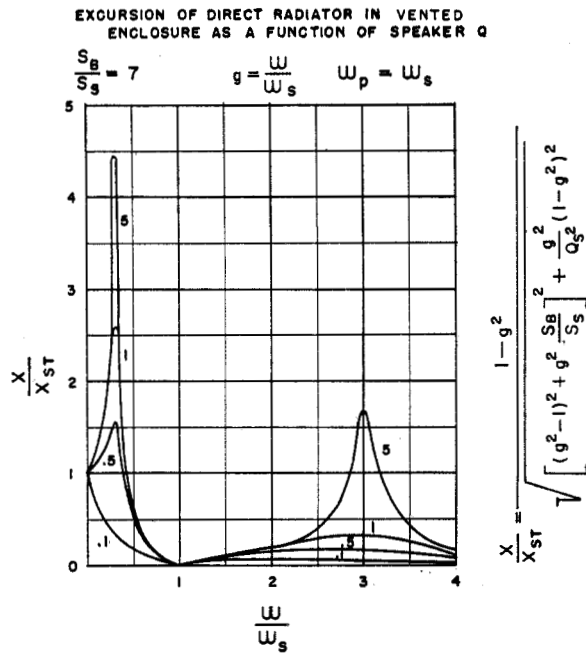


Fig. 7.

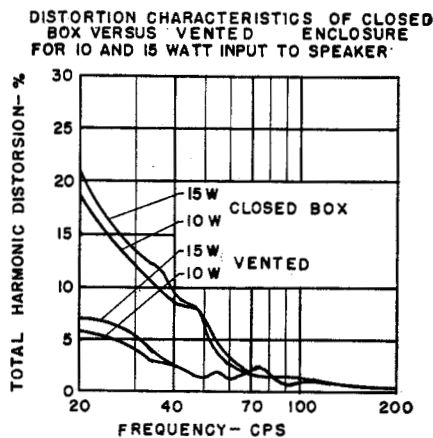


Fig. 8.

stiffness ratio of 7 corresponds to a 12-inch high-compliance speaker in an enclosure volume of about $2\frac{1}{4}$ cubic feet. The diaphragm amplitude remains very uniform down to speaker resonance for critically or overdamped speaker operation. Because the vent mesh consists of linear elements, the harmonic and intermodulation distortion are greatly reduced.⁶

Fig. 8 shows the distortion characteristics of a $2\frac{1}{4}$ -cubic-foot closed vs vented box-speaker system. The speaker used in these measurements was a Jensen P12-NF high-compliance woofer operating with a Q of 0.5.

Fig. 9 illustrates the dependence of flat response on proper enclosure tuning. Fig. 10 is the experimental

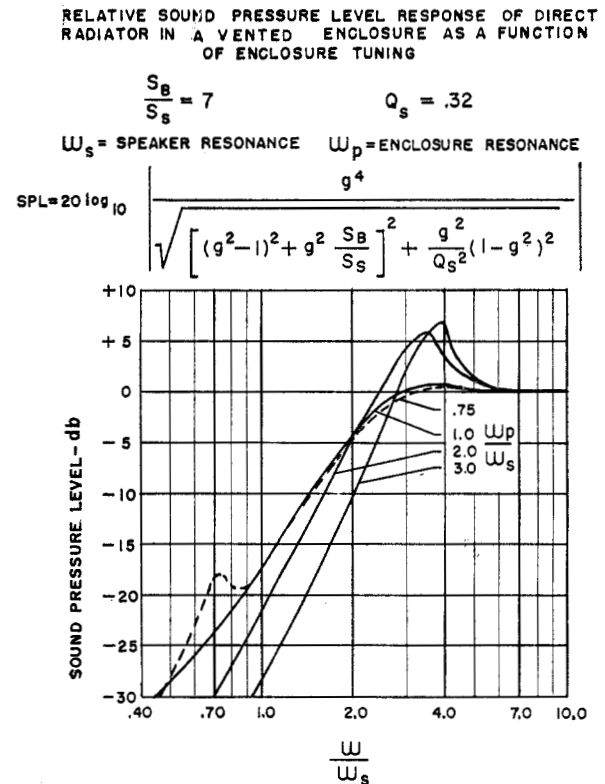


Fig. 9.

EXPERIMENTAL DATA SHOWING RELATIVE SOUND PRESSURE LEVEL RESPONSE OF DIRECT RADIATOR IN A VENTED ENCLOSURE AS A FUNCTION OF ENCLOSURE TUNING

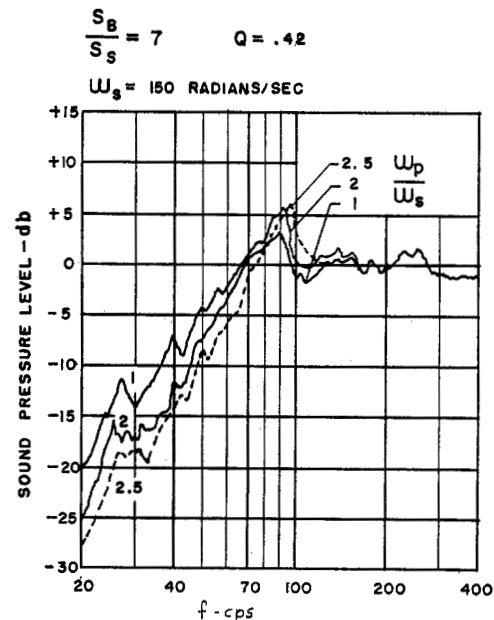


Fig. 10.

data verifying the theoretical data of Fig. 9. Figs. 11-13 show the theoretical sound pressure response of the vented enclosure with stiffness ratios of 1, 3, and 7, respectively, as a function of speaker Q . The experimental

⁶ H. S. Knowles, "Loudspeakers and room acoustics," in "Henry Radio Engineers Handbook," McGraw-Hill Book Co., Inc., New York, N. Y., pp. 760-761; 1950.

RELATIVE SOUND PRESSURE LEVEL RESPONSE OF DIRECT RADIATOR IN VENTED ENCLOSURE AS A FUNCTION OF SPEAKER Q

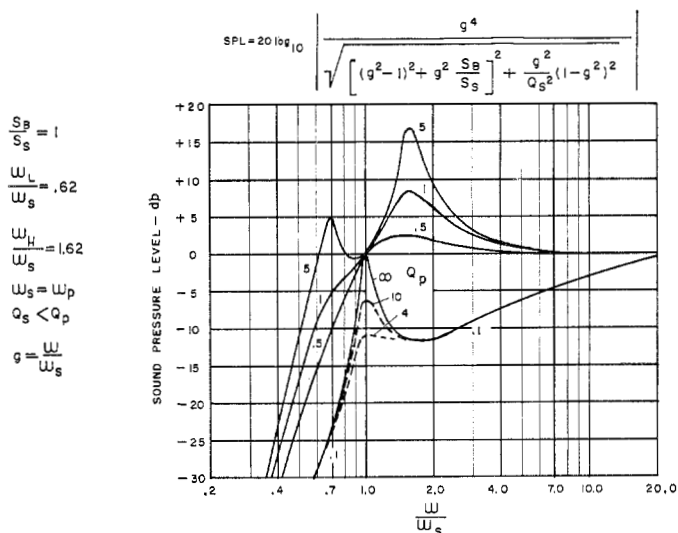


Fig. 11.

RELATIVE SOUND PRESSURE LEVEL RESPONSE OF DIRECT RADIATOR IN A VENTED ENCLOSURE AS A FUNCTION OF SPEAKER Q

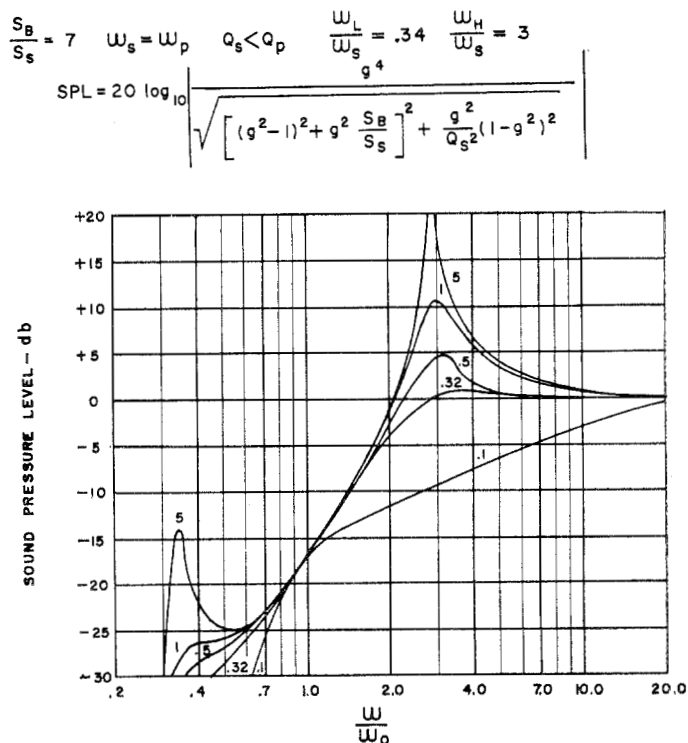


Fig. 13.

RELATIVE SOUND PRESSURE LEVEL RESPONSE OF DIRECT RADIATOR IN VENTED ENCLOSURE AS A FUNCTION OF SPEAKER Q

$\frac{S_B}{S_s} = 3$ $\frac{\omega_L}{\omega_s} = .458$ $\frac{\omega_H}{\omega_s} = 2.19$ $\omega_s = \omega_p$ $Q_s < Q_p$ $g = \frac{\omega}{\omega_s}$

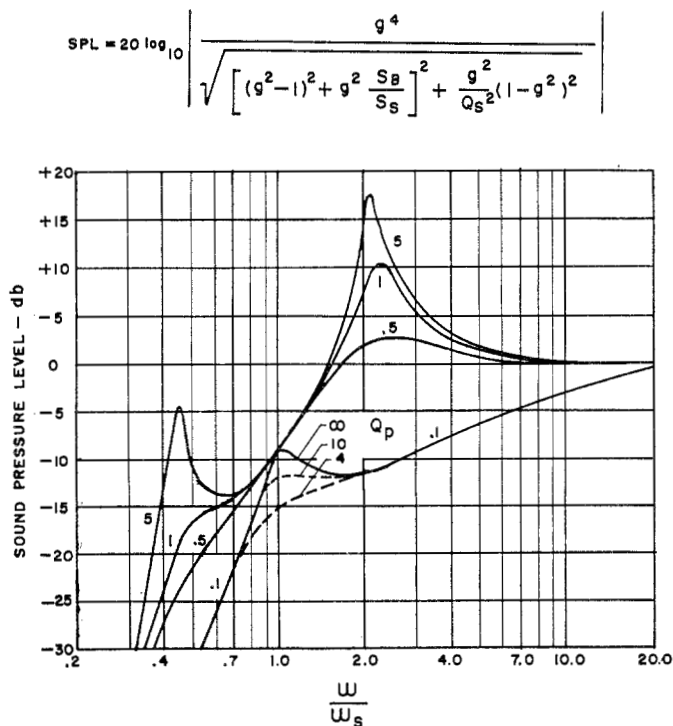


Fig. 12.

EXPERIMENTAL DATA SHOWING RELATIVE SOUND PRESSURE LEVEL RESPONSE OF DIRECT RADIATOR IN A VENTED ENCLOSURE AS A FUNCTION OF SPEAKER Q

$\frac{S_B}{S_s} = 7$ $\omega_s = \omega_p = 151 \text{ RAD./SEC.}$

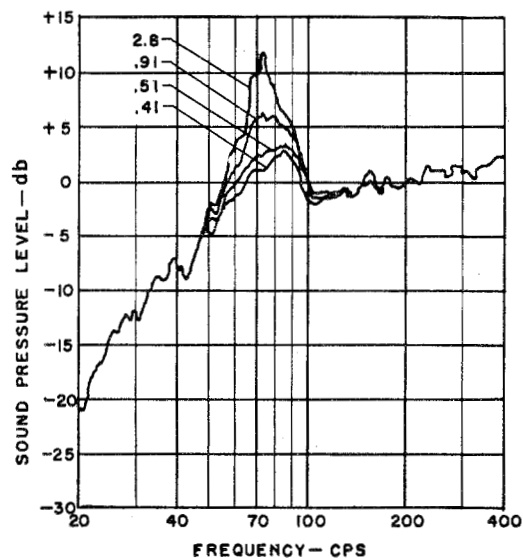


Fig. 14.

data in Fig. 14 verify the theoretical data of Fig. 13 and substantiate the validity of the theoretical equations. These response curves certainly demonstrate that a large box still outperforms a small box. An important result of this data is that flat response is in all cases obtained with a speaker Q less than 0.5. The data also show that the acoustic output from small enclosures is only slightly improved by venting. Consider the case of a stiffness ratio of 7 and assume a speaker resonant frequency of 20 cps.

Fig. 13 shows that flat response is maintained to 54 cps for a speaker Q of 0.32 with the 10-db down point occurring at 28 cps. The same stiffness ratio applied to a closed box would increase speaker resonance to 60 cps. Fig. 3 shows that a speaker Q of 1 will give flat response to 60 cps with the 10-db down point occurring at 32 cps. The increase in output from 28 to 60 cps in the vented enclosure is barely 1 db. The transient response, harmonic and intermodulation distortion are, however, greatly improved. In cases where the enclosure volume can be more generous, venting offers considerable gains in output.

CONCLUSIONS

Many considerations lead to the conclusion that a vented enclosure can do everything the closed box can do, and with certain advantages. But honest comparison of the two must include all factors, not just those favoring one system.

Important points against venting which could result in poor performance if proper consideration is not given them are:

1) Drop in output below the resonant frequency is 18 db per octave compared to 12 db per octave for the closed box. It is commonly thought that the transient effects resulting from a cutoff as sharp as 18 db per octave can be bothersome.

2) Very small enclosure volumes coupled with a very low speaker resonant frequency may require a vent so small as to be ineffective because of excessive viscous losses in the vent. The operation is then about the same as that of the closed box. In extreme cases of this sort, rectification has been observed in the air flow through the vent. This caused a displacement in the dynamic center of diaphragm amplitude. The resulting distortion was greater than for the closed box.

3) Mistuning or incorrect speaker Q can exaggerate the output in the vicinity of the upper critical frequency. This can cause an aggravated booming sound generally too high in frequency to give even a good impression of false bass.

The optimum system goal as defined by the four design criteria must overcome these limitations. This can be accomplished in the following manner:

The speaker resonant frequency must be low enough that the 18 db per octave slope occurs below 20 cps.

Subjective listening tests did not indicate the presence of any troublesome transients. An acceptable resonant frequency would be between 20 and 30 cps.

The vent must have small enough resistance so that the volume velocity is not impeded seriously. Volume velocity will be independent of vent area so long as viscous losses are minimized. If the vent area becomes too small, a larger area with a duct must be used.

The amplifier must have a damping factor sufficient to maintain at least critical damping in the speaker mesh. Speaker Q 's ranging from 0.3 to 0.4 give best results.

Having disposed of the limitations by proper design, the vented enclosure has definite and important advantages over the closed box. All exist in the octave or octave and one half in the region of speaker resonance. They stem from the one characteristic differentiating the vented enclosure from the closed box—better diaphragm loading in the region where the box is an active element.

The vent relieves the diaphragm of much of the necessity to move. The reduced diaphragm amplitude is not at the expense of sound output because the vent also assumes the task of radiation of sound energy. The advantages of the vented enclosure become: 1) lower harmonic distortion because the linear vent operation and reduced diaphragm amplitude minimize the effects of speaker nonlinearities, 2) reduced intermodulation distortion because of reduced diaphragm amplitude, 3) improved transient response because the speaker is at least critically damped, 4) greater acoustic output for a given amount of distortion, and 5) less of the deliberate efficiency loss for purposes of response leveling is required because the speaker must be operated at a lower Q .

The Q can be decreased by increasing the speaker efficiency. It has been found that the efficiency averages at least 3 db better, for equivalent response trend, than would have been allowable with a closed box system.

The author believes that in spite of its simple appearance the design of a vented enclosure is sufficiently difficult that it should not be attempted by the layman unless he is exceptionally well informed and has adequate test facilities.

APPENDIX

A piston whose diameter is less than one third wavelength is essentially nondirectional at low frequencies. It can, therefore, be approximated by a hemisphere whose rms volume velocity is equal to the product of voice coil velocity and effective speaker cone area,

$$U_e = v_c A_d.$$

The magnitude of rms sound pressure at a distance r (in the far field) from the speaker is⁷

⁷ Beranek, *op. cit.*, p. 188.

$$|P| = \frac{|U_e| f \rho}{r} \quad (1)$$

The equations describing sound pressure response of the closed and vented enclosure can be obtained from the equations of motion for the two systems by substituting for volume velocity in (1).

CLOSED BOX

The equation of motion obtained from Fig. 1 is

$$(-M_M \omega^2 + S_M + jR_M \omega)x = F. \quad (2)$$

Eq. (2) is solved for x . The equation describing amplitude becomes

$$x = \frac{x_{ST}}{\sqrt{(1 - g^2)^2 + \frac{g^2}{Q_s^2}}} \quad (3)$$

The expression for voice coil velocity is obtained by multiplying (3) by the angular frequency.

$$v_s = \omega x. \quad (4)$$

Volume velocity is obtained by multiplying (4) by the effective speaker cone area.

$$U_s = A_d \omega x. \quad (5)$$

The sound pressure level is obtained by substituting (5) into (1).

$$P = \frac{A_d \omega x_{ST} f \rho}{r \sqrt{(1 - g^2)^2 + \frac{g^2}{Q_s^2}}} \quad (6)$$

A reference volume velocity is defined by

$$U_{\text{ref}} = \frac{x_{ST} \omega A_d}{g^2} \quad (7)$$

This is the actual volume velocity above resonance under the special condition that the expression under the radical in (3) is proportional to g^4 ; *i.e.*, the diaphragm is completely mass-controlled,

$$\text{SPL} = \left| \frac{P_s - P_p}{P_{\text{ref}}} \right| = 20 \log \left| \frac{g^4}{\sqrt{\left[(g^2 - 1)(g^2 - h^2) - g^2 \frac{S_B}{S_s} \right]^2 + g^2 \left[\frac{h^2 - g^2}{Q_s} \right]^2}} \right| \quad (15)$$

$$\left((g^2 - 1)^2 \gg \frac{g^2}{Q_s^2} \right).$$

A reference sound pressure is defined for low frequencies by substituting (7) into (1).

$$|P_{\text{ref}}| = \frac{x_{ST} \omega A_d f \rho}{g^2 r} \quad (8)$$

An expression for relative sound pressure level (SPL) in decibels is obtained by taking the ratio of (6) and (8).

$$\text{SPL} = 20 \log \left| \frac{g^2}{\sqrt{(g^2 - 1)^2 + \frac{g^2}{Q_s^2}}} \right| \quad (9)$$

VENTED ENCLOSURE

The equations of motion obtained from Fig. 4 are

$$\left. \begin{aligned} (-M_s \omega^2 + S_s + S_B + jR_s \omega)x_s - S_B x_p &= F \\ -S_B x_s + (-M_p \omega^2 + S_B + jR_p \omega)x_p &= 0 \end{aligned} \right\} \quad (10)$$

Eq. (10) is solved for x_s and x_p .

AMPLITUDE

In order to simplify the expressions, Q_p is assumed to equal infinity. The resulting error is very small because the typical vent mesh Q is 20 to 40 times greater than the speaker mesh Q . The speaker and vent amplitudes become

$$x_s = \frac{x_{ST}(h^2 - g^2)}{\sqrt{\left[(g^2 - 1)(g^2 - h^2) - g^2 \frac{S_B}{S_s} \right]^2 + g^2 \left[\frac{h^2 - g^2}{Q_s} \right]^2}} \quad (11)$$

$$x_p = \frac{x_{ST} h^2}{\sqrt{\left[(g^2 - 1)(g^2 - h^2) - g^2 \frac{S_B}{S_s} \right]^2 + g^2 \left[\frac{h^2 - g^2}{Q_s} \right]^2}} \quad (12)$$

The volume velocities are obtained exactly as in the closed box case:

$$U_s = A_d \omega x_s \quad (13)$$

$$-U_p = -A_d \omega x_p. \quad (14)$$

The total sound pressure is obtained by substituting (13) and (14) into (1) and adding the two pressures. A negative sign is used for U_p because, except for the phase shift introduced in the vent mesh, the radiation from the back of the speaker cone is 180° out of phase with the front radiation. Using the concept of reference sound pressure, the relative sound pressure level in decibels is

This allows the calculation of response for any condition of tuning.

Maximizing (15) with respect to enclosure tuning shows that maximum output at speaker resonance is obtained when the enclosure is tuned to speaker resonance, *i.e.*, $h = 1$. The expression for relative sound pressure level becomes

$$\text{SPL} = 20 \log \left| \frac{g^4}{\sqrt{\left[(g^2 - 1)^2 - g^2 \frac{S_B}{S_S} \right]^2 + \frac{g^2}{Q_S^2} (1 - g^2)^2}} \right|. \quad (16)$$

DESIGNING THE ENCLOSURE

The enclosure volume should be based on the maximum amount of space available. The object is to get the lowest possible value of S_B/S_S . It is necessary to know only two factors, S_B/S_S and Q_S , in order to calculate the response.

DETERMINATION OF S_B/S_S

First measure free air speaker resonance, denoting this by f_1 . Then install the speaker in the unvented enclosure and measure resonance again. Denote this by f_2 . The stiffness ratio is then given by

$$\frac{S_B}{S_S} = \left[\frac{f_2}{f_1} \right]^2 - 1.$$

DETERMINATION OF SPEAKER Q

Q_S is determined from the velocity curve as described in Fig. 15. The width, Δf cps, is measured between the points of the curve on either side of the resonance peak where the voltage is 3 db down (0.707) from the maximum voltage.

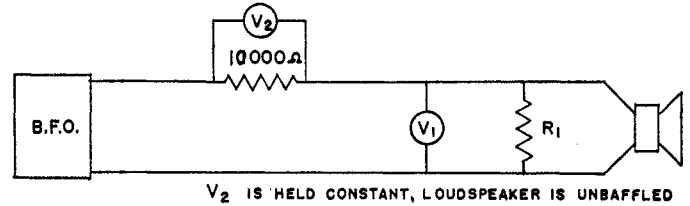
The area of the vent is obtained from

$$f_r = 2155 \sqrt{\frac{A_p}{V_B(t + 0.96\sqrt{A_p})}}. \quad (17)$$

The enclosure should be tuned to the resonant frequency of the speaker. It is permissible to tune to a higher frequency if increased output is desired in the region of the upper critical frequency. In some cases, the output in the region of the upper critical frequency may tend to peak up. Peaking can be minimized by increasing the damping on the speaker by increasing amplifier damping factor. A negative damping factor may have to be used in some instances. If the damping factor cannot be changed, the enclosure should be tuned lower than speaker resonance. While this will reduce the peak, it will also result in some losses in output lower in frequency.

Since the vent behaves as a simple source at low frequencies, the amount of power radiated is independent of the vent area for any given volume velocity. It is permissible, therefore, to use any value of vent area so long as the desired enclosure resonance is obtained.

The use of resistive loading over the vent or the use of a series of small holes distributed over a large area (another way of adding resistive loading) is generally not recommended. An important reason for using a vented enclosure is that the loudspeaker produces far less distortion in the octave above speaker resonance than would be the case if the box were closed. Adding resist-



(1) RESONANT FREQUENCY

VARY OSCILLATOR FREQUENCY UNTIL V_1 IS MAXIMUM.

(2) LOADED Q

(A) OPEN CIRCUIT R_1 . OBTAIN SHAPE OF VELOCITY CURVE BY PLOTTING AGAINST FREQUENCY THE QUANTITY

$$e = V_1 - \frac{R_{VC}V_2}{10000} \quad Q_1 = \frac{f}{\Delta f}$$

(B) INSERT R_1 (ABOUT 80 OHMS). PLOT NEW VELOCITY CURVE

$$Q_2 = \frac{f}{\Delta f}$$

(C) DETERMINE MECHANICAL RESISTANCE, R_M , FROM

$$R_M = (R_1 + R_{VC}) \left(\frac{Q_1}{Q_2} - 1 \right)$$

(D) LOADED Q IS GIVEN BY

$$Q_L = \frac{R_g + R_{VC}}{R_g + R_{VC} + R_M} Q_1$$

$$Q_L = \frac{R_{VC}}{R_{VC} + R_M} Q_1 \quad \text{IF } R_g \ll R_{VC}$$

R_{VC} = VOICE COIL RESISTANCE - OHMS

R_g = GENERATOR INTERNAL RESISTANCE - OHMS

GENERATOR INTERNAL RESISTANCE CAN BE OBTAINED FROM

$$\text{DB REGULATION} = 20 \log_{10} \frac{R_g + R_L}{R_L}$$

R_L = LOAD RESISTANCE - OHMS

Fig. 15.

ance to the vent will reduce the power radiated and will increase the speaker diaphragm amplitude and distortion. The vent area should not be allowed to be less than about 4 square inches. If (17) indicates an area less than this value, the area should be increased arbitrarily and a duct (installed behind the vent) is used to tune the enclosure properly. The expression for resonance now becomes

$$f_r = 2155 \sqrt{\frac{A_p}{(V_B - V_D)(l_d + 0.96\sqrt{A_p})}}. \quad (18)$$

The enclosure should be lined with a 2-inch thickness of fiberglass on at least three sides to eliminate any normal modes. The approximate location of the upper and lower critical frequencies is obtained from

$$f_{L,H} = (0.707)f_s \sqrt{2 + \frac{S_B}{S_S} \pm \sqrt{4 \frac{S_B}{S_S} + \left[\frac{S_B}{S_S} \right]^2}}. \quad (19)$$

The response shape can be calculated from either (15) or (16), depending on whether or not the enclosure is tuned to speaker resonance. The calculations can become rather tedious but unfortunately there is no short cut.