DRIVER DAMPING AND VARIATIONS OF THE CLOSED-BOX SYSTEM

6.1. Introduktion to Derived Systems

In the three preceding chapters, only the minimum, unavoidable losses were considered to be present in the loudspeaker systems analysed. For these systems, it is clear that the value of QT for the driver branch of the system must be carefully controlled to obtain smooth response. The penalty for allowing QT to be too high is invariably a resonant peak in the system frequency response.

Historically, the design and manufacture of loudspeaker drivers has often been divorced from the application of these drivers in loudspeaker systems. Economic competition has led to the manufacture of drivers which have small, inexpensive magnets and therefore possess high values of QES (and thus QTS) and often low values of x-max (and thus VD) as well. These drivers are consequently unsuitable for critical applications, even where acoustic power capacity is adequate, because they produce unacceptable response peaks. For economic reasons, system designers have sought ways to reduce or eliminate the response peaks produced by these drivers.

From (1-59), the high value of QES for these drivers must lead to low efficiency, even if the system response peak is somehow removed. Thus for a given acoustic requirement, the cost saved on magnetic material must be weighed against the added cost of a more powerful amplifier and a voice coil capable of dissipating greater input power. In some cases, there is in fact a definite economic advantage in using the small-magnet driver.

Attempts to reduce or eliminate the inherent response peak of a system using a high-Q driver usually involve the application of acoustic damping to some part of the enclosure. There are many ways of doing this [B3], Some more successful than others. A selection are analyzed in this and the following chapter. The various approaches are divided into systems which are derived by modification from the closed-box system and those derived from the vented-box system. The analyses are not as complete as those for the basic systems but are sufficient to show the principal advantages or disadvantages of each approach.

All of the systems which are able to employ a high-Q driver successfully suffer from the inherent low efficiency of the driver. In addition, most of the systems using acoustic damping in the enclosure, whatever the driver requirements, have less bandwidth and/or acoustic power capacity than comparable basic systems.

6.2. Acoustic Damping of the Driver

The simplest way to correct a high value of QTS is to apply acoustic damping directly to the driver using a shroud of acoustically resistive material as described in [G1] or [N3]. This effectively adds resistance in series with RAS and leads to a value of QM (Section 1.5.4) lower than QMS. By suitable choice of the kind and amount of damping material, almost any desired value of QT dan be obtained regardless of the value of QES or QE. The parameter measurement methods of Section 9.2 may be used to determine experimentally the amount of material needed to meet a particular requirement. Care must be taken that the material remains stationary.

Because this technique affects only the driver, it may be used in conjunction with any of the systems described in the previous three chapters. Provided that the added resistance is reasonably constant with frequency, the system acoustical performance will then be identical to that with a driver having optimum magnetic damping, except for efficiency. The principal difference is a lower value of kη (Q) as defined by (3-11) or (4-8); this leads to a lower value of kη and thus to a higher displacement-limited input power rating as shown by (3-24) or (4-17). The driver voice coil must be able to dissipate this higher input power if the system acoustic power capacity is to be fully realized.

6.3. The Densely-Filled Closed-Box System

The filling materials often used in the enclosures of closed-box systems (Section 3.8) contribute to the enclosure absorption losses. It is often assumed that this feature of the material may be used to greatly increase system damping and thus to make up for inadequate driver damping.

The magnitude of additional damping that can be obtained in this way is in fact very limited. The resistive losses in the material might amount to as much as 25 % of the total damping required in the system, but this is not enough to compensate for a badly underdamped driver. Further addition of material seldom improves the situation because the material begins to occupy too much volume and thus to decrease enclosure compliance, and it also tends to increase the effective moving mass of the system [A7]. Both effects cause the total system Q to increase in opposition to the resistive contribution, and the mass increase further reduces efficiency.

Gross underdamping in closed-box systems is more effectively dealt with by locating some very resistive material directly over the driver, i.e. using driver acoustic damping as described above, and adjusting the amount of filling material in the enclosure to optimise the system compliance in the normal manner.

6.4. The Resistance-Coupled Double-Cavity Closed-Box System

A variation of the closed-box enclosure which has been patented on various occasions and claimed to offer improvements is a closed box divided into two internal chambers or cavities which are separated by an acoustic resistance. The driver is mounted in the outer wall of one of these cavities. In a present-day commercial version of this enclosure (Dynaco Model A-50) the cavities are of equal volume.

The acoustical analogous circuit of the resistance-coupled double-cavity closed-box system is presented in Fig. 6-1. To facilitate direct comparison with the undivided or single-cavity closed-box system, the cavities are designated e CAB and (1-e) CAB so that the compliance of the total volume is CAB. The remaining circuit components are as defined in Chapter 1.

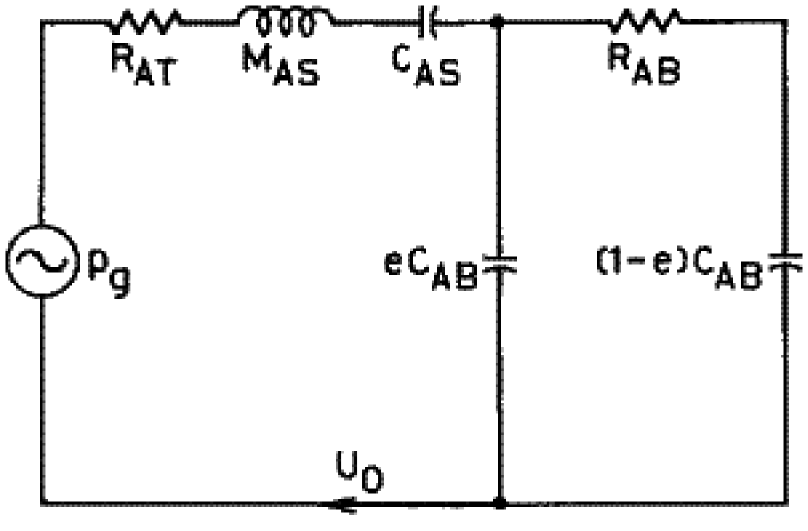

Fig. 6-1 Acoustical analogous circuit of resistance-coupled, double-cavity closed-box loudspeaker system.

The response function of this system is

$$G(s) = \frac{s^3 e(1-e) T_C^3 / Q_A + s^2 T_C^2}{s^3 e(1-e) T_C^3 / Q_A + s^2 [T_C^2 + e(1-e) T_C^2 / Q_A Q_{TC}] + s[T_C/Q_{TC} + (T_C/Q_A)(1-e)(a+e)/(a+1)] + 1} \tag{6-1}$$

where TC is defined by (1-33), QTC by (1-39) and (1-40) with RAB=O, QA by (1-31), and a by (1-42). It is clear that this function has the same asymptotic response as that of the single-cavity system using the same driver, (3-1). However, the extra pole and zero present in (6-1) indicate that the response characteristic can be of a different shape.

The performance obtainable from this system is best illustrated using the analog simulator to represent a typical application of the double-cavity box. The initial system selected is a closed-box system for which the total Q is so high that the response peak is 6 dB. The response of this system is shown by curve A in Fig. 6-2. The enclosure of this system is then divided in half (e = ½), which is representative of commercial design practice. The compliance ratio α, which does not in itself affect the response of the normal closed-box system, is important to this system because the first enclosure cavity determines the coupling of the enclosure resistance to the driver. A value of 5 is chosen here because it provides effective coupling and is also representative of the commercial product.

Fig. 6-2

Frequency response of sample resistance-coupled double-cavity closed-box system (from simulator). System parameters: α = 5, QTS = 0.8, e = 0.5. A: no coupling resistance. B: coupling resistance adjusted for maximum peak reduction (QA = 1.6). C: coupling resistance one-third the value used in B. D: coupling resistance three times the value used in B.

Curve B of Fig. 6-2 shows the response of the double-cavity system with QA adjusted to obtain minimum peak height (QA~116). Curves C and D show the response when the value of QA is made a factor of 3 higher or lower than its optimum value.

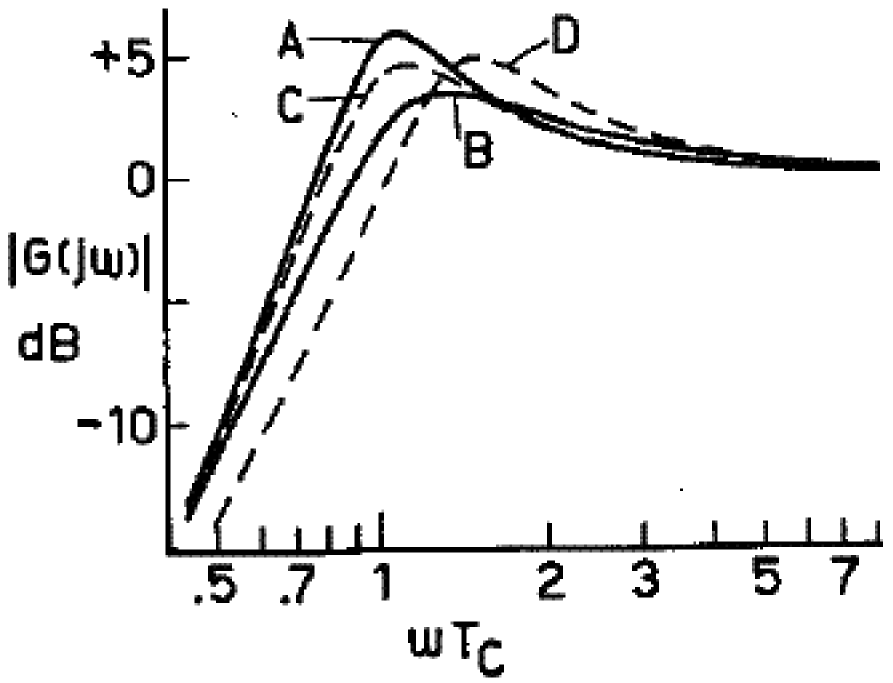

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Figure 6-3 compares the performance of the optimally adjusted double-cavity system to an identical single-cavity system with acoustic damping applied to the driver. Curve A is the minimum-peak-height response of the double-cavity system. Curve B is the response of the single-cavity system with the driver acoustic damping adjusted to obtain the same peak height as curve A; curve C is the response with the driver damping adjusted to obtain the same f3 as curve A.

It may be concluded that: 1) This system is of limited value in reducing large peaks; decreasing the enclosure Q beyond a certain optimum value causes the peak to become higher again, with reduced bandwidth.

2) The much simpler application of acoustic damping to the driver has a greater ability to reduce peaks. For the same peak height, it gives a lower cutoff frequency. For the same cutoff frequency, it provides a greater reduction of the response peak.

The performance of the double-cavity closed-box system can in fact be improved by the addition of a coupling mass between the cavities. The coupling resistance must remain, with the optimum value of QA still about 1.6 for the α = 5 , e = ½ system adjustment. Figure 6-4 illustrates the response of the double-cavity closed-box system with resistive coupling only, curve A, with series mass and resistance coupling, curve B, and with parallel mass and resistance coupling, curve C.

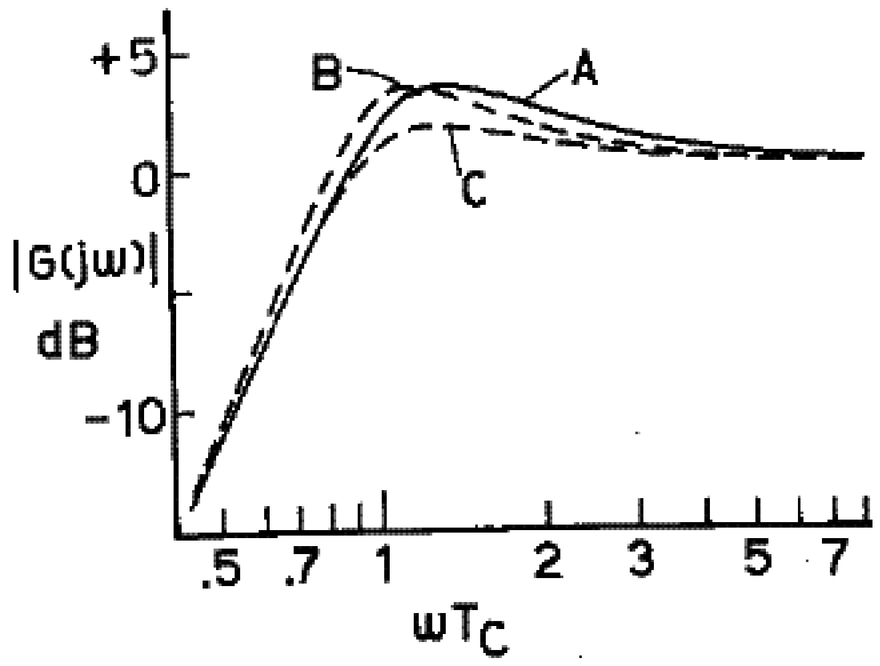
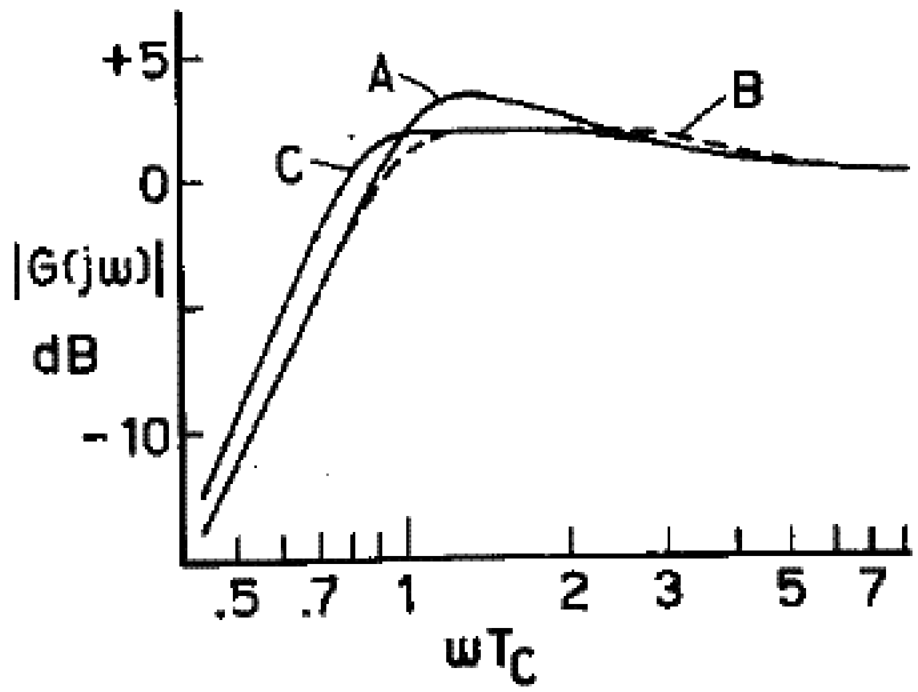


Fig. 6-3

Comparative frequency responses of resistance- coupled, double-cavity closed-box system and equivalent single-cavity closed-box system with driver acoustic damping (from simulator). A: double cavity with optimum coupling resistance (see Fig. 6-2B). B: single cavity, driver acoustic damping adjusted for same peak as A.
C: single cavity, driver acoustic damping adjusted for same cutoff frequency as A.



where QL is defined by (1-32). It is clear from (6-2) that this system has the pure third-order behavior described above.

Fig. 6-4
Frequency response of double-cavity closed-box system with resistance and with mass-resistance coupling (from simulator), System parameters: α = 5, QTS = 0.8, e = 0.5. A: optimum resistance coupling (QA = 1.6).
B: optimum series mass-resistance coupling. C: optimum parallel mass-resistance coupling.

Series mass and resistance does not alter the bandwidth from the resistance-only case, but it lowers the response peak and produces more of a plateau; the "peak" is level for a full octave, The parallel mass and resistance not only reduces the peak by about the same magnitude, it significantly lowers the cutoff frequency. In this respect, it is a definite improvement on the single-cavity system with acoustic damping of the driver; it is the only system investigated which shows such an improvement.

6.5. The Leakage-Damped Closed-Box System

This system may be regarded either as a closed-box system with an acoustic resistance located in a wall of the enclosure or as a vented-box system in which the vent mass is replaced with a pure resistance. This system was "discovered" in the course of the research while looking for a system that would have pure third-order response, i.e. three zeros located at the origin of the complex plane and three poles located away from the origin. It then became obvious that this system is a degenerate case of the damped-vent system described in the next chapter and is probably the intended construction of an existing commercial system (Dynaco Model A-25) called "aperiodic".

The acoustical analogous circuit of the leakage-damped closed-box system is presented in Fig. 6-5. The system response function is

$$G(s) = \frac{s^3 T_C^3}{s^3 T_C^3 + s^2 T_C^2 (1/Q_L + 1/Q_{TC}) + s T_C (1 + 1/Q_L Q_{TC}) + 1/[Q_L (a+1)]} \qquad (6\text{-}2)$$

where QL is defined by (1-32). It is clear from (6-2) that this system has the pure third-order behavior described above.

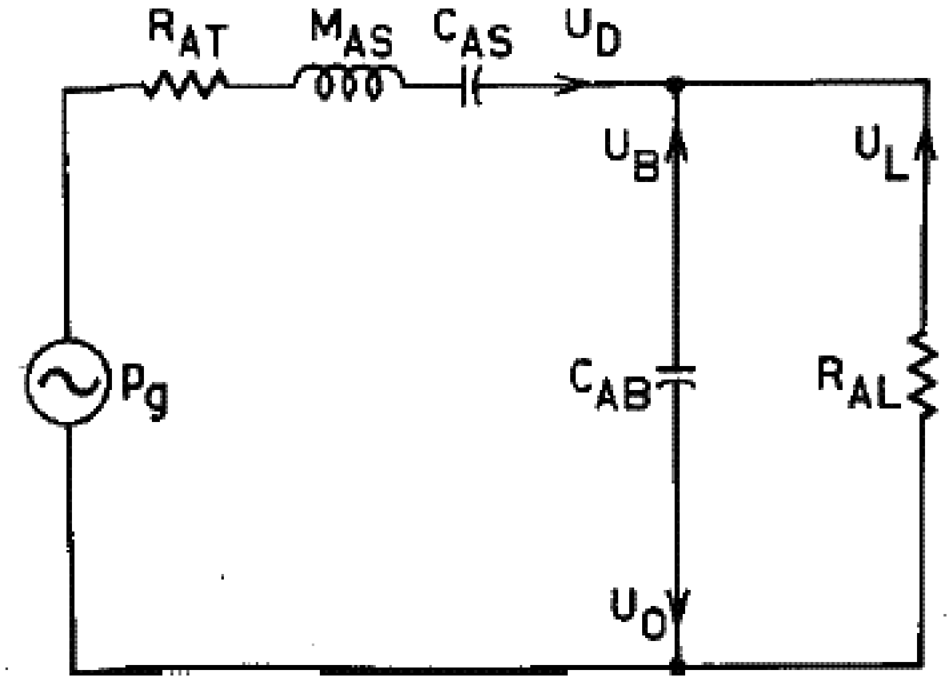


Fig. 6-5

Acoustical analogous circuit of leakage-damped closed-box loudspeaker system.
The system response function (6-2) has four independent variables TC , QL , QTC and α. These variables set the values of only three basic coefficients, so any desired alignment can be obtained with a variety of values of the four variables. Suitable alignments (Section 15.2) are third-order Chebyshev (C3), third-order Butterworth (B3), third-order Sub-Chebyshev (SC3), quasi-second-order Butterworth (QB2), or some form of degenerated Chebyshev response giving a deliberate response peak. An alignment chart for obtaining a B3 response is presented in Fig. 6-6. Note that quite high values of QT = QTC / (α+1)½ can be accomodated by this system,
When the leakage-damped closed-box system is compared to a normal closed-box system with acoustic damping of the driver, some very interesting facts emerge. Using the only commercial version of this system
(α = 5, QTS = 0.66) as a model, Fig. 6-7 shows the simulated response of the driver in a normal closed box, curve A, the response of the leakage-damped system with QL = 2.1, curve B, and the response of the closed-box system with driver acoustic damping adjusted for the same response peak as the leakage-damped system, curve C, or for the same cutoff frequency as the leakage-damped system,curve D.
Figure 6-7 shows clearly that for the same peak height, driver acoustic damping gives a greater bandwidth, and that for the same bandwidth, driver acoustic damping gives better control of the response peak. The same results are found with other system parameters and alignments. But this is not all. Figure 6-8 shows the diaphragm displacement of the systems having the responses illustrated in Fig. 6-7, curves B and D. In Fig. 6-8, curve A is the displacement corresponding to the system of Fig. 6-7(B); curve B corresponds to Fig. 6-7(D). These systems have exactly the same bandwidth, but the leakage-damped system has significantly greater diaphragm displacement which reduces its acoustic power capacity. In the passband this is about 1 dB; below the passband it is much greater, giving increased vulnerability to subsonic signals as for a vented-box system. This feature is also consistent for other alignments of the leakage-damped closed-box system.

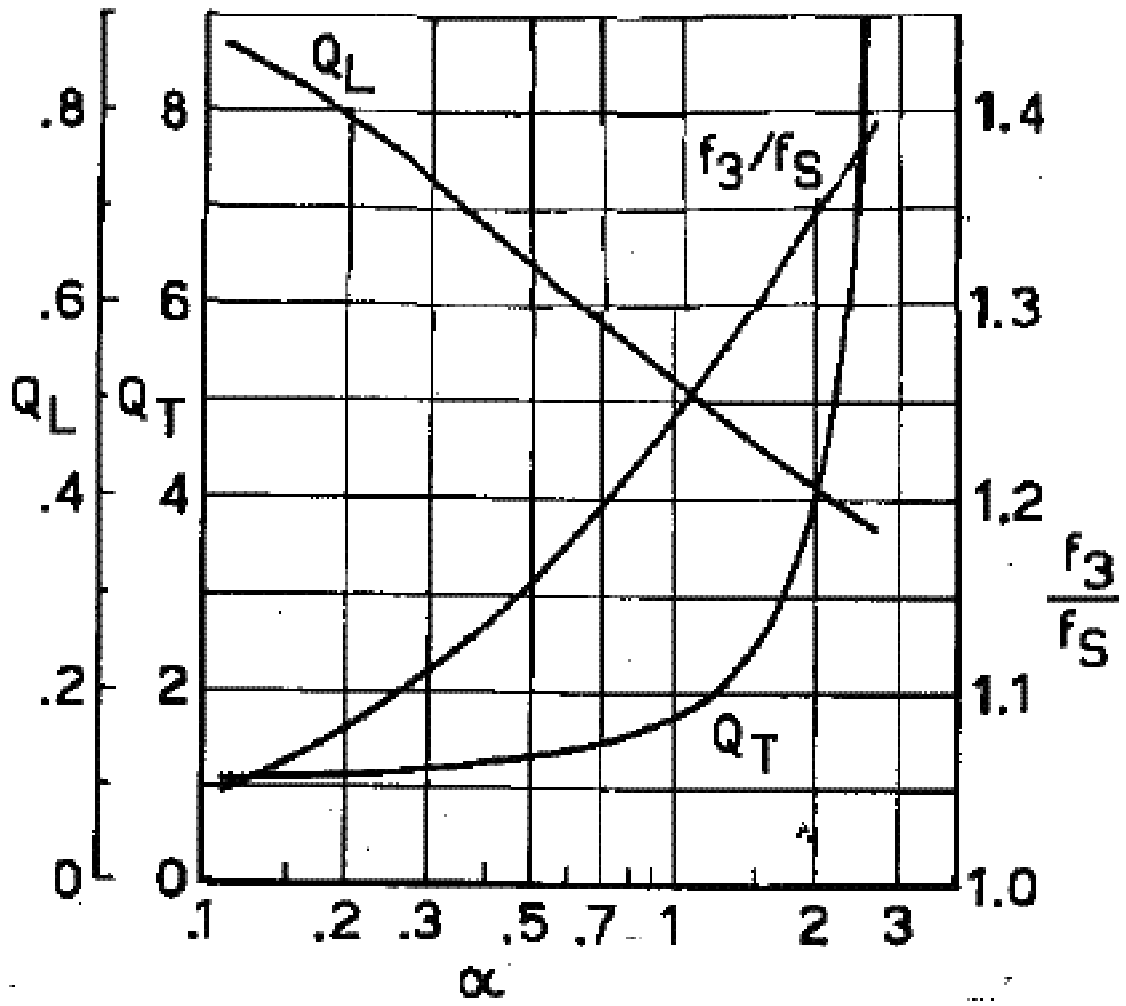


Fig. 6-6
Alignment chart for B3 alignment of leakage-damped closed-box system.

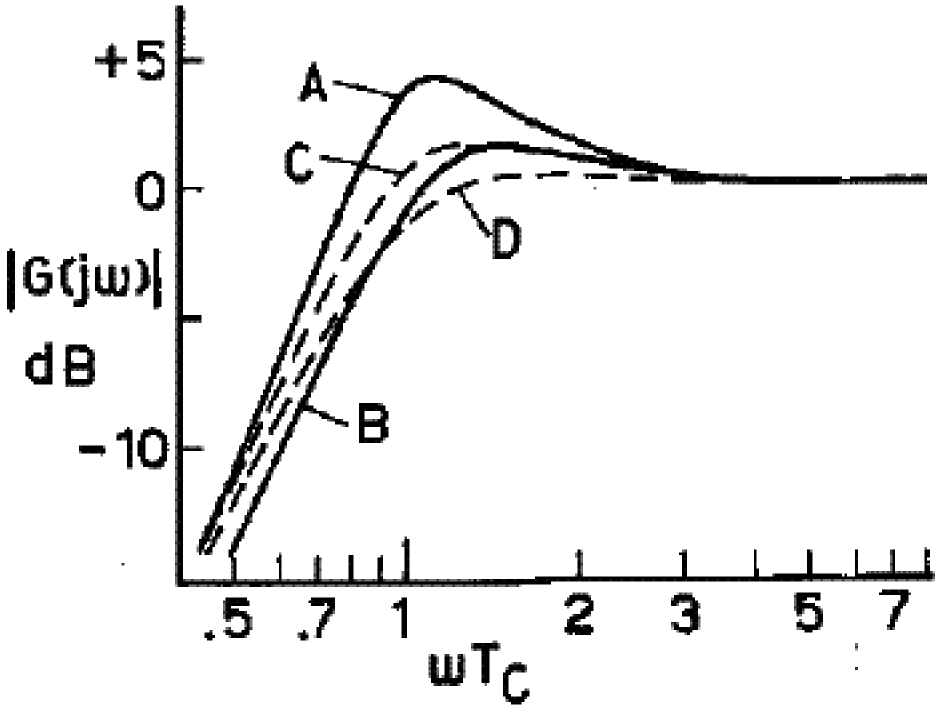


Fig. 6-7
Comparative frequency responses of leakage-damped and driver-damped closed-box systems (from simulator). System parameters: α = 5, QTS = 0.66. A: no damping. B.: optimum leakage damping (QL = 2.1). C: driver acoustic damping adjusted for same peak as B. D: driver acoustic damping adjusted for same cutoff frequency as B.

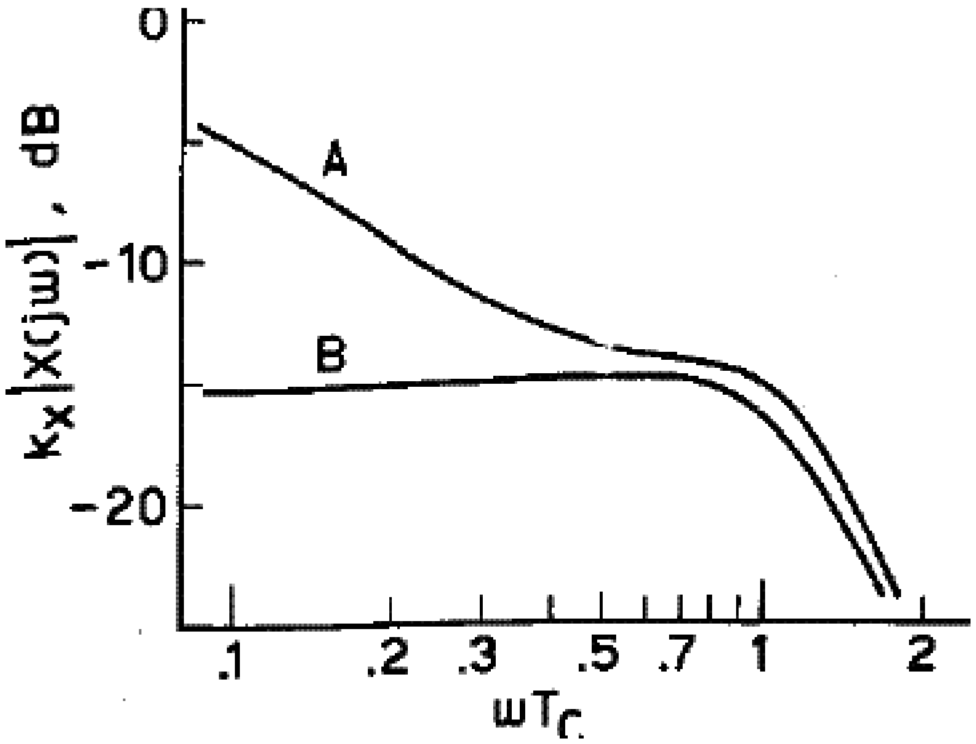


Fig. 6-8

Comparative diaphragm displacement of leakage-damped and driver-damped closed-box systems (from simulator). A: leakage-damped system of Fig. 6-7B. B: driver-damped system of Fig. 6-7D.

7. VARIATIONS OF THE VENTED-BOX SYSTEM

7.1. The Damped-Vent System

The most popular variation of the vented-box system which is supposed to temper the evils of an underdamped driver or of the tuned enclosure itself requires the addition of resistive dissipation to the vent. This may take the form of a resistive material placed over a normal vent, or it may take the form of a "distributed vent" [P1] which consists of many small holes or narrow slots in the enclosure in place of a normal vent. The intention is to retain the mass reactance of the normal vent but to add series acoustic resistance, i.e. to provide a large value of RAP in Fig. 4-1.

The essential components of the acoustical analogous circuit of the damped-vent system are shown in Fig. 7-1. This circuit is simplified to the extent that the absorption loss resistance RAB and leakage loss resistance RAL have been omitted. The justification for this is that the vent loss in this system is normally much greater than the other losses. Also, the usefulness of vent damping may be more easily assessed by comparing the simplified vent-damping-only model against the lossless vented-box system, the contributions of other losses being assumed to be about the same for real systems of either type.

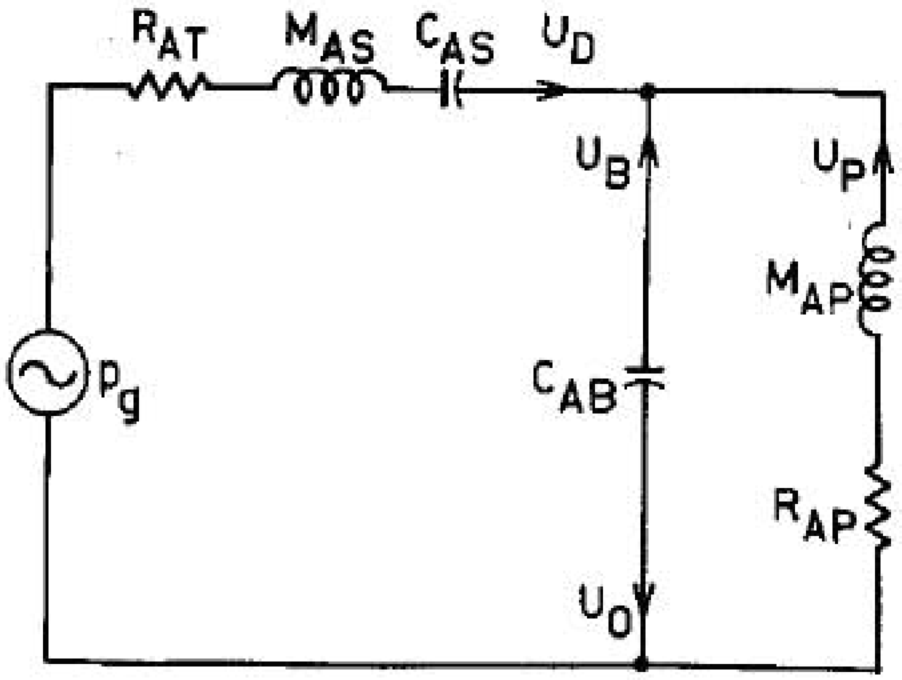


Fig. 7-1
Acoustical analogous circuit of damped-vent loudspeaker system.
From analysis of Fig. 7-1, the response function of the damped-vent system is

$$G(s) = \frac{s^4 T_B^2 T_S^2 + s^3 T_B T_S^2 / Q_P}{s^4 T_B^2 T_S^2 + s^3 (T_B^2 T_S / Q_T + T_B T_S^2 / Q_P) + s^2 [(a+1) T_B^2 + T_S^2 + T_B T_S / Q_P Q_T] + s [(a+1) T_B / Q_P + T_S / Q_T] + 1} \quad (7\text{-}1)$$

where the variables are as defined in Section 1.5. This function has third-order cutoff behavior, but has an extra pole and zero.

If the extra pole and zero are made equal, the response becomes pure third-order. The condition for this simplification is

$$1 / Q_T = h / Q_P + Q_P / h \quad (7\text{-}2)$$

this condition in fact restricts QT to values less than 0.5. The response may be specified to be C3, B3, SC3 or QB2 (Section 15.2). Figure 7-2 is an alignment chart for the B3 response. Clearly, large values of QT cannot be tolerated. If the B3 alignment for $\alpha = \sqrt{2}$ is compared to the lossless B4 alignment in Fig. 4-6, the QT requirement is found to be the same, but the cutoff frequency of the B3 alignment is much higher. Even a slight rise in QT requires a very much larger enclosure for this system, so it is clearly not a very advantageous one.

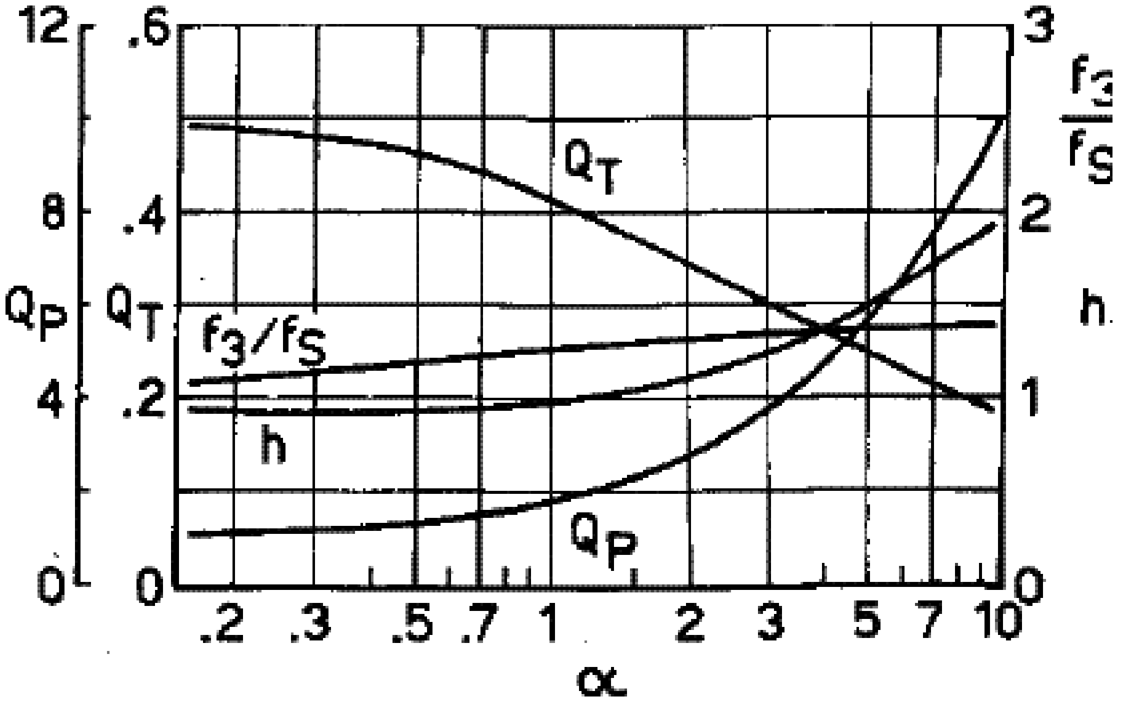
Allowing the extra pole and zero to be non-coincident does not improve matters much. "Flat" responses, e.g. as obtained for the general maximally-flat condition of equal numerator and denominator coefficients for like powers of $(\omega^2 T_0^2)$ in $|G(j\omega)|^2$ [W1], can only be obtained with QT values of approximately the same magnitude as required by the normal vented-box system.

Fig . 7-2
Alignment chart for B3 alignment of damped-vent system.

For a badly underdamped driver, experiments on the analog simulator show that damping of the vent provides only a partial reduction of the response peak. The effectiveness of vent damping in reducing response peaks is greater for higher compliance ratios and is improved by reducing the vent mass, i.e. by moving toward the conditions of the leakage-damped closed-box system of Section 6.5. But acoustic damping of the driver is much more effective in correcting the response peak; it also yields a higher acoustic power capacity, because it does not increase diaphragm displacement.

Figure 7-3 illustrates (A) the response of a simulated vented-box system with a compliance ratio of 5 and a QTS of 0.5 (just twice that required for a "flat" QB3 response) and the change in response that is obtained by (B) damping the vent, (C) damping the vent and reducing vent mass, and (D) applying acoustic damping to the driver to obtain the same response peak as in (C). Figure 7-4 illustrates the diaphragm displacement required of the driver in each case.

The relationships illustrated in Figs. 7-3 and 7-4 are quite representative of other compliance ratios as well. The damped-vent system is inferior to the normal vented-boxsystem with acoustic damping of the driver in three important respects: ability to reduce response peaks, bandwidth for a given peak reduction, and acoustic power capacity for a given response peak and bandwidth. The only apparent advantage of the damped-vent system is a reduced displacement sensitivity to subsonic signals over a particular frequency range.



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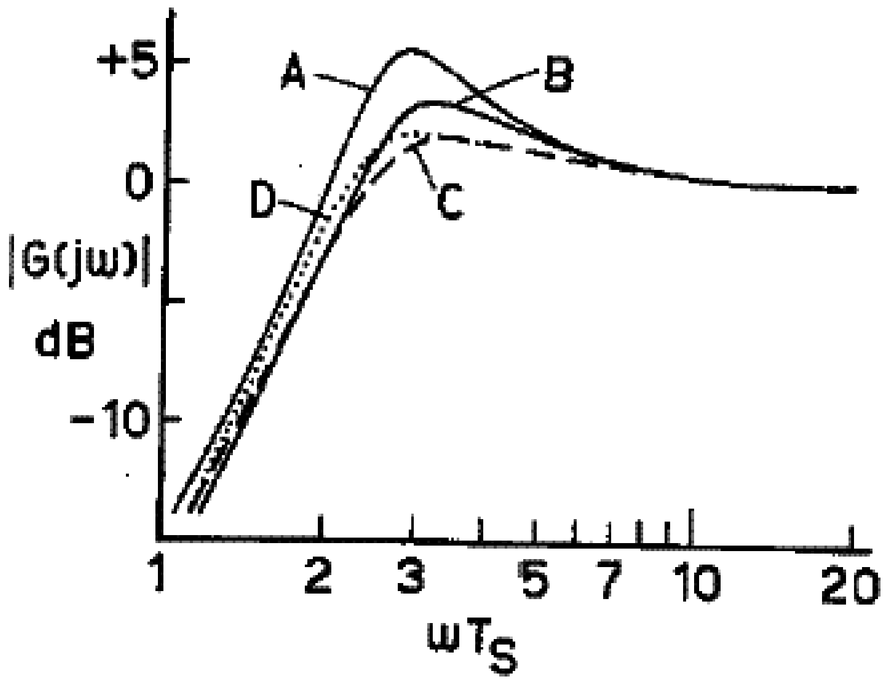

Fig. 7-3

Frequency response of damped-vent system (from simulator)+ A: underdamped system with α =5, QT = 0.5, h = 1.49. B: optimum amount of vent damping added (QP = 0.67). C: best experimental combination of vent damping and reduced vent mass (h = 2.9, QP = 0.17). D: original system with driver acoustic damping adjusted for same peak as C (QT = 0.32).

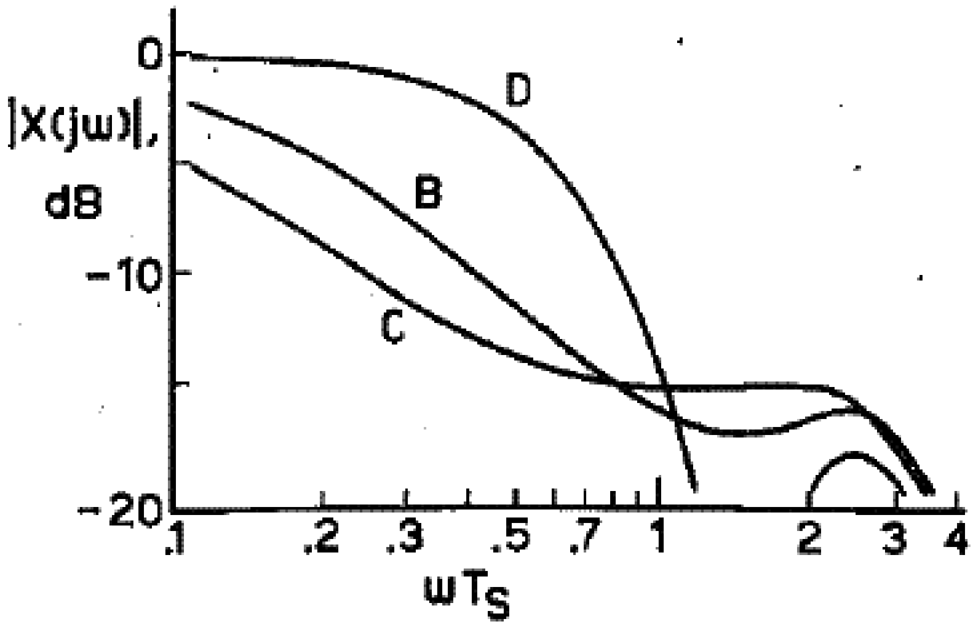

Fig. 7-4

Driver diaphragm displacement for the corresponding system adjustments of Fig. 7-3 (from Simulator).

If the response peak of a vented-box system is relatively modest, it is easily shown both theoretically and experimentally that closing the vent entirely, i.e. changing to a closed-box system, will provide a satisfactory peak reduction with a better bandwidth than the damped-vent system.

7.2. The Leakage-Damped Vented-Box System (ARU)

Another popular modification to the vented-box system is the addition of a large, deliberate enclosure leak. This takes the form of an acoustic resistance mounted in a wall of the enclosure which is separate from and in addition to a normal high-Q vent. The system was first proposed by E. J. Jordan [J1],[J2] and is also known as the ARU system after the "acoustic resistance unit" which Jordan designed to provide the functions of both vent and leak.
Jordan's original vague description of this system led to some misinterpretation of its operation [N4]. No quantitative analysis or description has ever been published. Jordan appears to have ignored the vent and leakage volume velocities in his study of the system and to have relied on interpretation (or misinterpretation) of voice-coil impedance measurements in assessing results.
The acoustical analogous circuit of the leakage-damped vented-box system is identical to Fig. 4-3, and the response function is given by (4-1). The response is purely fourth-order and can be compared directly to the "normal" vented-box system. The distinguishing difference is the much lower value of QL that is used in the ARU system. From Fig, 4-15, it is to be expected that values of kη (G) will be very low for this system.

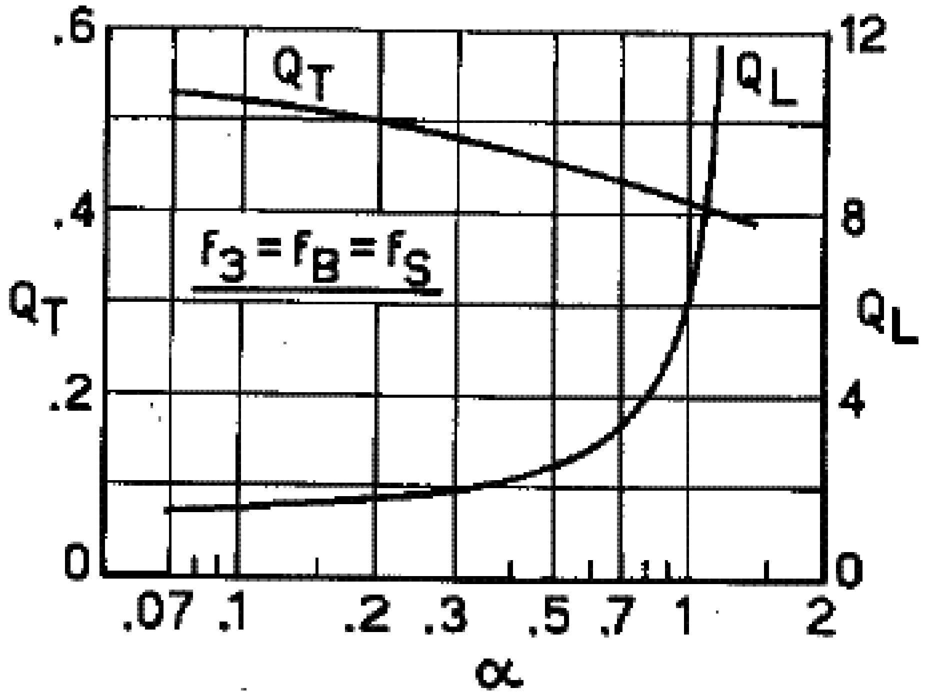

Fig. 7-5

Alignment chart for B4 alignment of leakage-damped vented-box system.
Figure 7-5 is an alignment chart for the ARU system to provide a B4 response. Because this system has one extra variable (QL), this response can be obtained with an infinite variety of alignments. The ARU system tolerates a higher value of QT than a normal low-loss vented-box system, but not very much higher. And quite significantly, a modest rise in QT requires a considerable increase in enclosure size to maintain the same cutoff frequency. For similar performance, the ARU system always requires a larger enclosure than a normal vented-box system with acoustic damping of the driver.
The ability of the ARU system to control response peaks was investigated on the analog simulator. Starting with a normal vented-box system having double the required value of QT, i.e. a 6 dB response peak, it was found that the addition of enclosure leakage could eliminate the peak completely but also raised the cutoff frequency almost an octave above that of the normal system. Allowing a "tolerable" peak of 2 dB the cutoff frequency was still from 20 to 30 % higher than that of the normal system using acoustic damping of the driver to obtain the same peak. The tests were for a values of 1.4 to 5, quite representative of typical designs. Retuning of the enclosure and readjustment of the leakage resistance to maintain the same 2 dB peak showed that lower values of h produced better bandwidth, but still 10 to 15 % worse than the normal system; the diaphragm displacement also increased. Higher tuning raised the cutoff frequency.

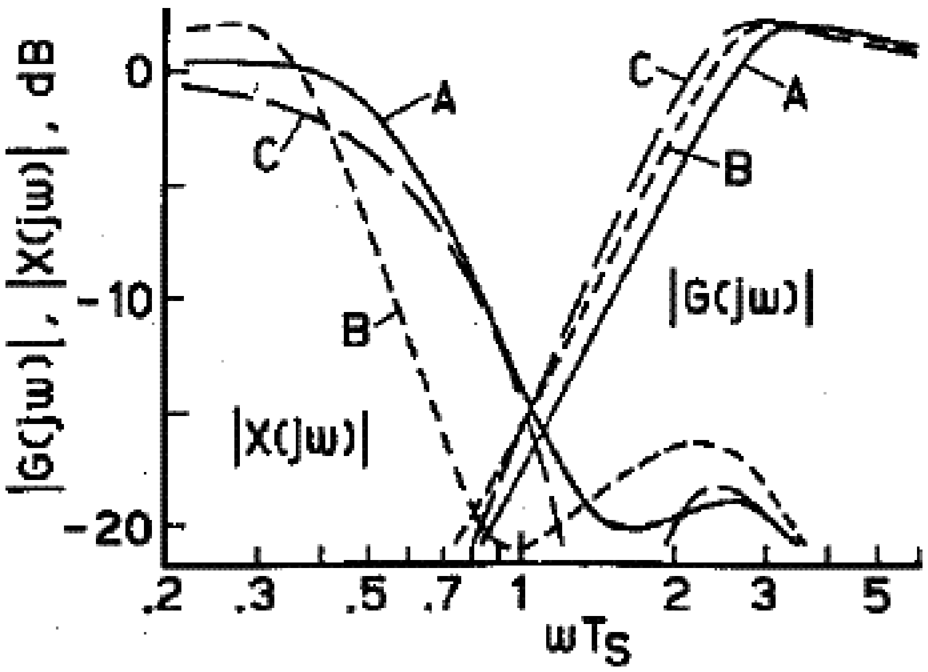

Figure 7-6 illustrates the response of the ARU system for α = 5, QT = 0.5, with both normal and low tuning (curves A and B) and the response of the normal system with driver acoustic damping, curve C. A response peak of 2 dB has been allowed in each case. Shown on the same figure at the left is the diaphragm displacement for each system. The low-tuning case gives significantly increased passband displacement compared to the normal system despite its poorer bandwidth. The undamped vent in the ARU system allows diaphragm displacement to rise rapidly below fB in contrast to the damped-vent system treated earlier
(Fig. 7-4).

Fig. 7-6

Comparative frequency responses and diaphragm displacements of leakage-damped and driver-damped vented-box systems for α = 5 and QT = 0.5 (from simulator), A: normal tuning, leakage damping (h = 1.49, QL = 1.36). B: low tuning, leakage damping (h = 0.9, QL = 1.8). C: driver acoustic damping (QT = 0.32) . The damping is adjusted for a 2 dB response peak in each case.

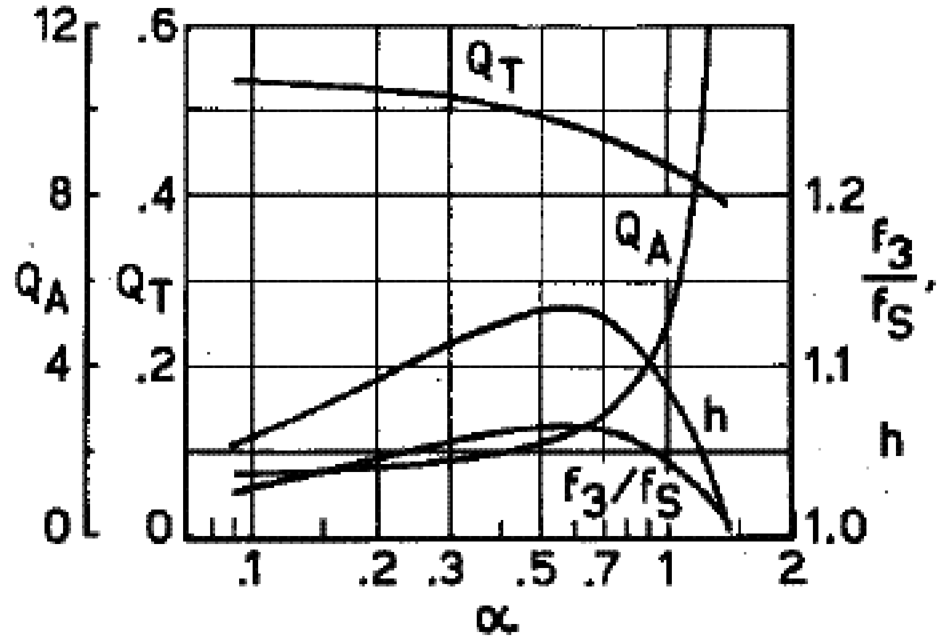

Fig. 7-8
Alignment chart for B4 alignment of absorption-damped vented-box system.

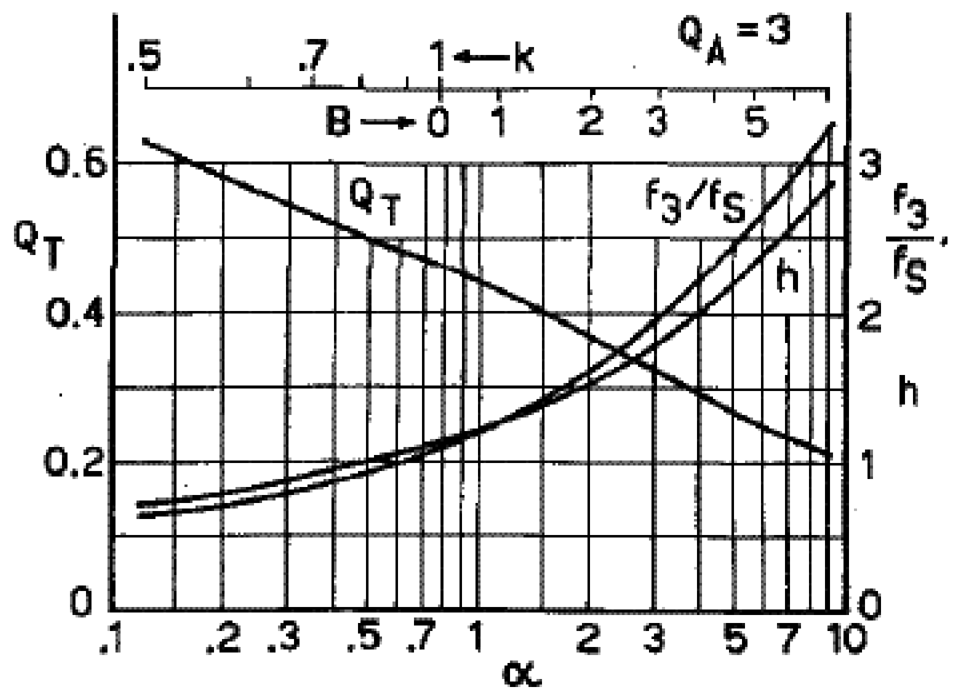

Fig. 7-9
Alignment chart for absorption-damped vented-box system with QB=QA=3.

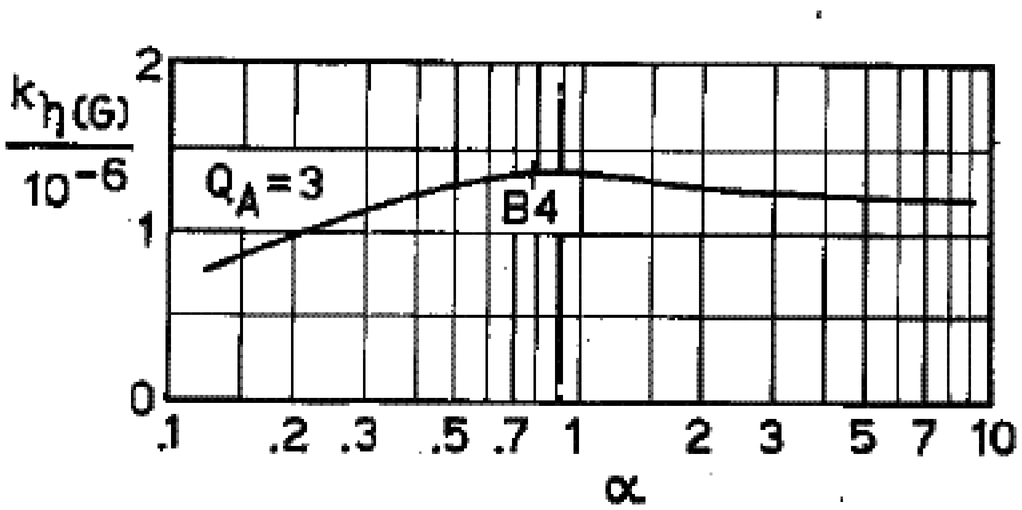

Fig. 7-10
Response factor kn(G) of efficiency constant as a function of α for the absorption-damped vented-box system with QB=QA=3.

In Fig. 7-10, the value of kn(G) is plotted as a function of α for the alignments given in Fig. 7-9. For low values of α, this has the same values as the QL=3 curve in Fig. 4-15, but for high values of α it is about 25 % lower. The overall value of kn for practical systems of this type must include an additional factor kn(C) given by

$$kn(C) = V_{AB}/V_B \,, \qquad (7\text{-}4)$$

7.3. The Absorption-Damped Vented-Box System

Another method of adding damping to the enclosure of a vented-box system is to fill the enclosure with an acoustically resistive damping material similar to that used in closed-box air-suspension systems. Such a system has been described in recent times under the name "resistive reflex" [06],[07],[08]. In addition to the assumed virtues of the added enclosure damping, the filling material provides some direct acoustic damping of the driver [06, p. 749]. A further advantage apparently missed by the advocates of this system, despite its evident effects in their voice-coil impedance data, is that the material increases the effective compliance of the enclosure just as it does for a closed-box system. These effects may be analyzed separately.

The value of enclosure absorption damping is assessed by assuming that the damping added to the system is reflected entirely in a low value of QA, i.e. that the acoustical circuit may be simplified to that of Fig. 7-7, that the values of QL and QP remain large enough to be ignored, and that QM remains unaffected. In practice, this requires that the damping material be kept reasonably clear from the vent and driver. If it is allowed to obstruct the inside of the vent aperture, a low value of QP results; if it is allowed to cover the rear of the driver, QT is reduced by acoustic damping.

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From analysis of the circuit of Fig. 7-7, the system response function is

$$G(s) = \frac{s^4 T_B^2 T_S^2}{s^4 T_B^2 T_S^2 + s^3(\alpha T_B^3/Q_A + T_B^2 T_S/Q_T + T_B T_S^2/Q_A) + s^2[(\alpha+1)T_B^2 + T_B T_S/Q_A Q_T + T_S^2] + s(T_B/Q_A + T_S/Q_T) + 1} \qquad (7\text{-}3)$$

The system response is thus fourth-order and may be compared to the normal or leakage-damped vented-box systems. Figure 7-8 is an alignment chart for the absorption-damped vented-box system to provide a B4 response. This may be compared directly with Fig. 7-5: the allowable values of QT are only slightly higher, but the normalised cutoff frequency is also higher. Figure 7-9 is an alignment chart for the C4-B4-QB3 alignments with QA fixed at 3. This is a physically attainable absorption Q using light-weight filling materials. Figure 7-9 may be compared directly with the QL=3 chart of Fig. 4-11.

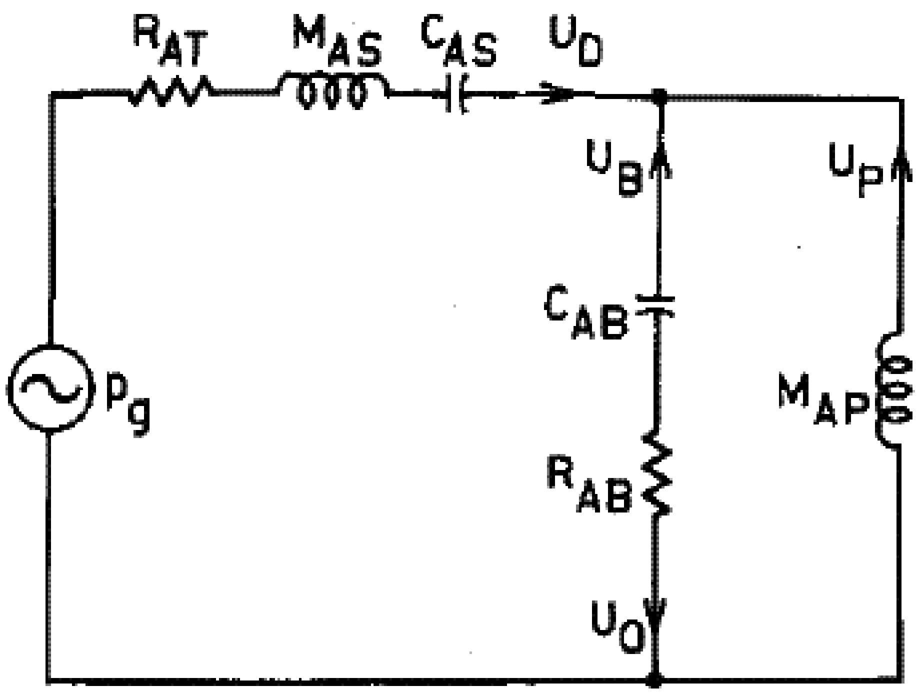

Fig. 7-7
Acoustical analogous circuit of absorption-damped vented-box loudspeaker system.

because the enclosure compliance is increased by the filling material. Also, the effective value of kn(Q) is reduced by the effects of driver damping for the material near the driver.

In practice, the compliance increase for closed-box systems is typically 15 % and seldom more than 20 % when filling material is added. Assuming the same values, the absorption-damped system is likely to have a higher overall value of kn compared to the ARU system for low- α alignments but a lower overall value for high- α alignments. Both systems have a substantially lower value of kn than a normal minimum-loss vented-box system, and both -- for flat response -- require values of QT which are only slightly higher than those for the normal system.

The ability of enclosure absorption damping to control response peaks in systems with underdamped drivers has been investigated with the analog simulator. The results are very similar to those for the leakage-damped system in that control of peaks results in higher cutoff frequencies and greater diaphragm displacement than comparable control obtained from direct acoustic damping of the driver. The degree of control is somewhat better, with proportionately less loss of bandwidth, than for the leakage-damped system. However, this is not surprising in view of Fig. 4-2.

Figure 7-11 illustrates the response of the absorption-damped system for α =5 and a value of QT that would produce a response peak of 6 dB in a normally aligned system. QA has been-adjusted to just flatten the response in curve A. In curve B, acoustic damping is added to the driver to obtain a flat response. At the left of the figure is the diaphragm displacement for each case. The cutoff frequency with absorption damping is 30 % higher, and the diaphragm displacement at and below fB is significantly higher. Even after allowance for a decreased compliance ratio due to the damping material, the cutoff frequency is still higher. As with the leakage-damped system, raising the enclosure tuning reduces bandwidth, while lowering the tuning extends the response only slightly but causes greatly increased diaphragm displacement in the passband as shown by curve C.

The absorption-damped vented-box system is appealing to the extent that the enclosure filling increases compliance and tends to damp standing waves in the enclosure at higher frequencies and thus reduce coloration. The effects of material losses, however, are much more drastic than in the closed-box system. The use of thick medium-density lining materials (e.g. 2 to 3 inch thick fiberglass) in some commercial systems would appear to gain some benefit as far as both compliance increase and damping of standing waves are concerned, without contributing excessive losses either in the form of enclosure absorption or vent friction. It is possible that the effects of the moderate additional absorption losses caused by the thick material are just about offset by the effects of the compliance increase in the region occupied by the material.

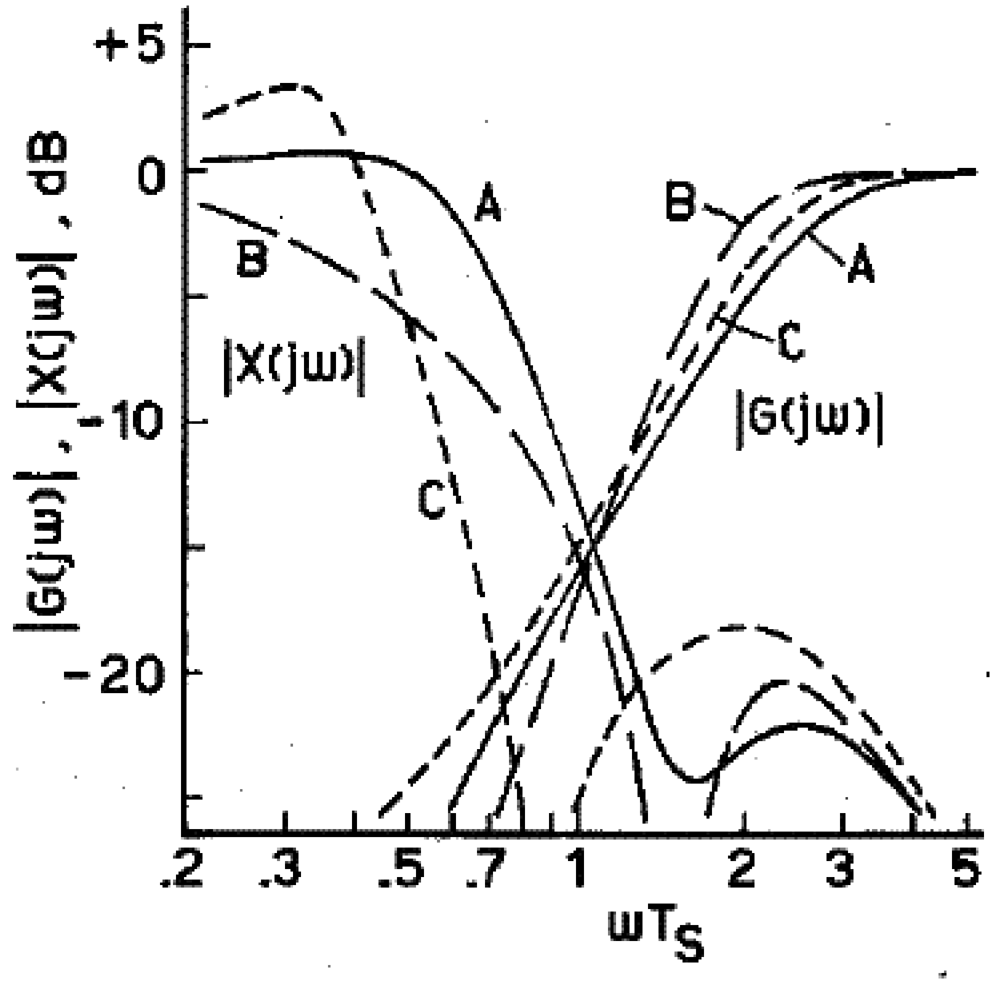

Fig. 7-11

Comparative frequency responses and diaphragm displacements of absorption-damped and driver-damped vented-box systems for α =5 and QT=0.5 (from simulator). A: normal tuning, absorption damping (h-1.49, QA=2.3). B: driver acoustic damping (QT=0.25). C: low tuning, absorption damping (h=.87, QA=5.2). The damping is adjusted for a flat response in each case.

7.4. The Damped-Box, Damped-Vent (Aperiodic) System

Another variation of the vented-box system requires the application of damping to both the vent and the enclosure. The enclosure damping is in the form of internal absorption, not leakage, This system was proposed by de Boer and called the "aperiodic" system [D3]. The acoustical analogous circuit of this system shown in Fig. 7-12 corresponds exactly to that given by de Boer. The apparent intention behind this system is to adjust the enclosure impedances so that

$$R_{AB} = R_{AP} = R = (M_{AP}/C_{AB})^{1/2} \quad (7\text{-}5)$$

The enclosure under these conditions is equivalent to a pure acoustic resistance of value R as seen from the driver aperture. It follows that the driver behavior is then the same as if the driver were mounted on an infinite baffle and provided with acoustic damping.

It is physically possible to obtain the conditions called for in (7-5). However, de Boer's conclusion that the system response will then be that of an infinite-baffle system is quite wrong. He assumed that because the vent is heavily damped its volume velocity must be negligible; it is not. It is in fact substantial, and it opposes that of the driver. The driver volume velocity is indeed that for an infinite-baffle system, but the system output depends on the total enclosure volume velocity.

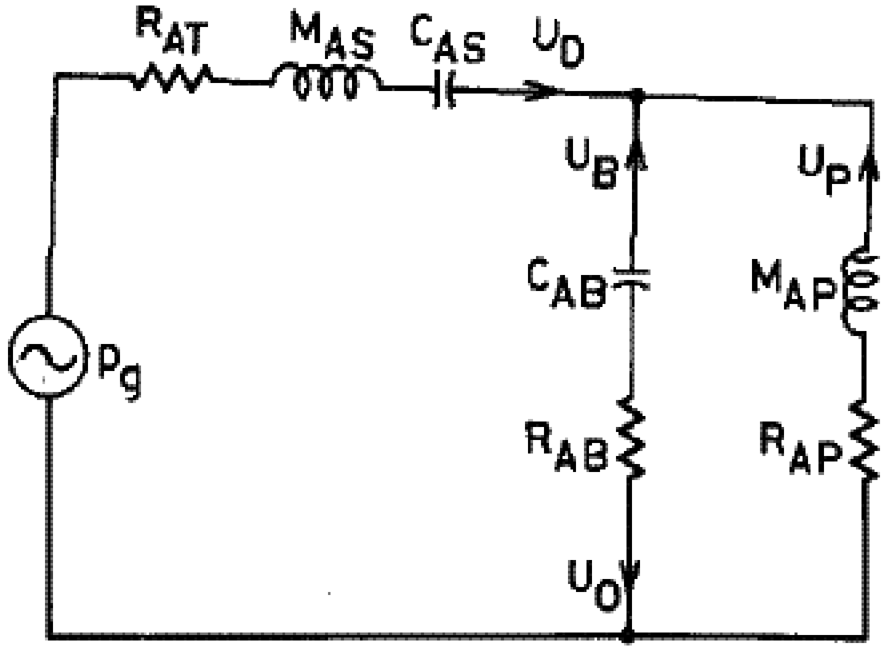

Fig. 7-12

Acoustical analogous circuit of damped-box, damped-vent "aperiodic" system.
From the circuit of Fig. 7-12, the system response function is

$$G(s) = \frac{s^4 T_B^2 T_S^2 + s^3 T_B T_S^2 / Q_P}{\begin{aligned} & s^4 T_B^2 T_S^2 \\ & + s^3[\alpha T_B^3/Q_A + T_B^2 T_S/Q_T + T_B T_S^2 (Q_A+Q_P)/Q_A Q_P] \\ & + s^2[(\alpha+1)T_B^2 + T_S^2 + \alpha T_B^2/Q_A Q_P + T_B T_S (Q_A+Q_P)/Q_A Q_P Q_T] \\ & + s[T_B/Q_A + (\alpha+1)T_B/Q_P + T_S/Q_T] \\ & + 1 \end{aligned}} \qquad (7\text{-}6)$$

The response is clearly third-order, with an extra pole and zero. As with the damped-vent system, the response becomes pure third-order for the condition imposed by (7-2), but this requires QT to be less than 0.50 for all cases.
From Fig. 4-2 it would appear that adding acoustic resistance to the enclosure (reducing QA) should be somewhat more effective in reducing response peaks than adding resistance to the vent alone (reducing QP). This is found to be true, but the "aperiodic' system is still inferior to a normal vented-box system using acoustic damping of the driver.
Figure 7-13 illustrates the results of aligning the aperiodic system in the manner described by de Boer using the analog simulator. Curve A is the output of the driver for all alignment conditions. Curves B and C are the total output for systems with a equal to 1 and 2 respectively and QT=2.4. Curve D is the total output for α=2, QT=1.1. Not only is the cutoff frequency of every system much higher than hoped for, the excess diaphragm displacement given by the difference between curves A and B, C or D severely reduces acoustic power capacity for the effective bandwidth and increases modulation distortion [K2].

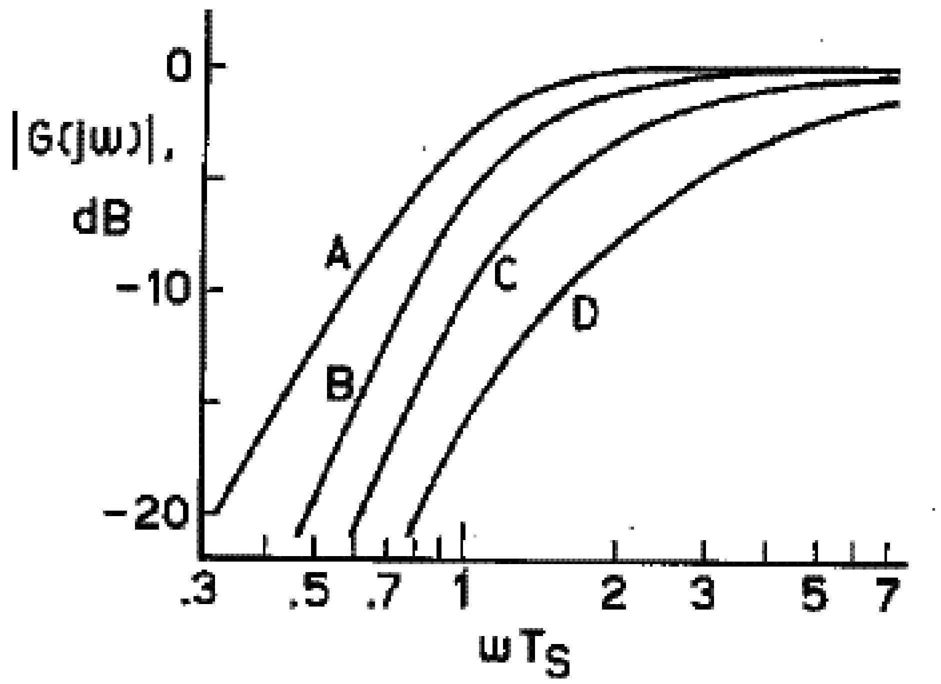

Fig. 7-13
Frequency response of aperiodic system for several alignments (from simulator). A: driver output for all cases. B: total output for α =1, QTS=2.4. C: total output for α =2, QTS=2.4. D: total output for α =2, QTS=1.1.

7.5. The Double-Cavity Vented-Box System

Various schemes have been proposed for improving the performance of vented-box systems by dividing the enclosure into two cavities. The driver is mounted in an aperture of the first cavity, while the vent is mounted in an aperture of the second. The two cavities are then coupled by a vent (i.e. an acoustic mass), an acoustic resistance, or both.

A version of the mass-coupled double-cavity vented-box system was produced commercially as the "Acoustical Corner Ribbon Enclosure" [J1, Part 2]. The acoustical analogous circuit of this system is shown in Fig. 7-14. The coupling mass is designated MAB. It is easily shown that the response of this system has a notch in the passband. This is illustrated in the simulator curve A of Fig. 7-15. The notch may be reduced by resistive damping, i.e. by placing damping material across the coupling vent. This corresponds to placing a resistance RAB in series with MAB in the circuit of Fig. 7-14. Curve B of Fig. 7-15 illustrates the effect of such a resistance on the response.

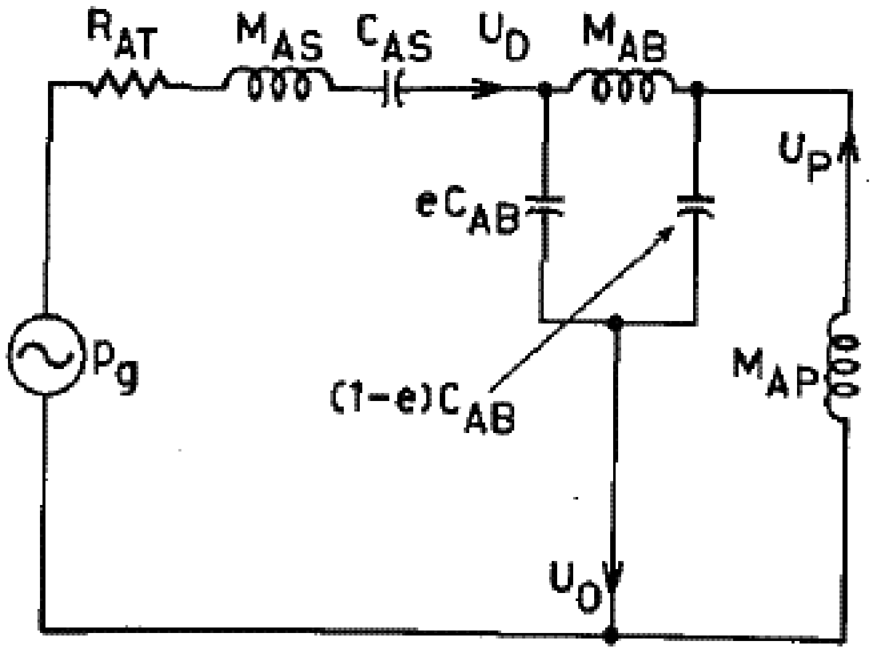

Fig. 7-14
Acoustical analogous circuit of double-cavity vented-box loudspeaker system.

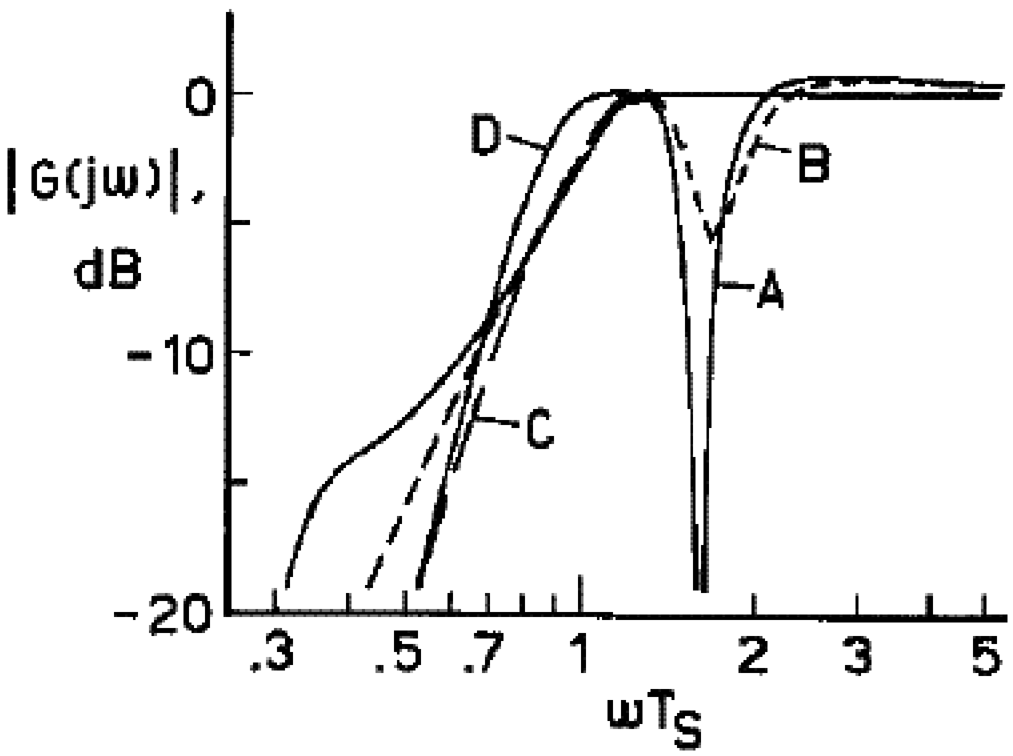

Fig. 7-15

Comparative frequency responses of double-cavity and normal vented-box systems (from simulator). System parameters: α =1, QTS=0.5, e=0.5. A: mass-coupled double-cavity system, no damping. B: parallel damping resistance added to system of A. C: resistance-coupled double-cavity system. D: single-cavity (normal) vented-box system with same driver and total enclosure size as A, B and C but with driver acoustic damping adjusted for flat response (QT=0.42).

The inherent notch present in the above system is completely eliminated by removing the coupling mass MAB and using only resistance as a coupling element. Such a resistance-coupled double-cavity vented-box system has been used in the past by the BBC [S1]. Curve C of Fig. 7-15 illustrates the smooth response that can be obtained in his way. This particular system as set up on the analog simulator has a higher value of QT than could be tolerated by a normal vented-box system. The technique can thus be used to control response peaks that would occur in a normal system, but only by paying the same penalty as the nearly equivalent method of damping the enclosure of a single-cavity system as in Section 7.3. Curve D of Fig. 7-15 is the response of the same system with a normal single-cavity enclosure of the same total volume using acoustic damping of the driver to flatten the response.

There is no advantage in dividing the enclosure of a vented-box system into two (or more) cavities. If an under- damped driver must be used, acoustic damping of the driver in a normal enclosure provides superior performance in terms of bandwidth and acoustic power capacity. In recent times, the latter approach has been used successfully by the Australian Broadcasting Commission in its monitoring loudspeakers [A. N. Thiele, private communication] and advocated by Novak [N3]. J. E. Benson, who at one time designed a successful double-cavity vented-box system [B5], has recently studied the problem more thoroughly and come to the conclusion that "the optimum double cavity is a single cavity" [private communication].