Eq. (37) may be written in the form

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

where k_P is a power rating constant given by

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

Values of (f_3/f_S) may be calculated for any alignment. From Fig. 14 the quantity $k_x|X(j\omega)|$ has two maxima, one within and one below the passband, just as for the vented-box system. For the passband maxima, the magnitudes are very little different from those of comparable vented-box alignments. The alignment data are also similar, particularly for large δ. Thus for average program material having most of its energy within the system passband, the power ratings must be about the same as for vented-box systems, i.e. [5, eq. (41)],

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

For a graphical illustration of this relationship between acoustic power rating, cutoff frequency, and driver displacement volume, see [5, Fig. 19].

Note that this rating is not affected by the displacement reduction that occurs at *very* low frequencies for the passive-radiator system, because this reduction does not extend to the frequency range near cutoff. However, it is reasonable to expect that the passive-radiator system should be somewhat less vulnerable to *very*-low-frequency signals such as amplifier turn-on and turn-off transients and the too hastily lowered pickup stylus.

Electrical Power Rating

The displacement-limited electrical input power rating P_{ER} of the passive-radiator system may be obtained by dividing the acoustic power rating by the system reference efficiency. The dependence of this rating on the important system parameters is observed by dividing Eq. (39) by Eq. (29):

$$P_{ER} = \frac{P_{AR}}{\eta_0} = \frac{k_P}{k_\eta} f_3 \frac{V_D^2}{V_B} . \qquad (42)$$

6. PASSIVE-RADIATOR REQUIREMENTS

The effective surface area of the passive radiator is usually made equal to that of the driver. This condition is not necessary for successful operation, but several factors encourage it. It was stated earlier that the passive radiator is often made from the same frame and suspension as the driver; the economic advantages of this approach are readily apparent, and it results in equal areas.

The use of a passive radiator which is substantially larger than the driver is seldom feasible because of the required baffle area. In most cases the size of both driver and passive radiator are limited by the enclosure dimensions, and it is impractical to make the passive radiator area more than about twice that of the driver.

The alternative of making the passive radiator smaller than the driver is almost never encountered. The principal reason for this is that the volume displacement required of the passive radiator is quite substantial. A small area therefore requires a very large linear displacement capability which can be difficult to achieve in practice.

In Section 5 the power capacity of the passive-radiator system is determined on the basis that the limiting factor is the displacement volume of the driver. If this power capacity is to be realized in practice, the passive radiator must be designed so that it is capable of displacing the maximum volume required of it by the system at rated power output. This volume displacement requirement is normally larger than that of the driver and is the physical reason for the relatively high power rating constant of the system.

The relative maximum volume displacement requirements for the driver and passive radiator may be found from Eqs. (23) and (25), recognizing that at zero frequency the passive-radiator volume displacement must be $\delta/(\delta+1)$ of that of the driver as noted in Section 2. Fig. 15 illustrates the relative displacements as a function of

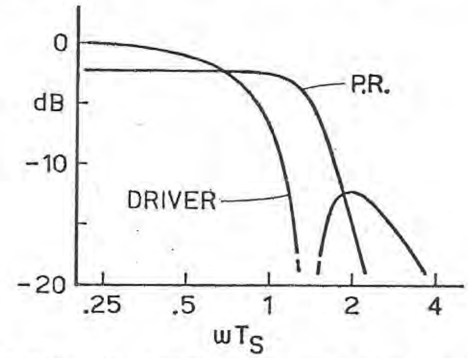

Fig. 15. Normalized displacements of driver and passive-radiator as a function of normalized frequency for lossless maximally flat $\delta=\alpha$ passive-radiator system alignment.

frequency for the lossless maximally flat $\delta=\alpha$ alignment. The maxima occur at different frequencies, but, most importantly, high passive-radiator displacement is required within the system passband.

For program-rated systems, the passive radiator displacement volume V_{PR} must typically be about twice the rated driver displacement volume V_D. Fig. 16 is a plot of the required ratio of V_{PR} to V_D as a function of α for all of the $\delta=\alpha$ alignments. If driver and passive radiator have the same effective surface areas, the maximum linear displacements must be in this ratio.

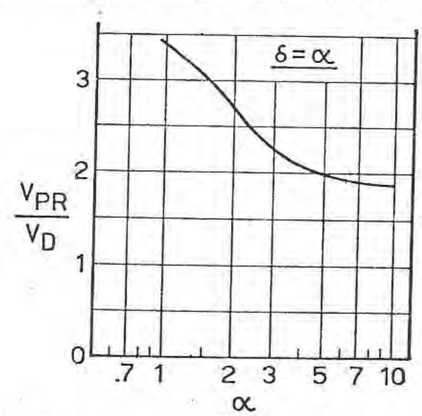

Fig. 16. Required ratio of passive-radiator displacement volume V_{PR} to driver displacement volume V_D as a function of α for program-rated $\delta=\alpha$ passive-radiator systems (from simulator).

Not all high-quality drivers have a suspension capable of more than twice the linear displacement that the magnet/voice-coil structure can provide with good linearity. For this reason, optimum design of passive-radiator systems may require that the passive-radiator suspension be somewhat different from that of the driver. The "convenience" of using the same suspension may in fact result in limited power capacity compared to that which could be achieved with a specially designed passive radiator.

An interesting feature of the $\delta=\alpha$ alignments is the small variation of the required value of $y=(f_P/f_S)$. For the most common alignments, a passive radiator made from the same frame and suspension as the driver (assuming adequate displacement capability) consistently requires a diaphragm mass almost twice that of the driver for correct system alignment.

The general requirements for a passive radiator may be summarized as acoustic mass and displacement volume roughly twice those of the driver, acoustic compliance equal to or greater than that of the driver, and suspension losses as low as possible.

7. MUTUAL COUPLING IN PASSIVE-RADIATOR SYSTEMS

Mutual coupling in passive-radiator systems takes the same form as for vented-box systems [5, sec. 8]. However, the effects are generally even smaller than for the vented-box system.

If the diameter of the passive radiator is equal to that of the driver, as is usual, the minimum center-to-center aperture spacing is greater than for the vented-box system, and the mutual coupling mass is therefore smaller. Furthermore, passive radiators are most often used in smaller loudspeaker systems which require relatively heavy driver cones to obtain extended low-frequency response. The mutual-coupling mass under these conditions represents only a tiny fraction of the total driver moving mass, giving quite negligible effects on both performance and measurement.

8. PARAMETER MEASUREMENT

Voice-Coil Impedance

The voice-coil impedance function of the passive-radiator system is given by Eq. (26). The steady-state magnitude $|Z_{VC}(j\omega)|$ of this function has the shape plotted in Fig. 17. This shape is exactly the same as that for the vented-box system [5, Fig. 20]. The plot has two maxima, at the frequencies labeled f_L and f_H. Between these maxima, there is a minimum at a frequency near f_B which is labeled f_M. At f_M the minimum impedance is slightly greater than R_E; the additional resistance is contributed by enclosure and passive-radiator losses and designated R_{BM}.

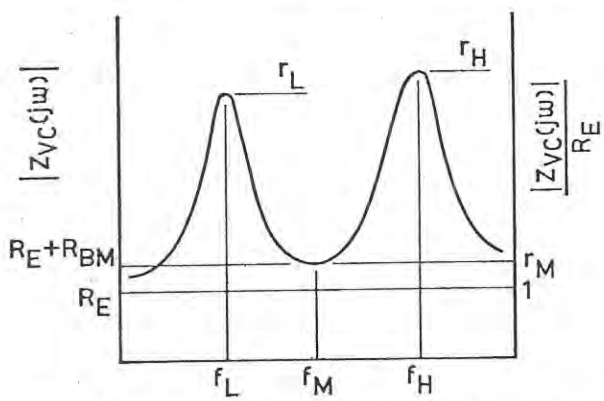

Fig. 17. Voice-coil impedance magnitude of passive-radiator loudspeaker system as a function of frequency.

Small-Signal Parameter Measurement

The measured impedance curve of a passive-radiator system conforms closely to the shape of Fig. 17. The impedance maximum at f_L is usually lower than that at f_H because of passive-radiator losses. As in the case of the vented-box system, the basic system parameters may be evaluated with satisfactory accuracy by ignoring enclosure and passive-radiator losses for initial calculations and then calculating the system losses using the approximate system data.

Ignoring enclosure and passive-radiator losses, and assuming that $f_M \doteq f_B$, Eq. (26) may be used to derive the following parameter-impedance-plot relationships:

$$\frac{\delta+1}{\alpha+\delta+1}=\frac{f_B^2\, f_{SB}^2}{f_L^2\, f_H^2} \tag{43}$$

$$\frac{\alpha\,\delta}{\alpha+\delta+1}=\frac{(f_H+f_B)\;(f_H-f_B)\;(f_B+f_L)\;(f_B-f_L)}{f_L^2\, f_H^2}. \tag{44}$$

These relationships do not give an immediate solution for any of the passive-radiator system parameters as do their counterparts for the vented-box system [5, eqs. (44) and (45)]. This is because only the same amount of information is available from the impedance curve while the system has the additional parameter δ to be evaluated.

However, it is relatively easy to evaluate α. If the passive radiator can be removed from the enclosure, it can be replaced temporarily by a vent. Then f_{SB} and α can be calculated as for a vented-box system from [5, eqs. (44) and (45)]. The passive-radiator aperture can also be blocked off and α evaluated as for a closed-box system from [8, eq. (48)]. Alternatively, the driver resonance frequency f_S may be measured and adjusted to correspond to the air-load mass applicable in the enclosure; then, using the passive-radiator system impedance-plot data,

$$\alpha=\frac{f_H^2+f_L^2-f_B^2}{f_{SB}^2}-1 \tag{45}$$

where Eq. (45) is derived directly from Eqs. (43) and (44).

With α and f_{SB} determined, δ may be found from either Eq. (43) or Eq. (44). A useful check for errors of measurement, calculation, or approximation is the computation of δ from both equations and comparison of the values obtained. Using the measured values of δ and f_B, f_P may be calculated from Eq. (21).

The remaining system parameters are measured in the manner described in [4, Appendix] and [5, sec. 6]. The value of Q_B computed from [5, eq. (49)] includes the effect of passive-radiator losses; assigning a value about 30–40% greater than this to Q_L gives a very satisfactory picture of the system response for evaluation purposes.

REFERENCES

[1] H. F. Olson, "Loud Speaker and Method of Propagating Sound," U.S. Patent No. 1,988,250, application Feb. 17, 1934; patented Jan. 15, 1935.

[2] H. F. Olson, J. Preston, and E. G. May, "Recent Developments in Direct-Radiator High-Fidelity Loudspeakers," *J. Audio Eng. Soc.*, vol. 2, pp. 219-227 (Oct. 1954).