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AN EFFICIENCY CONSTANT COMPARISON BETWEEN LOW-FREQUENCY HORNS AND DIRECT-RADIATORS

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Evaluation of the efficiency constant for exponential horns reveals that the horn is quite wasteful in its use of enclosed volume when compared to direct-radiator systems. The main advantage of horns lies in the realizibility of rather high efficiencies in the 10% to 40% range which is beyond the capabilities of most direct-radiators. Use of direct-radiators in arrays increases the low-frequency efficiency but not without a decrease of high-frequency bandwidth. The areas discussed in this paper are illustrated by comparative experimental measurements on three low-frequency systems: (1) A dual driver front-loaded folded horn, (2) A single driver direct-radiator vented-box system and (3) A four driver vented-box system consisting of a 2x2 array of the single driver system of (2).

INTRODUCTION: A number of authors recently have pointed out the inter-relationship of cabinet volume, low frequency cutoff, and efficiency for direct radiator systems [1]-[3]. These same relationships are found to apply in general to all forms of acoustic radiators whether direct-radiator, horn or some variation. The relationship is found to reduce to one which depends only on the radiated wavelength and the relative size of the radiator i.e. a radiator which is small compared to wavelength must inherently be inefficient [7] (just as in other radiators such as radio antennas).

The figure of merit of different forms of acoustic radiators may be evaluated by calculating the so called efficiency constant of R. H. Small's for each configuration [3], [4], [7]. Application of this concept to exponential horns and the resultant comparison with direct radiators yields some very interesting results.*

THEORY

Small indicates how the efficiency of any direct-radiator loudspeaker system can be shown in the following form [3]:

$$\eta_o = k_\eta \; f_3^{\,3} \; V_B \qquad (1)$$

where η_o = reference power available efficiency (usually defined for half-space radiation),

k_η = efficiency constant dependent on system type and on certain secondary system properties,

f_3 = frequency at which the system response is 3 dB below passband level, and

V_B = net internal box volume.

This equation represents the fundamental small-signal performance limitation of direct-radiator loudspeaker systems.

The author has wondered if the same relationship holds for horn type loudspeakers. Until a recent paper was given [9] a specific value for exponential horn V_B could not be determined because the mouth size of an exponential horn was rather indeterminate. Research in [9] shows that there is an optimum mouth size for a horn of specific cutoff to minimize reflected waves from the horn's mouth. The following derivations show that the exponential horn, strictly speaking, does not conform to relationship (1) but that valid comparisons can be made between horn and direct radiator systems once the horn's k_η is known for a specific combination of horn parameters.

*Since the first draft of this paper was written (October, 1973), Lampton [8] has made a cursory application of the figure of merit to exponential horns.

Eq. (1) may be solved for k_η yielding:

$$k_\eta = \frac{\eta_o}{f_3^{\,3}\ V_B} \qquad (2)$$

The efficiency constant k_η can of course be computed for any arbitrary system once η_o, f_3 and V_B are known.

The following equation which is derived in Appendix I shows how the exponential horn net internal box volume V_B and cutoff frequency f_C are related (for radiation into a half-space, optimum mouth size assumed):

$$V_B = \frac{(0.935)^2\ c^3}{6\pi\ f_C^{\,3}} \qquad (3)$$

where f_C = exponential horn cutoff frequency, and

c = speed of sound in air (=343 m/sec).

Equation (3) is independent of driver parameters and depends only on the physical parameters of the horn.

The horn's midband efficiency η_o which is independent of V_B and f_C depends only on driver parameters and the throat-diaphragm area ratio [10, p. 263, eq. (9.7)] . η_o is constrained to be $\leqslant$50% by definition of the power available efficiency.

Substitution of (3) into (2) yields:

$$k_\eta = \frac{16\pi^2\ \eta_o}{(0.935)^2\ c^3\ (f_3/f_C)^3} \qquad (4)$$

Experience has shown that a reasonable value for f_3/f_C for a finite, well designed exponential horn is roughly one. Evaluation of (4) in SI units gives:

$$k_\eta \approx 4.5\ \text{X}\ 10^{-6}\ \eta_o \qquad (5)$$

Typical horn efficiencies extend over the range of 5% to 50% ($0.05 \leqslant \eta_o \leqslant 0.5$) yielding efficiency constants from 2.24 X 10^{-7} to a maximum of 2.24 X 10^{-6}. Choosing an average efficiency of 30%, k_η for the exponential horn becomes 1.34 X 10^{-6}. These values of k_η can be compared to the vented box SI value of 3.9 C 10^{-6} [11] (4th order Butterworth) and to the closed-box k_η of 2.0 X 10^{-6} [12] (2nd order Chebyshev with 1.9 dB ripple). Table 1 illustrates a comparison between the direct-radiator systems and an exponential horn system all of which have the same efficiency (10%) and f_3 (40 Hz).

TABLE I - THEORETICAL LOUDSPEAKER SYSTEM COMPARISONS

(All systems have same efficiency and low end limit, half-space load, full-space values in parenthesis)

SYSTEM TYPE	CUTOFF FREQUENCY f_3, Hz	EFFICIENCY η_0, %	NET VOLUME V_B, m^3	EFFICIENCY CONSTANT K_η, $Hz^{-3}m^{-3}$
Exponential Horn With Optimum Mouth Size	40	10	3.49 (5.72)	4.5×10^{-7} (2.75×10^{-7})
Closed-Box, Direct-Radiator, Chebyshev 1.9 dB Ripple	40	10	0.78 (1.56)	2.0×10^{-6} (1.0×10^{-6})
Vented-Box, Direct-Radiator, Butterworth 4th order	40	10	0.40 (0.80)	3.9×10^{-6} (1.95×10^{-6})

The tabulation of V_B clearly shows the superiority of the vented system over both the closed-box and horn systems in its efficient use of internal volume. It must be pointed out that a 10% direct-radiator system borders on the upper limit of realizability while a horn system adjusted to this same efficiency is clearly realizable but quite mismatched considering the potential efficiency of 50% given the correct driver.

These comparisons show that a horn system must be adjusted for maximum efficiency (30 to 50%) in order for its use and added complexity to be worthwhile. If the horn's efficiency is allowed to drop into the region where a direct-radiator system could be synthesized to yield the same efficiency and response, the direct-radiator system would be preferable because of its smaller size. High efficiency direct-radiator systems may be synthesized by use of direct-radiators in multiple arrays where the efficiency increases roughly in direct proportion to the number of units in the array [13]. Research also shows that the high-frequency band-width decreases roughly as the square root of the number of units (the familiar gain-bandwidth tradeoff).

EXPERIMENT

The areas discussed in this paper are illustrated by comparitive experimental measurements made on three low-frequency systems: (1) A dual 12 inch driver 50 Hz cutoff front-loaded folded horn with a gross internal volume of 11.9 cu. ft., (2) A single 15 inch driver direct-radiator vented-box system with a gross volume of 3.2 cu. ft. and (3) A four 15 inch driver vented-box direct-radiator system consisting of a 2 x 2 array of the single driver system of (2) with a gross volume of 12.8 cu. ft. The multiple direct-radiator system of (3) has roughly the same internal volume as the horn of (1). The specific details of each of these systems along with a list of each systems driver parameters can be found in Appendix II.

The tests on each system included anechoic chamber measurements of near [14] and farfield sine-wave frequency response and a complete set of one-third octave bandwidth polar responses (both vertical and horizontal). The polar curves were analyzed to determine beamwidth (-6 dB) and directivity [15] versus frequency. The frequency response and directivity data were inturn used to derive a curve of efficiency versus frequency. (Efficiency = acoustic power output divided by nominal power input (E_{in}^2/R_{min}), see Appendix III).

Figures 1 to 3 show the result of these measurements and computations along with a curve of each systems input impedance magnitude versus frequency.

Figure 4 shows a comparison of the efficiencies of the three systems all plotted on the same graph. Figure 5 indicates the results of a comparitive analysis based on one-third octave bandwidth room curves taken with a real time spectrum analyzer (three mic-location average) on the horn and quad direct-radiator systems set up in EV's listening room (the input powers were adjusted to be equal based on R_{min}).

ANALYSIS

Figures 4 and 5 show that the efficiencies of the horn and quad direct-radiator are roughly equal in the 90 to 250 Hz range at some 8 to 12%. The four driver system is found to have a significantly lower cutoff frequency with about 6 to 8 dB more efficiency than the horn at 50 Hz. The quad system suffers in comparison to the horn in the 250 to 500 Hz band, however. Above 600 Hz, the direct-radiators efficiency exceeds the folded horn's mainly because of the low-pass filtering action of the folded horns transmission path.

A comparison of the efficiencies between the single and quad direct-radiators shows that indeed the efficiency jumped up by a factor of 3 to 4 in the 60 to 200 Hz band. Above 300 Hz, however, the efficiency increase factor averaged only 1.6 to 2 (2 to 3 dB). Presumably closer center to center spacing of the four speakers would increase the effective range of efficiency increase. The quad system actually exhibited a decrease of f_3 by somewhat less than about one-third octave presumably because of a slight decrease in box resonance frequency and the different air mass loading that the array provides.

Table II displays the computations of efficiency constant for the three experimental systems.

TABLE 2 - EXPERIMENTAL LOUDSPEAKER SYSTEM COMPARISONS

(Derived from full-space anechoic chamber measurements)

SYSTEM NUMBER	SYSTEM DISC.	CUTOFF FREQUENCY f_3, Hz	EFFICIENCY η_0, %	NET VOLUME V_B, m^3	EFFICIENCY CONSTANT K_η, $Hz^{-3}m^{-3}$
1	Exponential Horn Dual Driver, f_c=50 Hz	75	12.5	0.34	8.7×10^{-7}
2	Vented-Box Direct-Radiator, Single Driver	61	3.2	0.091	1.5×10^{-6}
3	Vented-Box Direct-Radiator, Four Drivers	51	9.0	0.36	1.88×10^{-6}

FOLDED-HORN, TWO DRIVERS

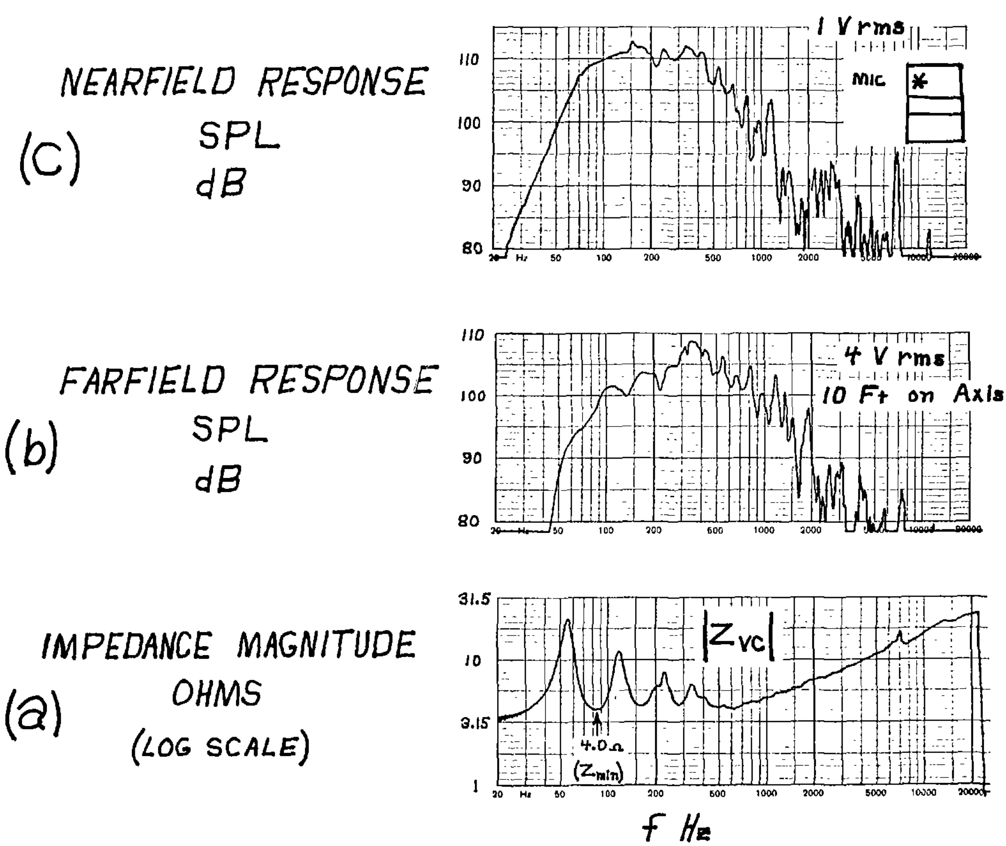

Fig. 1. Display of experimental measurements on the 11.9 cu. ft. dual-driver 50 Hz cutoff folded exponential horn system No. 1 described in Appendix II. The curves show: (a). input impedance magnitude versus frequency (Zmin = 4.0 ohms), (b). anechoic chamber farfield sinewave frequency response taken with 4 volts RMS applied with test mic 10 ft. on box axis, (c). nearfield sinewave frequency response with 1 volt RMS applied with test mic located in the horns mouth flush with the front edges of the horn,

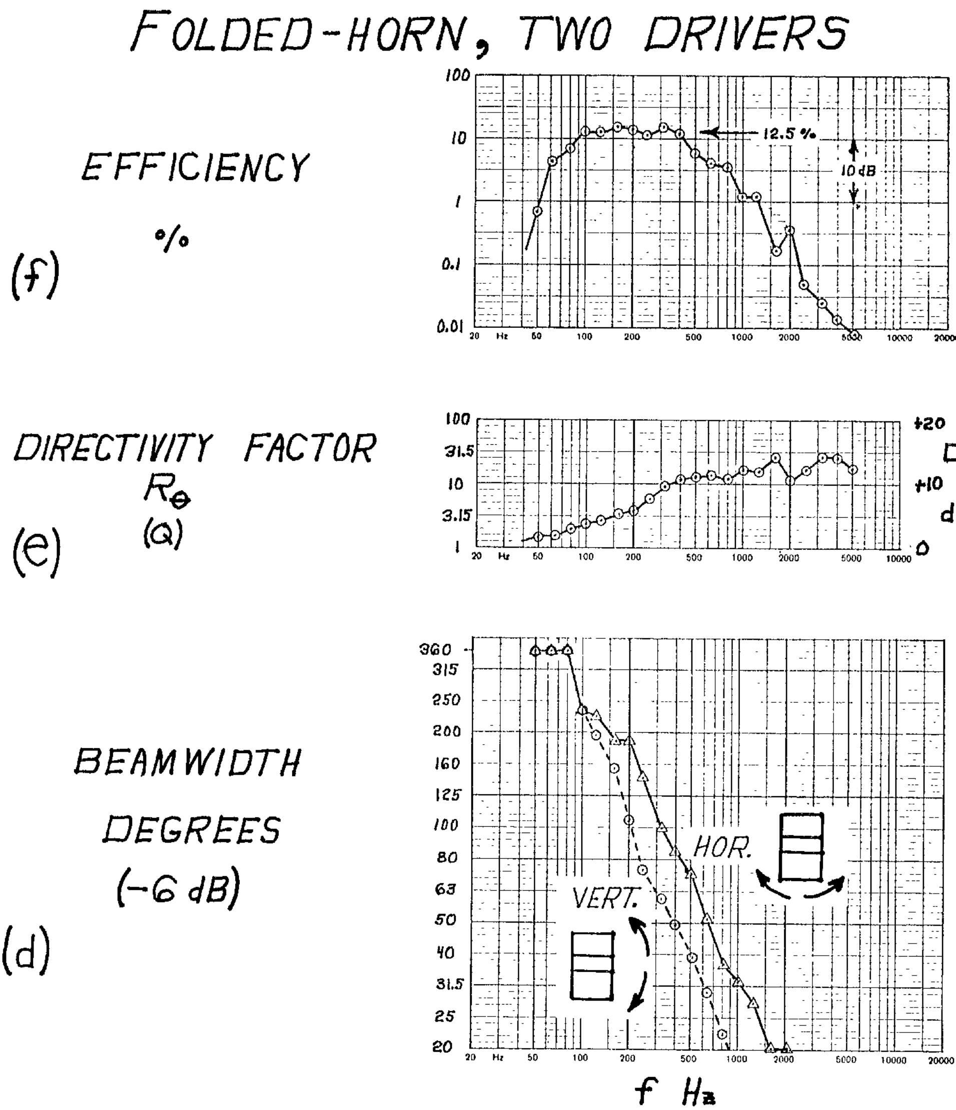

Fig. 1. Cont. System No. 1 Data. (d). plot of the 6 dB down beamwidth of the system for both horizontal and vertical orientations taken from the one-third octave bandwidth noise polar responses, (e). directivity factor R_θ (Q) and directivity index D_i versus frequency and (f). the systems nominal power transfer efficiency versus frequency plotted on a log scale (10 log η_0, see Appendix III, derived from the far-field response (b) and the directivity data (e)). Note the close correspondance between the nearfield response (c) and the plot of efficiency versus frequency (f).

DIRECT-RADIATOR, SINGLE DRIVER

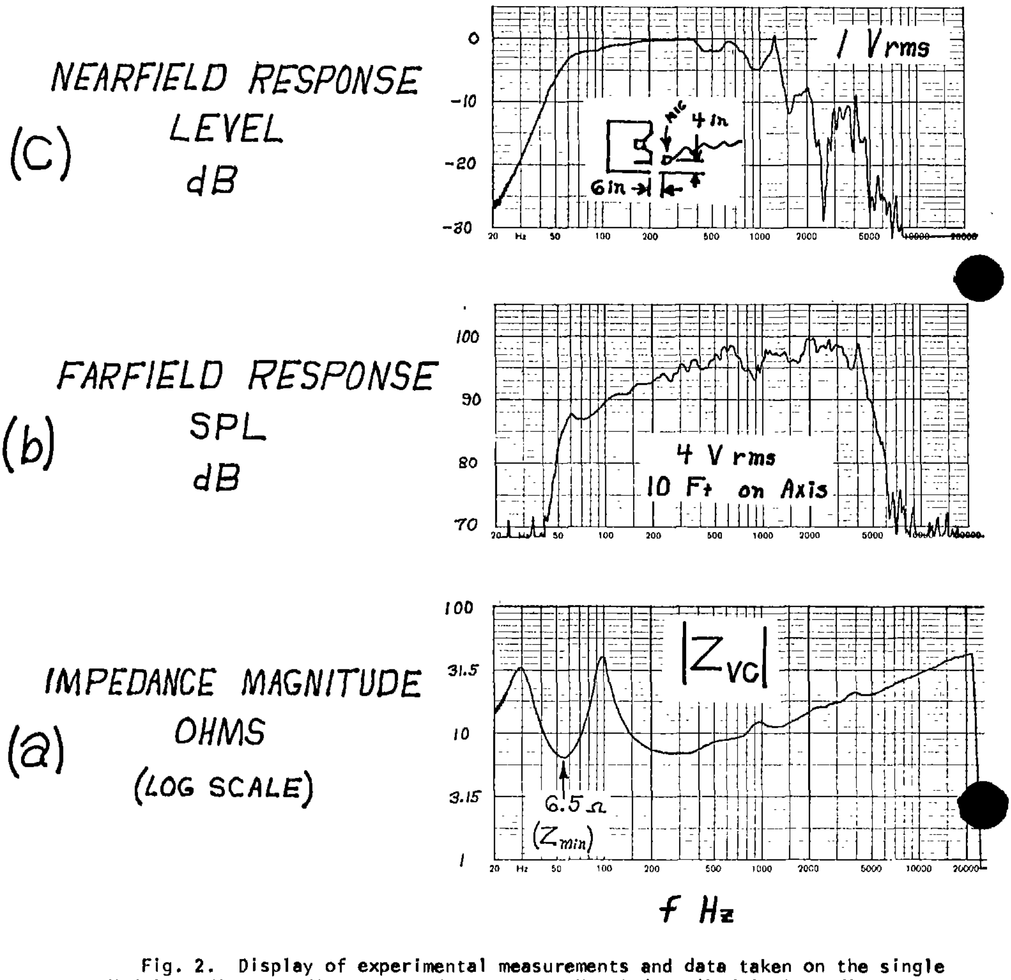

Fig. 2. Display of experimental measurements and data taken on the single 15" driver direct-radiator vented-box system No. 2 described in Appendix II. Measurements displayed include: (a). magnitude of driving point impedance versus frequency (Zmin = 6.5 ohms), (b). anechoic chamber far-field frequency response taken with 4 volts RMS applied with test mic 10 ft. on box axis, (c). nearfield sinewave frequency response with 1 volt RMS applied with test mic centered horizontally on box 6 inches away and 4 inches above bottom of box,

DIRECT-RADIATOR, SINGLE DRIVER

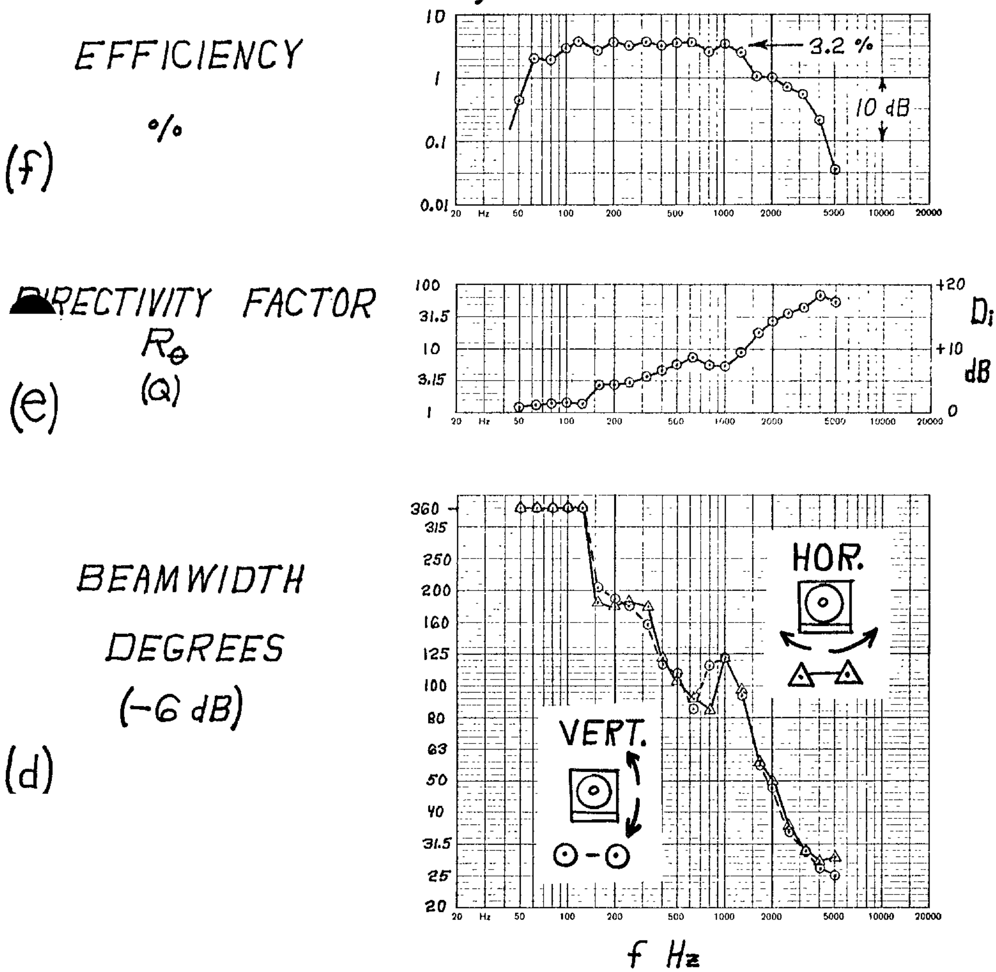

Fig. 2. Cont. System No. 2 **Data.** (d). plot of the 6 dB down beamwidth of the system for both horizontal and vertical orientations taken from the one-third octave bandwidth noise polar responses, (e). directivity factor R_θ (Q) and directivity index D_i versus frequency and (f). the systems nominal power transfer efficiency versus frequency plotted on a log scale (10 log η_o, see Appendix III, derived from the far-field response (b) and the directivity data (e)). Note the close correspondance between the nearfield response (c) and the plot of efficiency versus frequency (f) below 400 Hz.

DIRECT-RADIATOR, FOUR DRIVERS

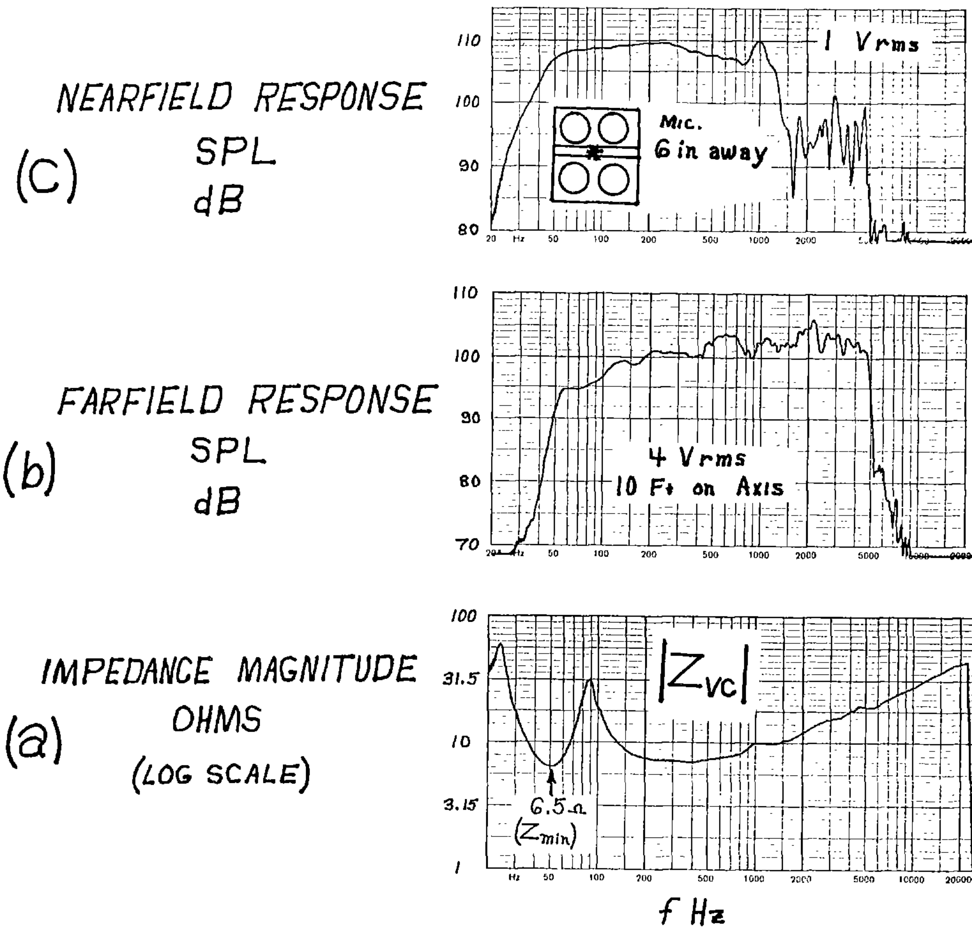

Fig. 3. Display of experimental measurements on the four 15" driver direct-radiator vented-box system No. 3 described in Appendix II. Note that this system is essentially a 4 X scale up of system No. 2. Measurements include: (a). input impedance magnitude versus frequency (Zmin = 6.5 ohms), (b). anechoic chamber far-field frequency response taken with 4 volts RMS applied with test mic 10 ft. on box axis, (c). nearfield sinewave frequency response with 1 volt RMS applied with test mic centered on box axis 6 inches away,

DIRECT-RADIATOR, FOUR DRIVERS

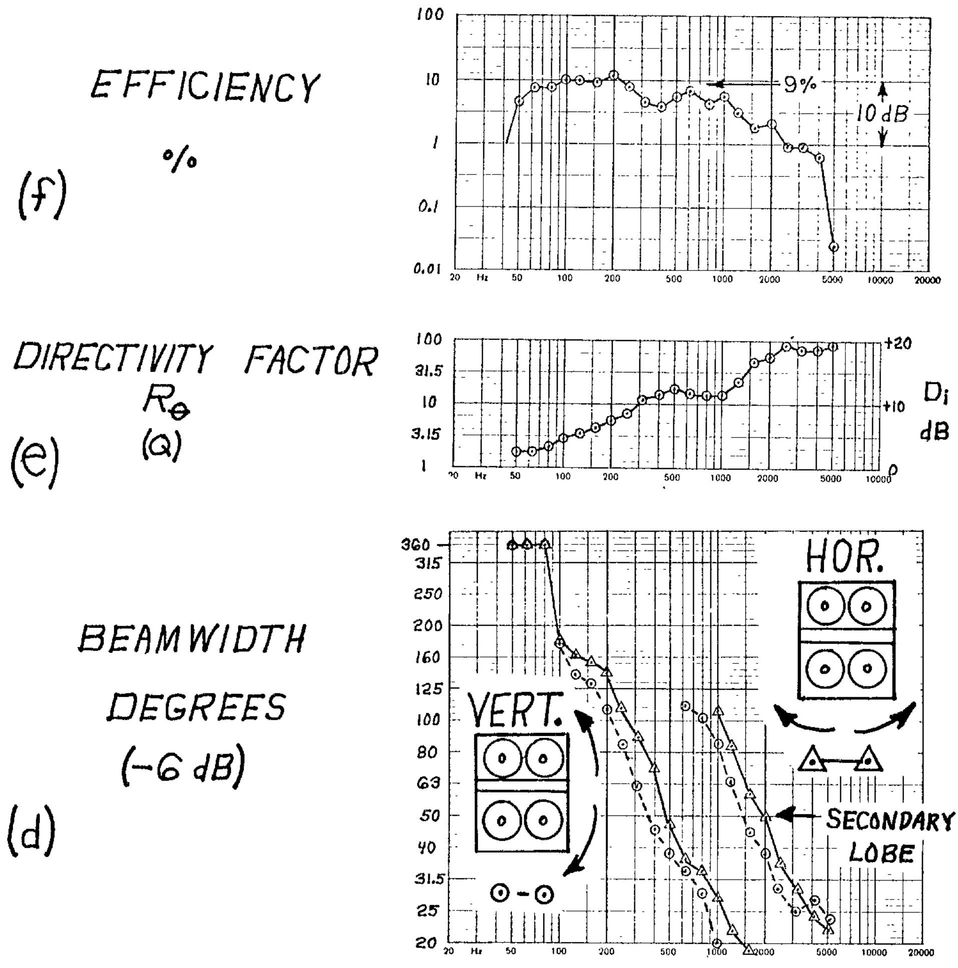

Fig. 3. Cont. System No. 3 Data. (d). plot of the 6 dB down beamwidth of the system for both horizontal and vertical orientations taken from the one-third octave bandwidth noise polar responses, (e). directivity factor R_θ (Q) and directivity index D_i versus frequency and (f). the systems nominal power transfer efficiency versus frequency plotted on a log scale. (10 log η_o, see Appendix III, derived from the far-field response (b) and the directivity data (e)).

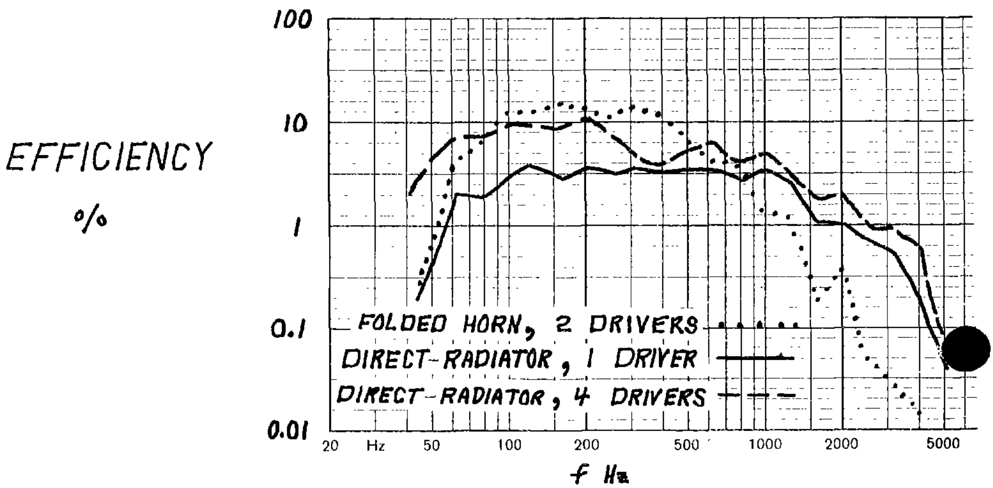

Fig. 4. Comparative display of the efficiency data on the three systems of Figs. 1 to 3 which are described in Appendix II.

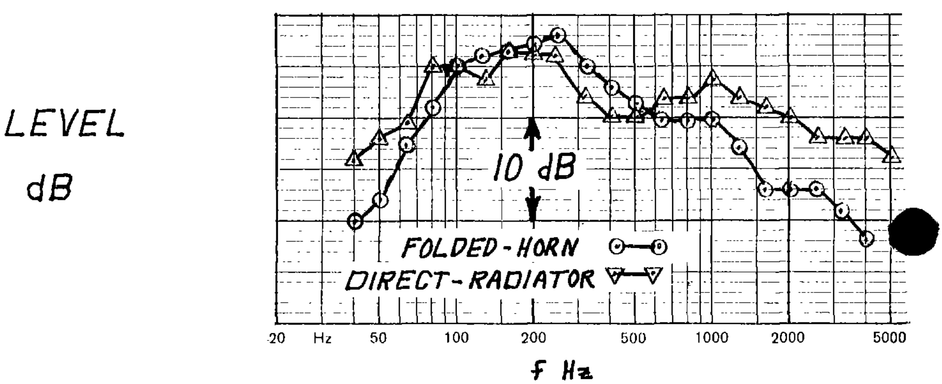

Fig. 5. Display of comparative one-third-octave real-time spectrum analyzer curves on the horn (system 1) and the four driver direct-radiator (system 3) (described in Appendix II) taken in EV's listening room (5,270 cu. ft., 25.7 ft. X 23.3 ft. X 8.8 ft. with about a 0.7 second reverberation time). The systems were placed side by side on the floor, with their backs against the wall, centered along the 23.3 ft. wall. A three microphone space average was taken with the input voltages to the systems adjusted for equal nominal electrical input powers (horns level reduced by 10 log 4/6.5 = -2.2 dB).

It must be noted that these computations are based on anechoic chamber measurements taken in full-space (4π steradian solid angle) and must be compared to the full-space values listed in Table I.

The experimental efficiency constant numbers for the vented boxes are just slightly less than the theoretical values. The horn's value, however, is roughly twice the predicted value but still smaller than the vented box by a factor of 2.3. The measured experimental horn differed from the analyzed theoretical horn in that the mouth area was roughly 3/8 the optimum full-space value required for the 50 Hz cutoff frequency (this horn was initially designed to operate in 1/4 to 1/2 space solid angles).* It may be that the horns efficiency constant is maximized by having a flare cutoff frequency f_C substantially lower than the desired f_3 and operating with a less than optimum mouth size for the chosen f_C. Research needs to be done in this area.

CONCLUSIONS

The theoretical and experimental analysis shows that direct-radiators can in some cases compete directly with horns in the important area of compact-size low-cutoff moderate-efficiency transducers. Low frequency horns are best suited for those situations where absolute highest efficiency is required and the resultant overly large size can be tolerated. In most cases the horn must be folded to yield an acceptable size format which means a complex hard to construct enclosure.

For the same low-end cutoff, a multiple direct-radiator system could be synthesized to yield roughly the same response in a much smaller enclosure. The basic tradeoffs boil down to:

I. Horn

a. Highest efficiency
b. Large complex enclosure
c. Small number of drivers

II. Direct-Radiator Vented-Box (Multiple Drivers)

a. Moderate efficiency
b. Small moderately simple enclosure
c. Large number of drivers
d. High power handling capacity (because of multiple drivers)

This roughly means that if one has a lot of space, not much money to spend on drivers and amplifiers, and lots of cheap labor—build a horn. If labor is

*The experimental measurements on this horn show just how real world horns act in some cases even though they were designed according to the best available theory. In this case the diaphragm-throat area is roughly correct for a 50% nominal midband efficiency. However, the much less than optimum mouth area coupled with the effects of the folded configuration, front cavity volume and non-negligible driver mass contribute to the attainment of a maximum efficiency of only 12.5%.

not cheap, you don't have much space, and you can afford drivers and amp power — build a direct-radiator.

Note that in some situations the multiple direct-radiator vented-box system ends up having more maximum acoustic output power (see [16], Appendix I) than the horn system. For the specific example experimentally analyzed in this paper, the maximum output of the four driver direct-radiator (at 400 watts input) exceeds the horn (at 200 watts input) for all frequencies except between 300 to 400 Hz. At 50 Hz the direct-radiators maximum output is a whopping 10 dB higher than the horn (at roughly the same distortion level).

APPENDIX I

HORN VOLUME DERIVATION

The equation which gives the area as a function of axial distance for the exponential horn appears as [10, p. 269]:

$$S = S_T \, e^{mX} \tag{6}$$

where S = area at distance X,

S_T = throat area (= S (0)),

m = horn flare constant ($=4\pi f_C/c$),

f_C = cutoff frequency, and

c = velocity of sound in air (=343 m/s at 20°C).

The finite horn is assumed to terminate at X = L = horn length, where $S = S_M$ = mouth area.

Neglecting back cavity volume and assuming $S_M \gg S_T$ an integration of (6) yields:

$$V_B \approx \frac{S_M}{m} = \frac{\pi a_M^2}{m} \tag{7}$$

where a_M = radius of mouth of circular crossection.

The optimized mouth radius for radiation into a half-space is found to be [9, Table I]:

$$a_M = 0.935/k_C \qquad (= 1.2/k_C \text{ for full-space}) \tag{8}$$

where $k_C = m/2$.

Substituting (8) into (7) and noting that $m = 4\pi f_C/c$ yields:

$$V_B = \frac{(0.935)^2 \, c^3}{16\pi^2 \, f_C^3} \qquad (= 2.23 \times 10^5/f_C^3 \text{ SI units}). \tag{9}$$

APPENDIX II

DESCRIPTION OF MEASURED SYSTEMS

The three measured low frequency systems consisted of: (1) A dual driver folded-horn, (2) A single-driver direct-radiator vented-box and (3) A four-driver direct-radiator vented box. They are described as follows:

1. Folded Horn System:

This system is a 50 Hz cutoff hyperbolic-exponential (T=0.6) folded horn designed for use with two 12" diameter high-power musical-market drivers. This system is fully described in a paper by R. Newman [17] and marketed by Electro-Voice as the TL5050. Figure 6 shows a photograph and an internal line drawing of the system. The characteristics of the system can be summarized as:

System type:	Folded Horn
Flare:	Hyperbolic-exponential (T=0.6)
Cutoff Frequency f_C:	50 Hz
Throat area:	81 sq. inches
Path length:	53 inches
Mouth area:	769 sq. in. (1040 sq. in. box frontal area)
Front cavity volume:	130 cu. in.
Rear cavity volume:	1.7 cu. ft.
Driver type:	12 inch direct-radiator (2 each)

Further information is shown in Table III.

Driver Parameters:

The parameters of the 12 inch driver (an EV EVM12L) used in the horn are listed as follows (all free-air, unenclosed):

Physical Parameters:

Effective moving mass	M_{MS}	=	31.4 g (includes air mass load)
Suspension compliance	C_{MS}	=	4.0×10^{-4} m/N
	BL	=	15.2 Tm
	R_E	=	5.6 Ω
Mechanical Q	(Q_{MS})	=	9.5

Thiele/Small Parameters:

f_S = 45 Hz
Q_{ES} = 0.215
Q_{MS} = 9.5
Q_{TS} = 0.210
V_{AS} = 5.06 cu. ft.

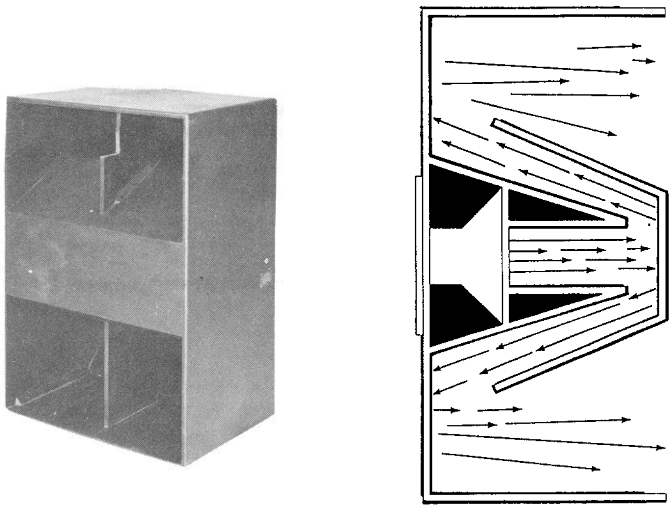

Fig. 6. Photograph and line drawing of System No. 1, the dual 12" driver folded-horn system described in Appendix II.

η_o	=	5.8% (half-space)
X_{max}	=	0.13 in.
V_D	=	10.1 in.^{3}
Effective cone area S_D	=	78 in.^{2}
$P_{E(max)}$	=	100 Watts
R_{min}	=	6.4 Ω
Advertised Dia.	=	12 in.
Effective Dia.	=	10 in.

All parameters are as defined by Small[4].

2. Single-Driver Direct-Radiator Vented-Box System

This is a vented-box system designed around a 15 inch diameter high-power musical market driver with a gross internal volume of 3.2 cu. ft. Figure 7 shows a photograph and line drawing of the unit.

The System Characteristics are (an EV TL606):

System type: Vented-Box Direct-Radiator
Alignment type: Quasi-Butterworth Third-Order (QB_3)

Further information is contained in Table III.

Driver Parameters:

The parameters of the 15 inch driver (an EV EVM15L) are listed as follows (all free-air unenclosed):

Physical Parameters:

M_{MS}	=	53.5g (with air mass load)
C_{MS}	=	1.75 X 10^{-4} m/N
$B\ell$	=	15.5 Tm
R_E	=	5.6 Ω
Mechanical Q (Q_{MS})	=	5.3

Thiele/Small Parameters:

f_S	=	52 Hz
Q_{ES}	=	0.41
Q_{MS}	=	5.3
Q_{TS}	=	0.38
V_{AS}	=	6.47 ft.^{3}
η_o	=	6.0% (half-space)
X_{max}	=	0.13 in.
V_D	=	17.3 in.^{3}

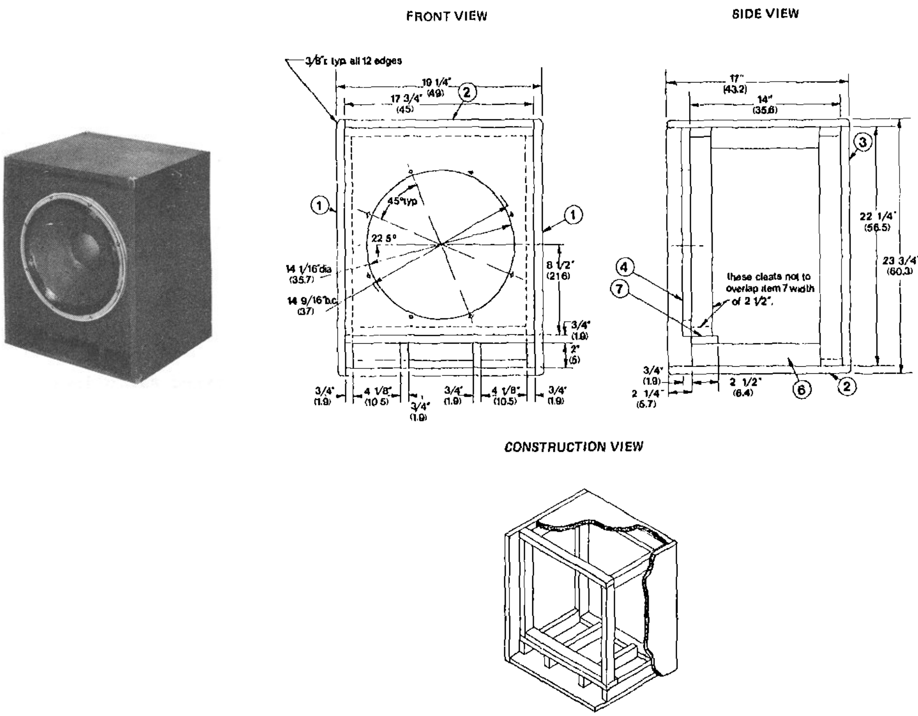

Fig. 7. Photograph and line drawing of System No. 2, the single 15" driver direct-radiator vented-box system described in Appendix II.

S_D	=	133 in.^{2}
$P_{E(max)}$	=	100 Watts
R_{min}	=	6.4 Ω
Advertised Dia.	=	15 in.
Effective Dia.	=	13 in.

3. Four-Driver Direct-Radiator Vented-Box System

This system is essentially composed of four of the systems described in 2. but combined in one box i.e. 4 drivers, 4 x volume, 4 x vent area and same vent length. Figure 8 shows a photograph and line drawing of this system. The system characteristics are (an EV TL606Q):

System type: Vented-Box Direct-Radiator
Alignment type: Quasi Butterworth Third-Order (QB_3)

Further information is shown in Table III.

Driver Parameters:

The parameters are the same as those listed for system number 2.

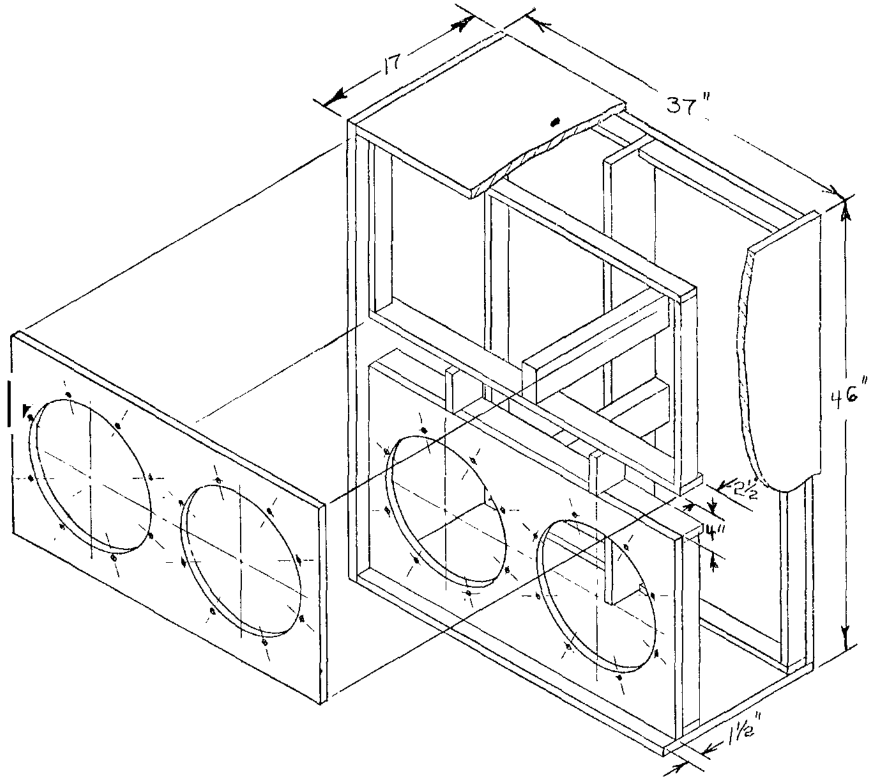

Fig. 8. Photograph and line drawing of System No. 3, the four 15" driver direct-radiator vented-box system described in Appendix II.

TABLE III - SYSTEM SPECIFICATIONS

	FOLDED-HORN	VENTED DIRECT-RADIATOR	
SYSTEM NUMBER	1	2	3
LOW FREQUENCY 3 dB DOWN POINT	70 Hz	63 Hz	55 Hz
USABLE LOWER LIMIT FREQUENCY[1]	55 Hz	45 Hz	42 Hz
USABLE UPPER LIMIT FREQUENCY[2]	850 Hz	1300 Hz	600 Hz
EFFICIENCY (Half-space)	20%	6%	18%
POWER HANDLING CAPACITY (continuous thermal limit)	200 W	100 W	400 W
MAXIMUM MIDBAND ACOUSTIC OUTPUT POWER	40 W	6 W	72 W
MAXIMUM SPL AT 10 FEET, FULL POWER (Avg. from 100 to 800 Hz)	121.5 dB	110 dB	123 dB
SPL AT 10 FEET, 1 WATT INPUT (Avg. from 100 to 800 Hz)	98.5 dB	90 dB	97 dB
BEAMWIDTH (−6 dB)			
400 Hz (Horizontal)	85°	124°	71°
800 Hz (Horizontal)	39°	87°	34°
400 Hz (Vertical)	50°	118°	47°
800 Hz (Vertical)	23°	115°	29°
BOX RESONANCE FREQUENCY			
Normal	Horn	55 Hz	53 Hz
Step-down	Loaded	40 Hz	40 Hz
DRIVER			
Type	EVM 12L	EVM 15L	EVM 15L
Diameter	12 in.	15 in.	15 in.
Quantity	2	1	4
IMPEDANCE			
Nominal	5 ohms	8 ohms	8 ohms
Minimum	4.0 ohms	6.5 ohms	6.4 ohms
BOX PHYSICAL CHARACTERISTICS			
Gross Internal Volume	11.9 cu. ft.	3.2 cu. ft.	12.8 cu. ft.
External Height	40.75 in.	23.75 in.	46 in.
External Width	27.75 in.	19.25 in.	37 in.
External Depth	21.5 in.	17.0 in.	17 in.
Net Weight	170 lb	54 lb	200 lb

1. System can generate 8 acoustic watts or more down to this frequency (1/2 acoustic watt for system number 2).

2. System is reasonably flat and exhibits a beamwidth no less than 40° up to this frequency (80° for system number 2).

APPENDIX III

EFFICIENCY COMPUTATIONS

The method used to compute efficiencies in this paper corresponds to that used by Small [18] and is defined as the acoustic power output divided by the nominal electrical input power.

The nominal electrical input power is defined here as the power delivered by the source into a resistor having the same value as the minimum impedance the system attains in its rated pass band ($P_{in}=E_{in}^2/R_{min}$). This differs somewhat from Small's definition in that the power is developed in the measured system minimum impedance rather than the rated driver DC voice-coil resistance (which is somewhat lower). This slight modification in definition allows the horn and direct-radiator transducers to be compared more on an equal footing. Other definitions of input power could have been used such as a method based on impedance averaging in the system's pass band (this would have roughly doubled the efficiency of all the systems).

The acoustic power output of the system is calculated knowing the on-axis sound pressure level SPL (re 20μPa), the directivity factor R_θ (Q), measuring distance r and certain physical constants:

$$P_o = \frac{4\pi\ r^2}{\rho_o\, c\ R_\theta}\, p_{Frms}^{\ 2} = \left(\frac{16\pi}{\rho_o c} \times 10^{-10}\right) \frac{r^2}{R_\theta}\, 10^{\frac{SPL}{10}} \qquad (10)$$

where ρ_o = density of air (≈1.21 kg/m^3 at 20^{o}C), and
c = velocity of sound in air (≈343 m/s).

For the specific case of r = 3.05m (10 ft) and E_{in} = 4V rms the efficiency in percent can be written as:

$$\eta \approx 7.0 \times 10^{-10} \cdot \frac{R_{min}}{R_\theta} \cdot 10^{\frac{SPL}{10}} \qquad (11)$$

where η = nominal power transfer ratio or efficiency in % for the specific case of E_{in} = 4V rms and a free-field (full-space) measuring distance of 10 ft or 3.05 meters.

R_{min} = resistance equal to minimum impedance systems attains in its passband in ohms,

R_θ = directivity factor of source derived from vertical and horizontal polars, and

SPL = sound pressure level re 2×10^{-5} μ P_a.

Eq (11) was used for all efficiency calculations in this paper.

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