HORN LOADING REVISITED

by R.M. Harris, B.Sc.

Horn loudspeakers provide good low-frequency performance and high efficiency, but are impractical for many uses because they are very large. However, R.M. Harris describes a design that is suitable for use in an ordinary living room.

The weakest link in any hi-fi system is undoubtedly the acoustical transducer or loudspeaker. Sound recording technology is reaching new levels of precision and noise reduction with digital recording and compact discs. Amplifiers can be perfected to almost any degree, if the price is right, with hitherto unheard of purity in terms of harmonic distortion and intermodulation products. In other words, it is possible to reproduce sound faithfully from DC to RF, but only in terms of **electrical signals**. For, while electronic technology has passed from thermionic valves through discrete transistors to integrated circuits, the loudspeaker has hardly changed in its essentials since its invention in 1925*.

Since middle- and high-frequency propagation is characterized by small-amplitude sound waves, which eases most engineering problems, this article is confined to the problem of obtaining good, clean low-frequency sound reproduction.

Two physical principles account for most of the engineering difficulties at low frequencies. The first is that for a plane propagating sound wave the pressure, p, and particle velocity, u, are related in terms of the specific acoustic impedance of the medium (air):

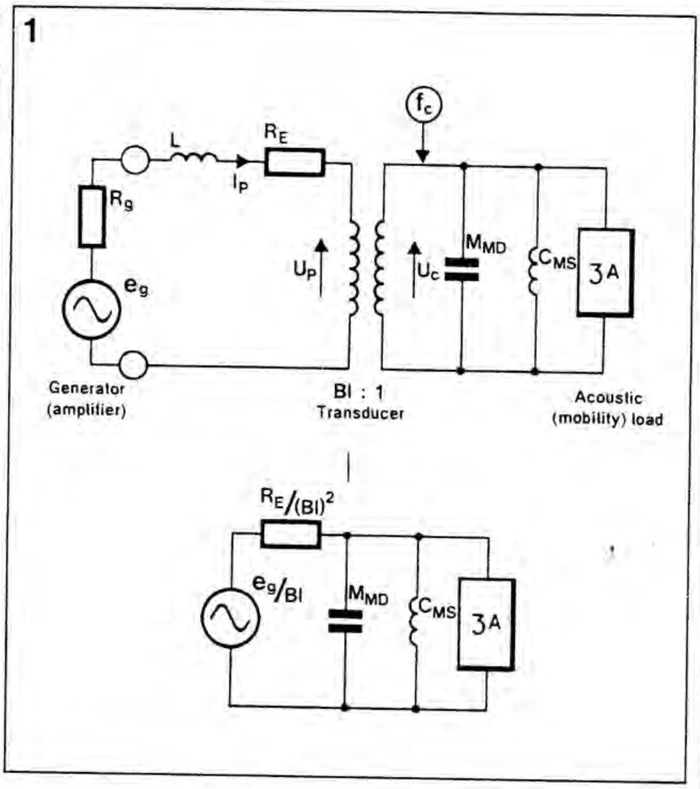
$$p/u = Z_o \quad [1]$$

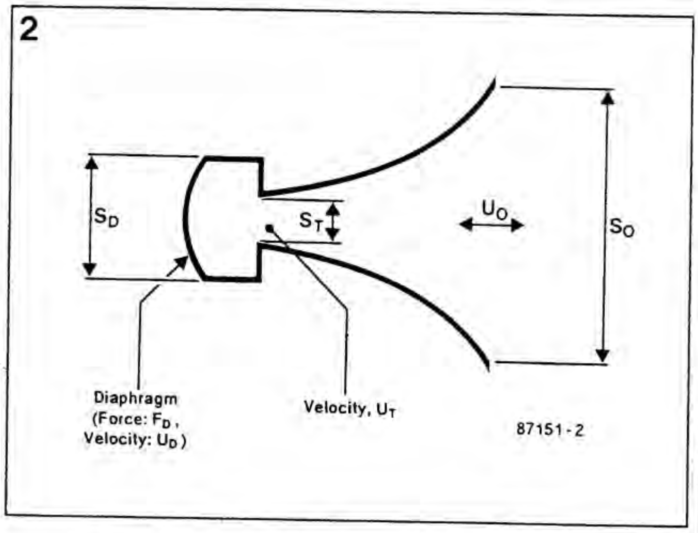
where

$$Z_o = \rho_o c \quad [2]$$

in which ρ_o is the density of the medium and c is the velocity of sound in the medium.

At a point in space, the instantaneous particle displacement, y, of a sinusoidal sound wave of amplitude a and frequency f Hz, is given by





$$y = a\sin 2\pi ft \quad [3]$$

and the instantaneous velocity is .

$$u_t = dy/dt = 2\pi af\cos 2\pi ft \quad [4]$$

and the maximum velocity is

$$u = 2\pi fa \quad [5]$$

For a given sound pressure level—SPL—of p, the amplitude is given by

$$a = u/2\pi f \quad [6]$$

$$= Z_o p/2\pi f \quad [7]$$

Both [6] and [7] simply state that the amplitude increases as the frequency decreases.

The second principle is that at low frequencies the wavelength in air is much larger than the dimensions of the sound source. This results in the radiation of spherical wave fronts from what is, in effect, a point source. The specific acoustic impedance, Z_s, is not the same for diverging waves, as given by the general formula

$$Z_s = \frac{p}{u} =$$

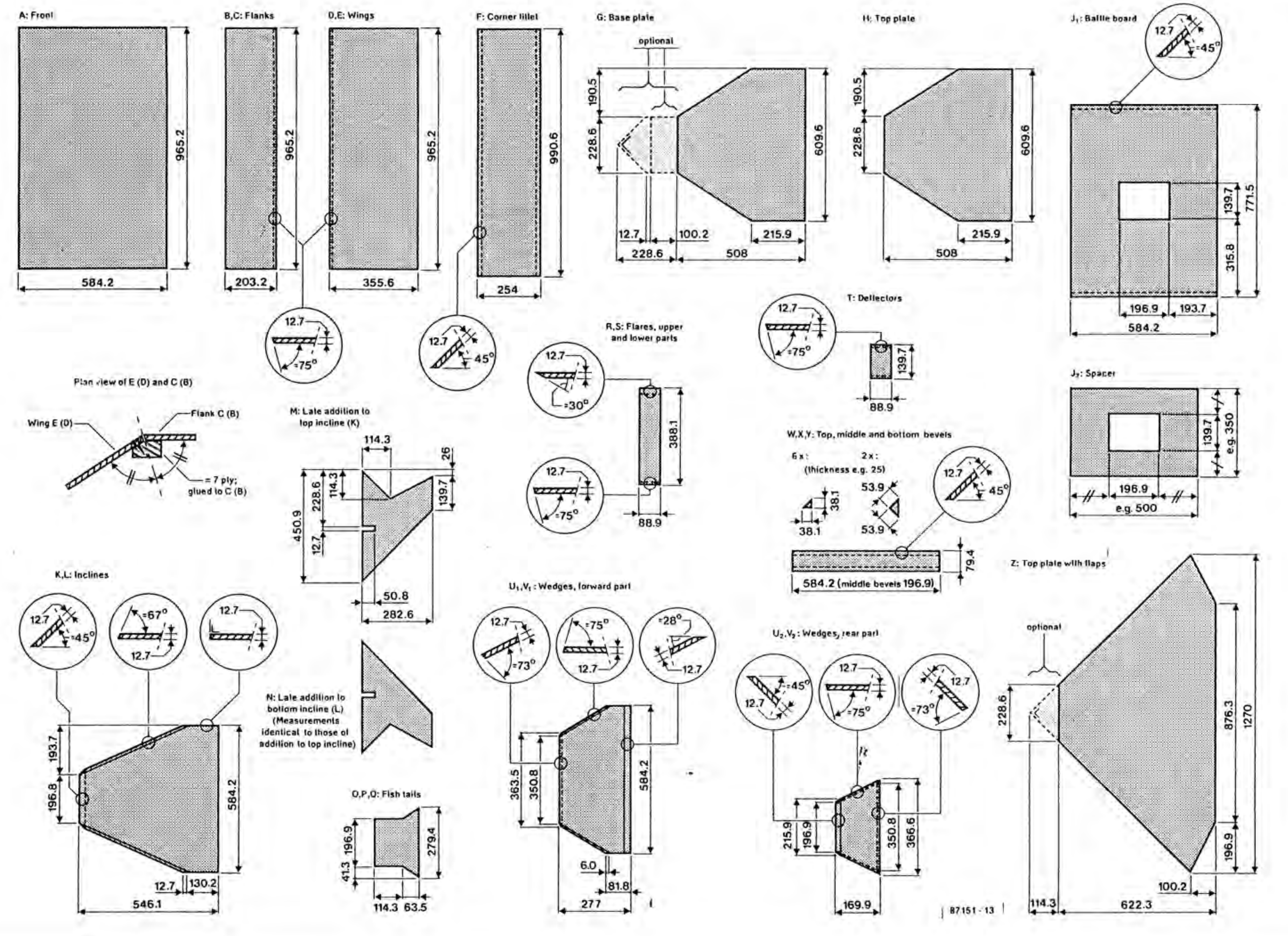
$$\frac{\rho_o ckr}{\sqrt{1+k^2r^2}}\ 90° - \arctan kr \quad [8]$$

where r is the radial distance from the point source, and $k = 2\pi/\lambda$.

When $kr >> 1$ ($r >> \lambda/2\pi$), the magnitude of Z_s approaches the value for a plane wave ($\rho_o c$) with u in phase with p. For $k2$ = 1, Z_s goes to zero and u tends to be in phase quadrature with p. The real part of Z_s (which is generally complex) falls to zero as r^2, which means that the mismatch between the transducer and the medium degrades in direct proportion to the square of the wavelength.

The corollary is that the speaker cone has to execute larger movements to maintain a constant SPL at lower frequencies if Z_s remains constant. If, however, Z_s (and in particular the real part of Z_s) falls at lower frequencies, the situation is exacerbated. So, all effects included, the cone amplitude needs to vary as λ^3 or as f^{-3}.

The foregoing general analysis clinches the central problem of designing for efficient bass reproduction. Large amplitudes are deprecated for moving-coil driver units: they introduce not only mechanical difficulties, but also distortion. It has been stated (Ref. 1) that any movement of the cone entails some distortion: the more it moves, 13


A: Front
B,C: Flanks
D,E: Wings
F: Corner fillet
G: Base plate
optional
H: Top plate
J1: Baffle board
J2: Spacer
T: Deflectors
R,S: Flares, upper and lower parts
Plan view of E (D) and C (B)
Wing E (D)
Flank C (B)
≈ 7 ply; glued to C (B)
M: Late addition to top incline (K)
W,X,Y: Top, middle and bottom bevels
6 x:
2 x:
(thickness e.g. 25)
K,L: Inclines
N: Late addition to bottom incline (L) (Measurements identical to those of addition to top incline)
O,P,Q: Fish tails
U1,V1: Wedges, forward part
U2,V2: Wedges, rear part
Z: Top plate with flaps
584.2 (middle bevels 196.9)
e.g. 350
e.g. 500
965.2
584.2
203.2
355.6
990.6
254
190.5
228.6
609.6
12.7
100.2
215.9
508
771.5
139.7
315.8
196.9
193.7
88.9
388.1
114.3
26
450.9
50.8
282.6
53.9
38.1
79.4
≈75°
45°
≈30°
≈67°
≈45°
≈73°
≈28°
196.8
546.1
130.2
41.3
63.5
279.4
363.5
350.8
6.0
81.8
277
366.6
169.9
876.3
1270
622.3
87151-13

the whole characteristic had shifted fortuitously downwards in frequency, so justifying his claimed "smooth response from 40 Hz to 400 Hz". A crossover is indicated at 400 Hz to a conventional mid-range and tweeter unit: both could be horn-loaded, of course.

Construction

Before tackling this daunting task, the author made two scale models, a 1:8 in cardboard, followed by a 1:3.6 in 3-ply. The working model exhibited a roll-off below 180 Hz, which scaled down to 50 Hz for the full-sized unit. The intrinsically self-bracing structure indicated relatively thin panelling. For cheapness (but not light weight!), half-inch, high-density chip board was used, costing less than £9.00 in all. The truss and some fillets were made in marine 9-ply, while blocking, corner fillets, air seals, and so on were made out of oddments of hardwood and deal.

Joints were glued and screwed except for the access door—one large side panel—which was secured with 14 wood screws. This was necessary for fitting (retrospectively) the CG12 and for possible maintenance. With the door off, the top and bottom fillets would have been precariously unsupported, were it not for centre-line fins, which were the author's distinctive (cf. Jecklin, Ref. 6) modification to Klipsch's constructional designs—see Fig. 11, 12, and 13, as well as the accompanying photographs.

References

1. *Paul W. Klipsch, Wireless World*, Feb. 1970

2. Leo L. Beranek, *Acoustics*.

3. Abraham B. Cohen, *Hi-fi Loudspeakers and Enclosures*, 2nd Ed., Newnes Butterworth, London, 1975

4. P.W. Klipsch, *A Low Frequency Horn of Small Dimensions, Journal of the Acoustic Society of America, Vol 13*, pp 137-144 (1941)

5. Rocard 1933 cited in Moire's *Acoustics*

6. *J. Jecklin, Wireless World*, Febr. 1969

* C.W. Rice and E.W. Kellogg

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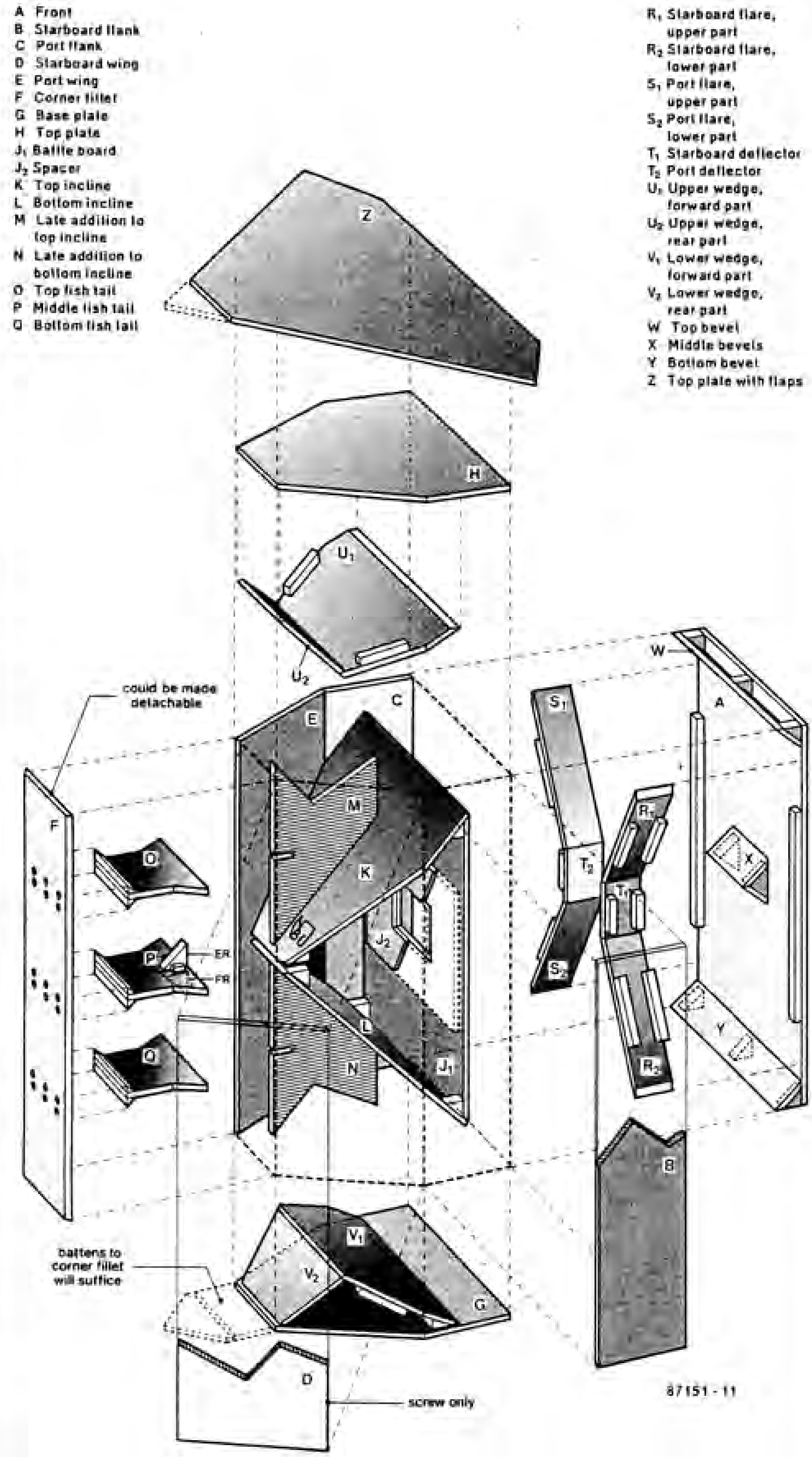

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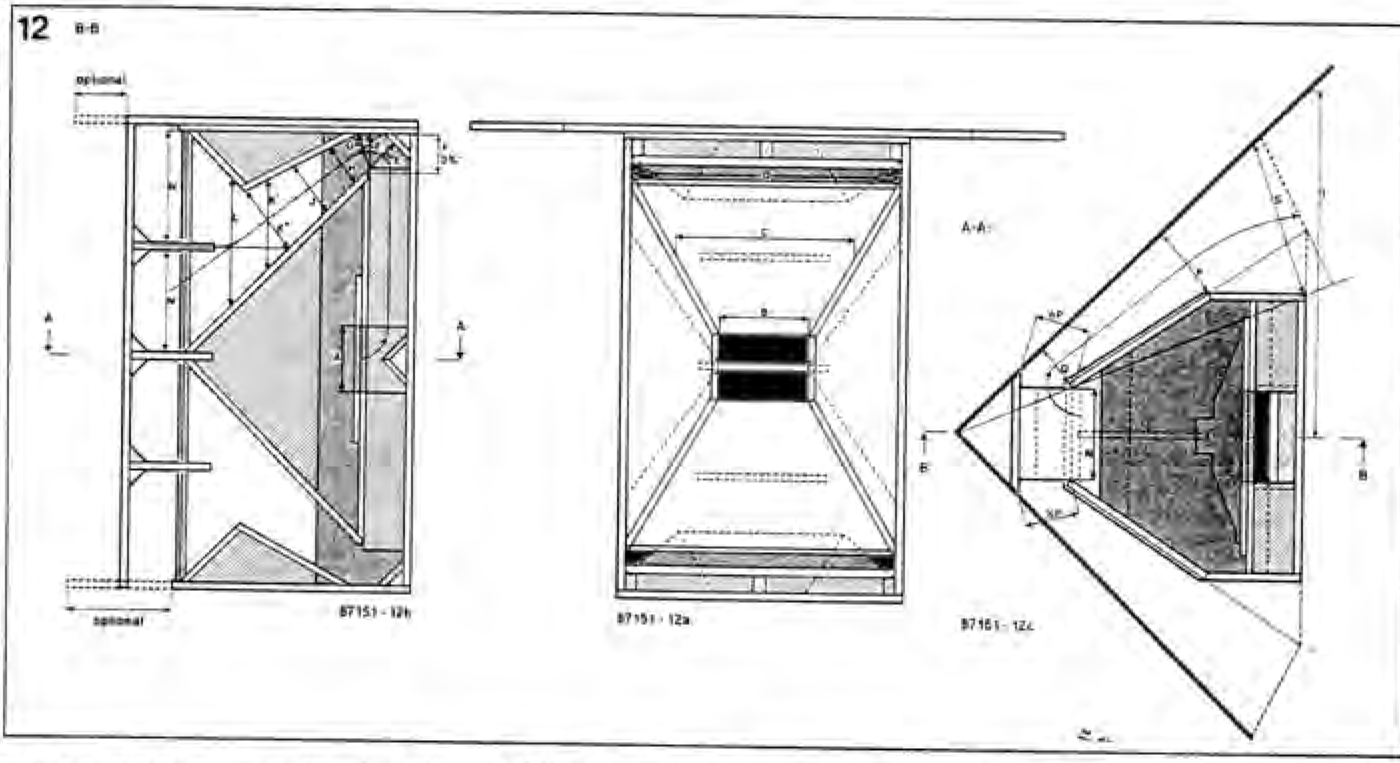

This time, the horn was split laterally and symmetrically if the cabinet was placed correctly. Fig. 7 shows the multi-flared profile, where the natural logarithm of the cross-sectional area has been plotted against channel distance.

Calculated performance with a Richard Allen CG12 driver

Klipsch measured the acoustic impedance, Z_T, of the horn channel with and without the air chamber (which was external in his prototype). The air chamber certainly brought down the peaky reactive part of the impedance: the result, with chamber, is plotted in Fig. 8.

More familiar to electronics engineers will be the Argand diagram in Fig. 9, where the measured acoustic impedance has been converted into its reciprocal, Z_T, "mobility ohms" as suited to the secondary circuit of Fig. 1b. The value of $R_E/(Bl)^2$ has also been plotted for the CG12, where R_E=6.5Ω and Bl=13.7 T m^{-1}. Maximum power is produced from a constant-voltage generator when Z_T equals $R_E/(Bl)^2$, and energy efficiency is just 50%.

The degree of mismatch between Z_T and $R_E/(Bl)^2$ corresponds to acoustical power loss. If Z_T exceeds $R_E/(Bl)^2$, efficiency does, indeed, rise, but not enough to compensate the drop in load current. If Z_T is less than $R_E/(Bl)^2$, efficiency falls rapidly, with more power being dissipated in the voice coil's ohmic resistance.

The other parameters for the Richard Allen CG12, i.e., moving mass and spring constant, were inserted into the complete electrical network and the equations for acoustical power were solved with the aid of a personal TI-55-II programmable calculator. The resulting frequency response for a 10 V r.m.s. excitation has been plotted in Fig. 10. The maximum predicted output of 5 W may be compared with the 12.3 W that would be developed in an eight-ohm resistance. The 10 V r.m.s. corresponds to a 12 W personal amplifier: the audible effect for bass guitar and concert music is atypical of a domestic 12 W system. The author can credit Klipsch's claim to have achieved a tenfold increase in loudspeaker efficiency over infinite baffle types. For comparison, Fig. 10 also shows the calculated performance of another driver unit (4 ohm), whose Bl factor was much lower at 3.35 T m^{-1}. The lower excitation voltage of 4 V r.m.s. would not develop more than 5 W, but the point to notice is the much more peaky response, showing that the reflected voicecoil resistance has not been located centrally on the Argand diagram in Fig. 9. The klipschorn is capable of handling up to 100 W with less than 1% second-harmonic distortion at 40 Hz (which is mainly due to non-linearity of the air in the throat region (Ref. 5).

The roll-off below 60 Hz looks at first like an admission of failure, but remember that Fig. 10, and Klipsch's own prediction for his Jensen 12A, is based on measurements made on a prototype design. When Klipsch evaluated the final cabinet design, he found that

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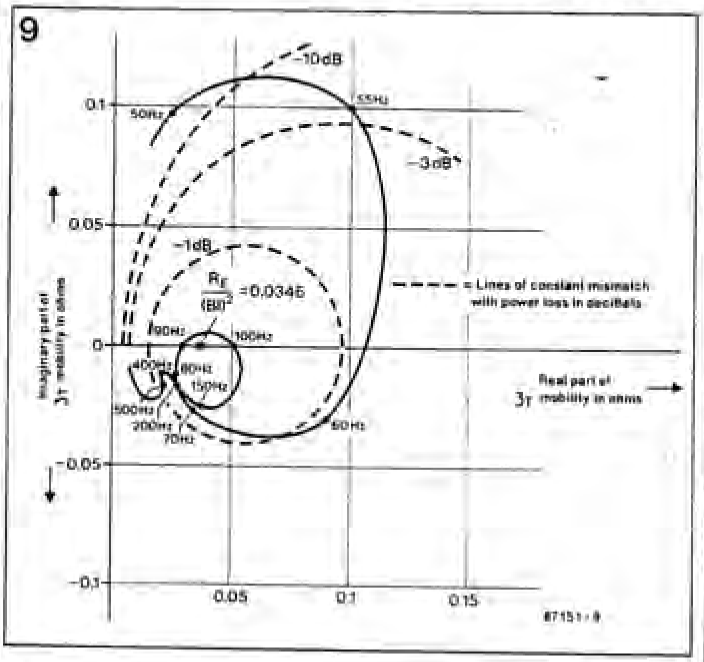

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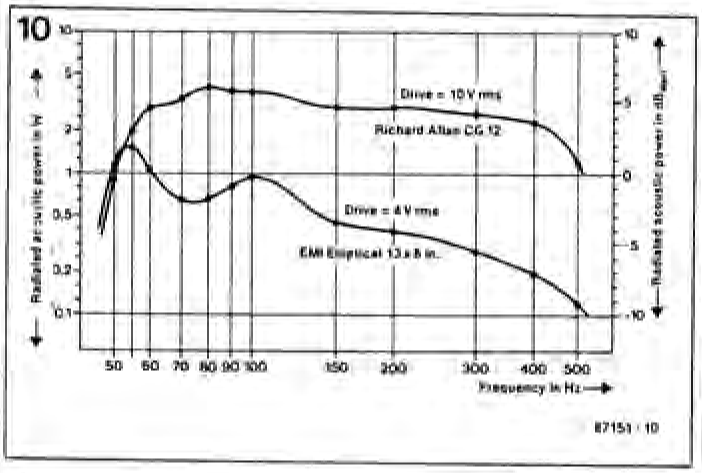


tance. It also indicates the way to compensate it, namely by the series connection of a positive capacitive reactance of the same magnitude. In acoustics terms, this amounts to placing a compliant volume of air behind the diaphragm, enclosed in an airtight chamber—see Fig. 4. Note that this action is not in any way a frequency-selective tuning operation: in principle, it is rather a broad-band reactance annulling process.

The volume of the air chamber is calculated by multiplying the throat area, S_T, by the length in which the horn doubles its cross-sectional area, and then by 2.9. Typically, this volume is small compared with the enclosed volumes required for infinite baffle designs.

For finite horns, the throat-impedance curves exhibit a degree of periodicity, with the depth of the oscillations increasing as the horn is made shorter and thinner. Fig. 5 shows the behaviour of the real and imaginary parts of the throat impedance, computed by Olson, for a horn with a mouth circumference of $0.71\lambda_L$, where λ_L is the wavelength at the cut-off frequency. Once the flare constant and the size of the mouth have been decided, the length of the horn depends only on the size of the throat. Often, the desirable length is quite impractical at low frequencies.

Practical constraints

Ideally, the horn would possess a wide mouth, say, 8.3 m circumference for good matching to the room at 40 Hz: this would entail a length of around 5 m. Some design compromise is clearly indicated if horn-loaded enclosures are to be adopted for semi-fixed or even portable applications. A number of actions can be taken to ease the dimensional limitations. The best-known of these is to fold the horn back onto itself once or twice to make a more compact, box-shaped structure. Less well-known, perhaps, is the method used by Lee in his catenoid design (Ref. 3), or that of P.W. Klipsch, in which the mouth of the horn is made to illuminate the room from one of its corners.

The effectiveness of the Klipsch method can be understood by considering the fact that placing a sound source close to a solid plane produces an in-phase image behind the plane. Similarly, a source located near the intersection of two planes gives rise to two images, and near a corner of a rectangular box three in-phase images accompany the source. The effect of placing the mouth of a horn near a corner is to quadruple the effective mouth area, which very usefully relaxes the earlier stated conditions on mouth circumference: it halves the circumference.

The klipschorn

P.W. Klipsch described a *Low Frequency Horn of Small Dimensions* (Ref. 4) in 1941. His experimental work was cut short by the Second World War, but provided enough data to establish the final design with confidence. It included all the design improvements described in the foregoing section in an ingeniously designed cabinet pictured in Fig. 6.

The design cleverly conceals a doubly split reflex horn of over 1.5 m in length, and makes use of the room walls for two sides of the horn. The mouth of the horn is formed by two rectangular, vertical slots (including the contiguous images in the walls and floor) which make a phased array that helps to beam sound in the horizontal plane, and at the same time increases the specific acoustic impedance.

I shall not repeat or even summarize Klipsch's detailed analysis, but rather assume his measured results and apply them to the present project, which consists of a klipschorn enclosure and a Richard Allen CG12 driver unit. Klipsch used a mains energized Jensen 12, 12-inch driver, which enabled some acoustical measurements to be made via the voice coil terminals simply by switching on or off the magnetic field. The wedge-shaped air chamber has a volume of 64 l of which 9.8 l were taken up by the driver unit.

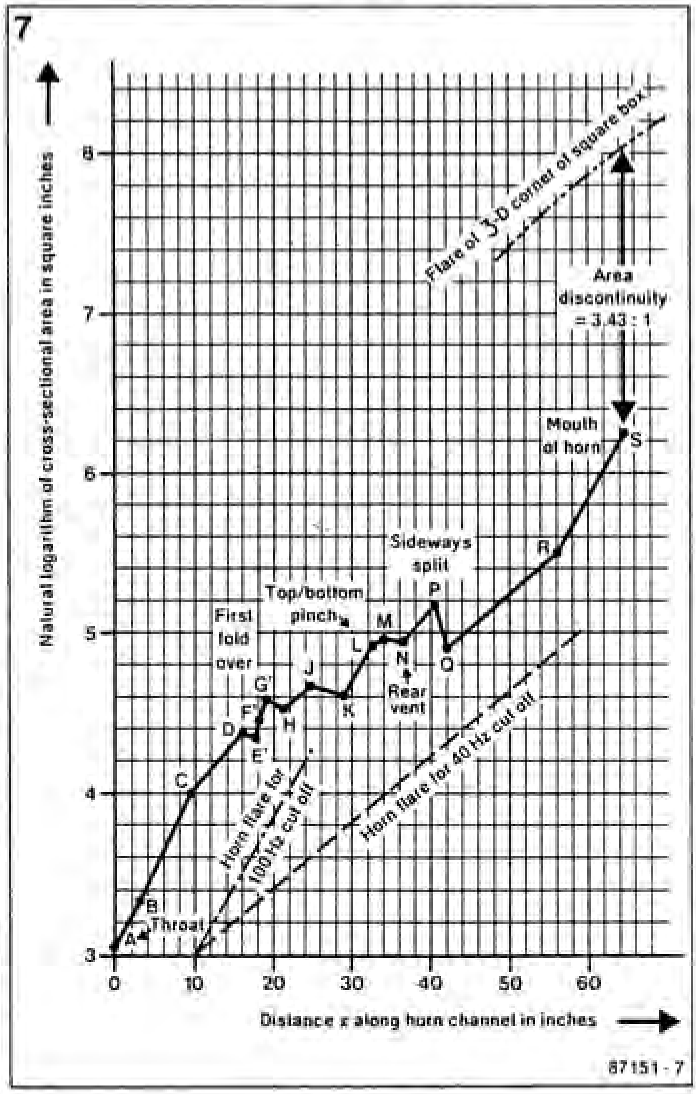
The chamber, by virtue of its pyramidal shape, was free of mid-range resonances and required no damping (which would reduce efficiency in any case).

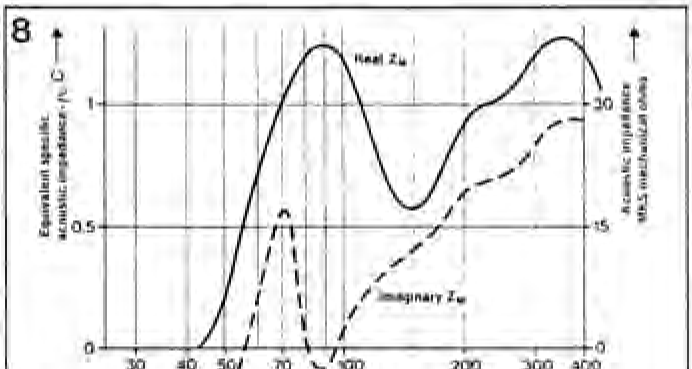
The piston diameter of the cone (10.5 in) gave an area $S_D = 0.0558$ m², which was reduced to 0.0322 m² at the entrance to the throat to give a ratio $S_D/S_T = 1.73$ in Eq. [11]. This was found to be too large at the lowest frequencies, and so a "rubber throat" was devised to give an effective throat area of 0.0644 m² at 40 Hz, reducing to 0.0322 m² at 100 Hz. The "rubber throat" was brought about by making the first section of the multi-flared horn cut off at 100 Hz, and the rest of the horn at 40 Hz.

The throat opened into a split horn with symmetrical channels pointing up and down for the 100 Hz cut-off section. The two channels folded around the top and bottom of the air chamber, constricting in the lateral dimension, but flaring in the vertical.

The 40 Hz cut-off flare constant was approximately maintained with the aid of a succession of short linear flares for ease of construction.

The sharp corner of the room was hidden by a fillet plate which deflected the now merged sound from upper and lower channels sideways between cabinet sides and walls.





therefore remains constant. On the other hand, the acoustic impedance is defined as the pressure divided by the flow (=velocity times cross-sectional area). So, in the exponential horn, acoustic impedance varies inversely as the cross-sectional area. The horn transforms a small flow at high pressure in the throat to a large flow at low pressure in the mouth. Conventionally, the horn is driven by a diaphragm (e.g., the cone of a loudspeaker) of larger cross-sectional area, S_D, than that of the throat, S_T, as shown in Fig. 2. The force, F_D, acting on the diaphragm is related to the pressure, p_T, in the throat by:

$$F_D = S_D p_T \qquad [10]$$

Also,

$$u_D = u_T S_T / S_D \qquad [11]$$

where u_D is the velocity of the diaphragm and u_T is the particle velocity in the throat.

The mechanical impedance, Z_M, is defined as force per unit velocity. So, for the diaphragm,

$$Z_M = F_o / u_D$$

$$= (S_D^2 p_T)/(S_T u_T)$$

$$= (S_D^2/S_T)(p_o/u_o) \qquad [12]$$

where p_o and u_o are the pressure and velocity respectively in the mouth of the horn.

For a horn of sufficient dimensions, the coupling to the outside world becomes 100% and the specific acoustic impedance at the throat approaches $\varrho_o c$. Strictly, this is true only for an infinite horn, but for practical purposes a horn of which the circumference of its mouth equals the wavelength of the lowest frequency to be propagated is adequate. Under these conditions, the mechanical impedance becomes

$$Z_M \approx \varrho_o c (S_D^2/S_T) \qquad [13]$$

To equal this value of Z_M, an infinite baffle design would require a cone diameter of λ/π, where λ is the wavelength. At 40 Hz, λ=8.28 m, resulting in an impractical cone of more than 2.5 m in diameter.

The most serious restriction relating to low-frequency applications of horns is the cut-off

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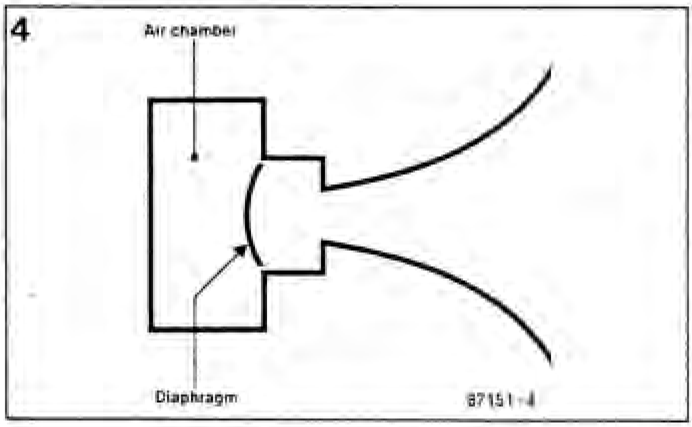

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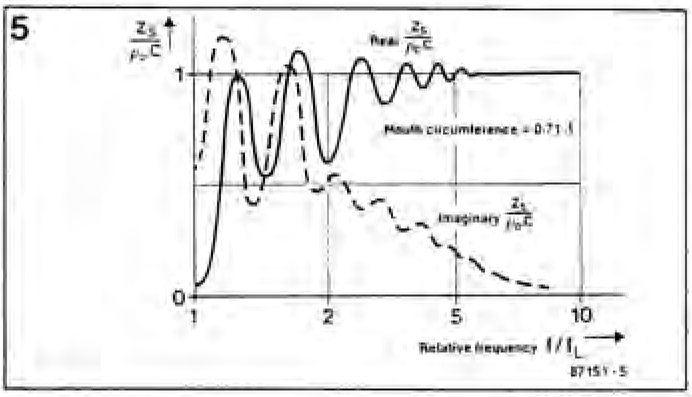

frequency, f_L, below which sound will not propagate. If the cross-sectional area of the horn varies with the distance, x, along the horn as

$$S_x = S_T \exp(mx) \qquad [14]$$

the parameter m, expressed in m^{-1}, is called the **flare** constant. The low-frequency cut-off is given by

$$f_L = mc/4\pi \qquad [15]$$

For instance, if m=0.3646 m^{-1}, f_L=40 Hz.

For an infinite exponential horn, the real part of the throat impedance falls abruptly to zero at the cut-off frequency—see Fig. 3. The imaginary (reactive) part rises to a maximum at f_L and then falls asymptotically to zero. Above f_L it resembles a mirror image of normal capacitive reactance, which is the reason that it is often called negative capacitance reac-

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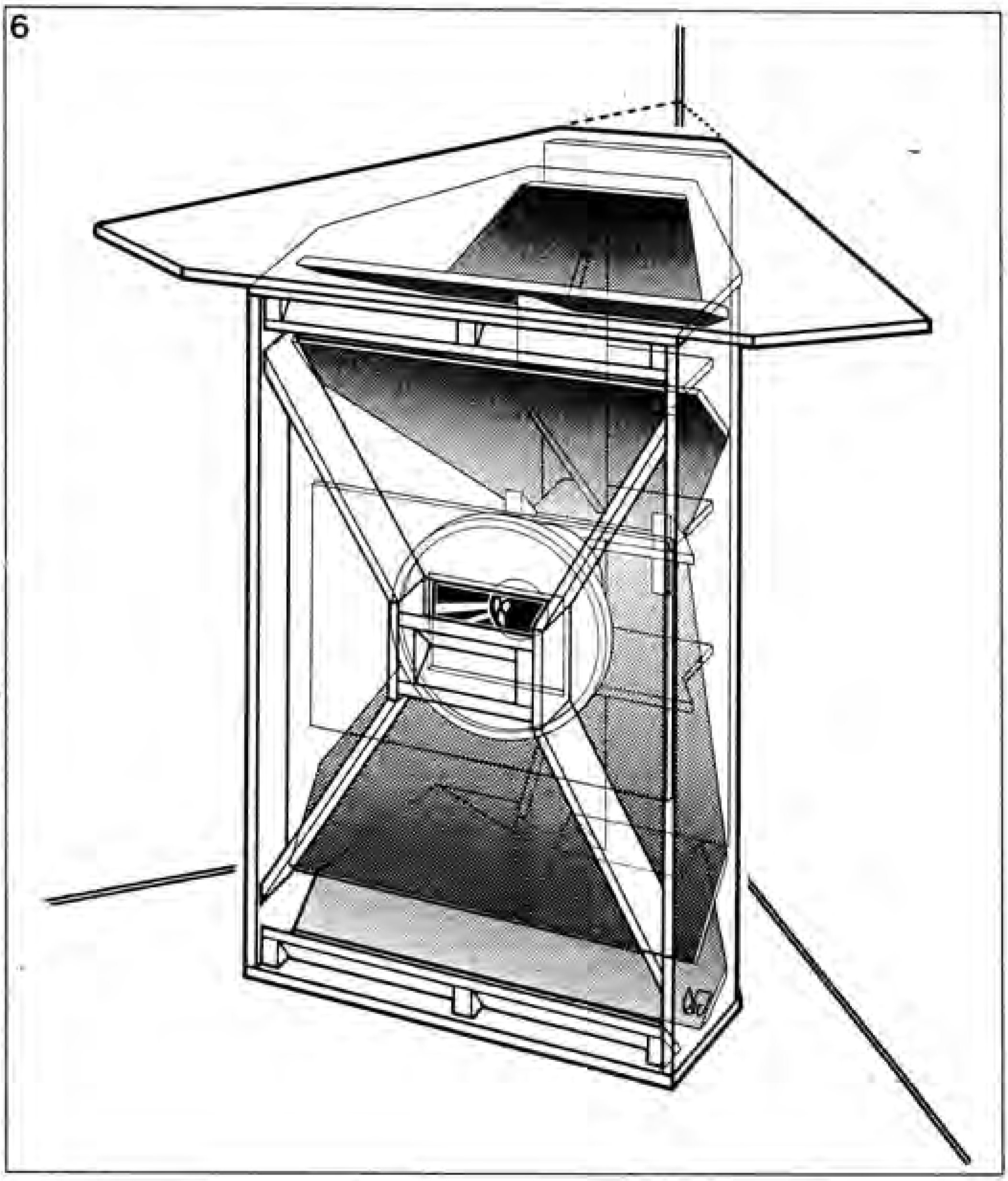

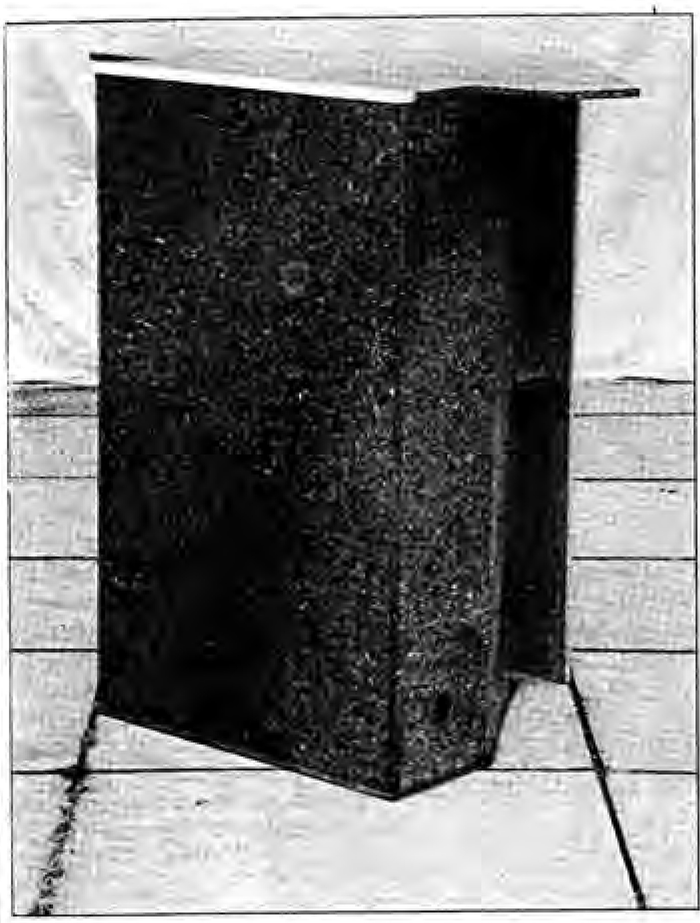

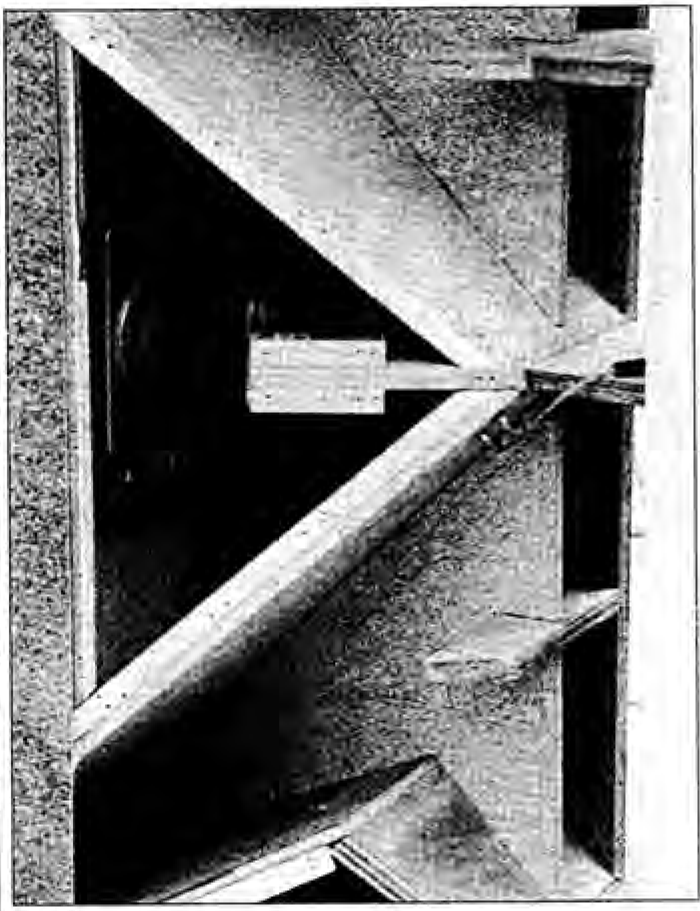

the worse the distortion. Unfortunately, the ear is adept at sensing very low levels of distortion, especially intermodulation distortion.

The concept of transforming electrical energy into acoustic energy introduces ideas of impedance matching and transducers. Without going into the intricacies of acoustics theory, the following simplified treatment of the method of "electrical analogy" may be helpful. The complete sound reproduction system from amplifier to air can be represented by an electrical equivalent circuit—see Fig. 1a. The constant-voltage generator has an internal resistance, R_g, which is typically 0.2Ω for modern amplifiers. The voice coil of the driver unit contributes inductance, L, and pure resistance, R. The essential function of the driver unit is that of energy transducer, which can be represented as a transformer with a turns ratio of Bl:1. The primary current, I_P, gives rise to a force f_P which, as it were, flows in the secondary circuit. The voltage induced back across the primary, V_P, reflects the velocity, u_c, of the voice coil. Thus, $I_P = f_c/Bl$ and $V_P = Bl u_c$.

The primary impedance, Z_1, is related to the secondary impedance, Z_2, as follows

$$Z_1 = V_P/I_P$$

and

$$Z_2 = u_c/f_c$$

so that

$$Z_1 = (Bl)^2 Z_2 \qquad [9]$$

The term Bl is the magnetomotive force factor, which is measured in tesla per metre (T m^{-1}).

Note that in the secondary circuit impedance is the inverse of the better known mechanical analogue where voltage corresponds to force and current to velocity. In the present system, known as the mobility system (Ref. 2), inertia becomes a shunt capacitance and spring resistance becomes a shunt inductance. Hence, the moving mass of the speaker is represented by capacitance M_{MD}, while the stiffness of the suspension and any enclosed volume (e.g. in infinite baffle designs) of air are represented jointly by inductance C_{MS}. The acoustic load is represented by $_A$, the acoustic mobility, which is the reciprocal of acoustic impedance.

For maximum energy transfer, the blocking effects of the mechanical inertia and stiffness have to be minimized. In the mobility system, this amounts to reducing the magnitude of β_{-A} (particularly the real part) below the shunt admittances of M_{MD} and C_{MS}. Then, β_{-A} will predominate in the secondary load.

The accompanying step is to reduce the size of the primary series impedances as reflected in the secondary—see Fig. 1b. At low frequencies, L may be disregarded. Considering the difficulty of achieving large enough values of acoustic impedance (i.e., low enough values of acoustic mobility, β_{-A}), every effort has to be made to reduce the moving mass and stiffness of the suspension. Furthermore, Bl needs to be maximized to minimize the dissipative effect of R_E. The closed cabinet or infinite baffle design reduces stiffness by the use of a large enclosed volume, increases Z_A by the use of a large cone, and then fails to reduce the moving mass (which includes a cylinder of air in front of the cone that is about a third of the cone radius deep). Such designs seldom achieve energy efficiencies much above 4%, in spite of decades of research.

The horn as an acoustical transformer

For a sound wave propagating along an exponential horn, the particle velocity and pressure vary in proportion to the diameter of the horn. The specific acoustic impedance

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