LOUDSPEAKER MANIFORLDS FOR HIGH-LEVEL CONCERT SOUND REINFORCEMENT

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ABSTRACT

The acoustical manifold is presented as a device for improving the performance of high-level concert sound reinforcement speaker systems. Recently developed bass, midbass, and high-frequency devices are discussed, each of which effectively sums the output of four loudspeakers, producing from the four a single coherent source. The advantages of manifolds over multiple sources are improved audience coverage, reduced polar response lobing, small-frontal area arrays, small transportable enclosures, and in certain cases reduced distortion and increased efficiency.

INTRODUCTION

High-level concert sound reinforcement places the greatest demands on current loudspeaker technology in the area of maximum acoustic power output. The sound pressure levels and component reliability required have been attained in the past by creating large arrays of sources. This practice reduces coverage pattern control and produces polar-response lobing. In addition, the practice results in large-frontal-area arrays and bulky loudspeaker systems to transport.

By combining the outputs of multiple drivers within a carefully defined space, a manifold, these effects can be reduced. The result is extremely high-level sound with minimized distortion, small-frontal-area arrays, small transportable enclosures, greatly reduced directive response anomalies, and in certain cases increased efficiency.

BACKGROUND

Manifolding is not a new idea. In the 1940's there was an RCA Y-throat for coupling two one-inch-throat drivers to a special horn. This and various other Y-throats have been used with some success in siren and voice warning systems, where the redundancy of drivers is a decided advantage. Their use in quality musical sound reinforcement has been limited primarily to midrange applications because of cancellations at high frequencies. Figure 1 is the summation response[1] of a popular large Y-throat (drivers with 49mm exits into a special horn), which reveals why many who have experimented with Y-throats describe them as "dull-sounding". We submit that this performance is characteristic of specific manifold designs and is not indicative of the potential of manifolding in general.

The general concept of manifolding is based on the prediction that two like drivers mounted on a manifold will be indistinguishable from a single driver with twice the diaphragm area and an equivalent motor. The two-driver version has the advantages of smaller diaphragms (which are capable of pistonic motion through the passband), greater power-handling (thermal-limit) per square centimeter of diaphragm, and greater reliability.

Furthermore, Henricksen [1] has shown that given currently known materials, a very abrupt limit is reached in the ultimate performance of compression drivers. As a result, a given system requirement may call for a compression driver that is not physically realizable, necessitating the use of multiple drivers. Compared to multiple horns, a smaller number of horns utilizing manifolds will produce less polar response lobing and better audience coverage. They will also form smaller arrays and cost less to implement.

But what happens when "like" drivers vary from one another, as all real drivers will? At low frequencies, that is frequencies at which there is mutual coupling between drivers, there is usually very little unit-to-unit variation between drivers. Furthermore, the type of variations that occur generally sum mathematically (and acoustically) to an output that is between that of the two contributors. At high frequencies on the other hand, compression drivers may show significant unit-to-unit variations, not

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1. Summation response, as used herein is defined as the difference between the 1/3-octave smoothed response of a horn-driver combination and the 1/3-octave smoothed response of the same horn with a manifold and similar drivers. The reason for the smoothing is to avoid attributing local response changes to the manifold which are really due to the change in horn length that occurs when a length of throat section is replaced with a manifold.

necessarily in average level but in the exact frequencies at which particular peaks and dips occur. Simple mathematical summation of randomly selected drivers may yield a frequency-response that is worse than the individual contributors. Fortunately though, as unit-to-unit variations become significant at higher frequencies, directive behavior (ray propogation) within the manifold occurs. It has been found that a manifold which effectively transfers sound "rays", as well as sound waves, performs well even with unmatched drivers and up to much higher frequencies than one which only addresses wave behavior.

The discussion so far has been concerned with manifolding of high-frequency compression drivers, since historically this has been the most prevalent application of manifolding. However, the concept has been found to be equally valuable at lower frequencies, where the benefits are not as intuitively apparent. After a compression-driver manifold design has been described, a mid-bass and then a low-frequency manifold will be discussed to illustrate the unique benefits of this design approach over a wide range of frequencies.

DESIGN OF A MANIFOLD FOR FOUR HIGH-FREQUENCY COMPRESSION DRIVERS

A manifold has been developed utilizing the concepts mentioned above as well as several others to achieve a summation response suitable for critical applications. The drivers are arranged as close together as possible to maintain small-cross-section acoustic paths. The wavefronts entering the manifold encounter reflective surfaces, rather than curved tubes. Additionally, two different entry-angles are used to obtain a smoother summation.

As a starting point, an array of drivers was created which produced the minimum possible spacing of the driver throats. This minimum spacing occurs when the four drivers are arranged radially with magnets nearly touching and exits facing a central point (like spokes of a wheel). Two facts become immediately obvious. First, a relatively severe bend must occur within the manifold (a 90-degree bend for the closest exit spacing). And second, convex-drive compression drivers are inherently better suited to this type of design than equivalent concave-drive, through-the-pole-piece drivers, since the convex-drive magnets are behind the diaphragms, permitting closer packing of driver exits.

Using the concept of ray tracing, a right-angle reflective bend was modelled and found to work very well, as shown in Figure 2. This curve shows the difference in frequency response of a compression driver and horn measured with and without a right-angle reflective bend in a round-cross-section connecting tube between the horn and driver. This data shows that by using an appropriate reflector the bend can be traversed with negligible loss of response. However, when several of these reflectors are brought together in a manifold, the path of each wavefront leaving its reflector is not constrained by a tube, as it is in the pure case. The resulting summation response is not as smooth.

The solution to this unevenness of response is based on the observation that the frequencies of cancellation may be predicted with a loosely related formula. The off-axis response of an ideal piston, as presented in Kinsler and Frey [2], shows a good correlation with these cancellation frequencies if a piston twice the size of the input tube is assumed. The ideal-piston off-axis response is described by:

$$p = A \cdot \frac{2 \cdot J1(k \cdot a \cdot \sin\theta)}{k \cdot a \cdot \sin\theta} \qquad (1)$$

in which;

- a = the radius of the piston (use twice the radius of the tube),
- i = the angle of the bend,
- k = the wave number (w/c),
- A = the unperturbed response (same source without bend),
- J1(x) = the first order Bessel function of the first kind.

The important aspect of this observation is that it holds for changing angle, i. Furthermore, if the function is plotted for various angles, it may be seen that certain pairs of these functions appear to be complementary - dips in one correspond to peaks in the other. Consequently, the manifold design was refined one step further by tilting two of the drivers 45 degrees, which still maintains the minimum exit spacing. Figure 3 shows a four-driver manifold with this configuration - notice the different angles of entry. Equation 1 is plotted in Figure 4 for a 23.6mm tube (47.2mm piston) at angles of 90 and 45 degrees. It shows that the first two nulls in the 90-degree response correspond to lobes in the 45-degree response. Likewise, the first null in the 45-degree response corresponds to a lobe in the 90-degree response.

If the contributions from both sets of drivers were highly correlated and well represented by equation (1), their summation would be similar to either contribution singly (the sum of equation [1] for two different angles is another function representable by equation [1]). Fortunately, though, the combining waves have some directionality and the individual contributions have much shallower dips and very little high-frequency roll-off. As a result, the two characteristics do have a general smoothing effect, which is further augmented by adjusting the shape of the reflective detail to distribute high-frequency energy evenly over the area of the manifold exit. The final result is the summation response in Figure 5. A photograph of a prototype manifold showing the driver arrangement and reflective interior detail appears in Figure 6.

DIRECTIVE PROBLEMS ASSOCIATED WITH LARGE-THROAT CONSTANT DIRECTIVITY HORNS

As a result of manifolding multiple drivers, the entrance to a horn designed to accept the manifold must be larger than it would have been for a single driver. In the case described above, changing from a 23.6mm entrance to a 49mm entrance caused some directive anomalies which required a more complex horn throat design.

Figure 7 is a beamwidth plot that is typical of constant directivity horns with 49mm entrances. The curious behavior at the upper end of its range is an on-axis dropout causing an apparent widening of coverage in the octave around 10.5 kHz. The collapsing beamwidth above 15 kHz is due to the excessive diameter of the throat (if the throat is too large, the initial dispersion is narrower than the coverage angle of the horn, so the horn walls have very little effect on the beamwidth). Figure 8 shows the on-axis dropout.

Once again, ray tracing provides the explanation. A circular-cross-section wavefront encountering a change to a rectangular cross-section will "attempt" to redistribute its energy so that it is distributed rectangularly. Since the area of a rectangle is distributed more toward the corners (as opposed to the center), the wavefront acquires an intensity component that is directed across the desired propagation path, toward the sides of the horn. When this component reflects off the straight walls of a constant-directivity horn, an interference pattern develops, resulting in on-axis dropout. Figure 9 shows, graphically, the paths of the rays. This postulate has been supported experimentally as it predicts, accurately, the dropout frequency for varying wall angles and horn entrance diameters. For a 49mm-throat horn, this frequency varies from 7 kHz in a 60-degree waveguide to 16 kHz in a 20-degree waveguide.

Once the mechanism causing the problem was understood the solution was straightforward. A pair of vanes positioned in the throat of the horn cures both of the directive problems by creating a sectoral horn for a few inches - just long enough to allow the wavefronts to take on the shape of the horn, but not so long as to produce the response ripples characteristic of sectoral horns. The position and size of these vanes is critical to the solution of the problem and to the maintenance of smooth, lossless response.

DESIGN OF A FOUR-DRIVER MANIFOLD FOR THE MID-BASS FREQUENCY RANGE

Some of the concepts used in the design of the high-frequency manifold were also applied to the design of a mid-bass manifold. Close driver packing was utilized to minimize the path length and cross-sectional areas of the manifold, and reflecting surfaces were used to achieve good summation.

Before the problems of manifolding could be tackled, several other problems had to be addressed. For a maximum-power system, a driver was required that would provide flat frequency response from 150 to 2000 Hz on a constant-directivity horn. A conventional high-power, 25cm cone-type loudspeaker was chosen as the basic drive unit, and a special phase plug was designed. Details of the development of the phase plug are beyond the scope of this paper, but to summarize; the phase plug makes use of the fact that at higher frequencies the amplitude of vibration is greater near the voice coil than it is near the cone edges. By loading the cone asymmetrically, high-frequency output is maximized. Additionally, the phase plug has a rectangular exit, making manifold construction easier and permitting a simpler interface to a wood midbass horn with a rectangular-throat opening.

As a demonstration of mid-bass manifolding, a single driver mounted on a mid-bass horn (a 60-degree by 40-degree horn with a mouth area of .47 square meters - see Figure 10) was compared with two drivers manifolded on a similar horn (also a 60-degree by 40-degree horn with a .47-square-meter mouth - see Figure 11). In the manifolded system the drivers face each other and address right-angle reflective bends. The frequency response of the single-driver system is presented in Figure 12 along with the response of the manifolded system in Figure 13. The increased low frequency level of the manifold system is due to mutual coupling, while the higher-frequency differences are due to a slight change in directivity as a result of the increased throat dimension of the manifold horn. It is apparent that no significant power-loss occurs in the manifold.

The final form of the system employs two of the above manifolds arrayed vertically and loaded by a slightly larger 60-degree by 40-degree horn. The frequency response of this system is shown in Figure 14. Note especially the sensitivity. This system, at 1200 watts continuous power handling capability, can deliver 138dB SPL (144dB peaks) at one meter over its full frequency range.

DESIGN OF A LOW-FREQUENCY SYSTEM UTILIZING A MANIFOLD CHAMBER

The advantages of manifolding woofers are not intuitively obvious, since phase cancellations and destructive interference typically are not problems at low frequencies (because of the long wavelengths involved); however, several benefits are derived that will become apparent later.

Over the years, two basic loudspeaker designs have proliferated in the concert sound industry for the reinforcement and reproduction of low frequencies; horn-loaded systems and direct-radiating vented-box systems. A horn-loaded design usually exhibits more efficiency than a direct-radiating vented-box design, but at the expense of a larger enclosure. When the smaller direct radiators are assembled into large arrays, however, their efficiency increases due to mutual coupling and begins to approach that of horn-loaded systems. Keele [3] has shown that direct-radiating woofers hold an advantage over horn-loaded woofers in acoustic power output per bulk volume occupied. Consequently, a direct radiating vented-box design was chosen for the low-frequency manifold.

To demonstrate the effects of manifolding woofers, two identical vented-box enclosures containing a pair of 46 cm woofers were built. Each had a net internal volume of 174 liters and was tuned to 44 Hz. The two enclosures were initially positioned side-by-side, facing forward as shown in Figure 15. Next the enclosures were turned so that the loudspeakers faced each other as shown in Figure 16. The space between the enclosures was sealed on three sides (top, bottom, and rear) so that the loudspeakers radiated into a common chamber (the manifold chamber) with a single exit, and the ports were relocated to the new front of the cabinets (outside the manifold chamber). Acoustic measurements were made on both configurations.

The on-axis frequency response of the direct-radiating system is shown in Figure 17 and the response of the manifolded system is shown in Figure 18. Comparing the two curves, there are several differences that are readily apparent; below 70 Hz the manifolded system has substantially more output than the direct-radiating system; above 100 Hz the manifold has slightly less output; and above 200 Hz the manifold rolls off abruptly.

The explanation for the manifold's 3-dB advantage below 70 Hz lies in the analysis of the mutual radiation impedance. In the classical analysis of the mutual radiation impedance between two adjacent identical pistons in an infinite baffle, Pritchard [4] points out that the resistive component is absolutely convergent for all values of a (the radius of the pistons) and d (the center-to-center distance between the two pistons), and approaches that of a single radiating piston when the wavelength is much larger than d. This effective doubling of radiation resistance gives us the well known 3-dB increase in efficiency when the number of closely-spaced loudspeakers is doubled. This effect holds equally true for both the direct-radiating and manifolded systems, with the difference between the two designs lying in the reactive component of the mutual radiation impedances. Pritchard points out that the reactive component increases rapidly with decreasing frequency when the wavelength exceeds the spacing d, and additionally, approaches infinity as d approaches a. In the case of the manifolded system both of these cases are encountered. As a consequence, the additional mass loading provided by the manifold is significant enough to raise the Q over the direct radiating case and increase the sensitivity below 70 Hz. The design of the manifold is critical to limit the additional mass loading to the low-frequency rolloff region. If the loading extends into the passband a decrease in midband efficiency will result.

From 70 Hz to 100 Hz the manifolded response is very similar to the direct-radiating case. As frequency increases the effect of the additional mass loading due to manifolding decreases, producing little effect above 70 Hz. Above 100 Hz the on-axis sound-pressure level is slightly higher for the direct-radiator system because of increased directivity due to the larger radiating area. At yet higher frequencies, the manifold chamber acts as a low-pass filter. This high-frequency rolloff is of no consequence for low-frequency operation (less than 200 Hz), but offers the unexpected benefit of significantly reducing distortion in the passband by acoustically attenuating higher harmonics.

A properly-designed manifold chamber can offer increased low-frequency efficiency over direct radiators with equivalent internal volume or, conversely, offer the same low-frequency output from a smaller box.

DESIGN OF A FULL-RANGE, ALL-MANIFOLDED SYSTEM

An example of this new manifold technology is the MT-4, a four-way-active, two-box system shown in prototype form in Figure 19. There are four drivers in each frequency band for a total of sixteen drivers. The enclosures have identical dimensions of .91m X .91m X .76m (36in. X 36in. X 30in.), providing integer divisors of a standard truck-bed. The prototype shown is of slightly different proportions.

The lower box, the MTL-4, is a vented-box design that contains four 46 cm woofers, each facing into a manifold chamber at the center of the enclosure. Additionally, the woofers are mounted "magnets-out" for more efficient heat transfer. In the frequency range below 70 Hz, the manifold design is nearly twice as efficient as typical horn-loaded systems of equivalent size. Its sensitivity of 101dB(1W/1m) and 1600 watt continuous power handling capability enable it to produce 133dB SPL (139dB peaks) at one meter in full-space. The sensitivity can, of course, be increased by placing the enclosure on the floor and/or using multiple enclosures.

The upper box, the MTH-4, is a three-way mid-bass/midrange/high-frequency system. The mid-bass section is the four-driver system described earlier. The mid-range section consists of four 51mm titanium-diaphragm compression drivers manifolded onto a 60-

degree by 40-degree constant-directivity horn. The high-frequency section consists of four 32mm titanium-diaphragm compression drivers on a horn identical to that used for midrange. Using identical horns and manifolds for both the midrange and high frequencies results in equal path lengths for the two sections.

The frequency response for the complete system with recommended crossover, equalization and delay compensation is shown in Figure 20. Crossovers are fourth-order Linkwitz-Riley filters at 160 Hz, 1600 Hz, and 8 kHz. Beamwidth plots for each section are shown in Figure 21.

CONCLUSIONS

The use of an acoustic manifold to combine the outputs of several loudspeaker drivers to provide a single acoustic source with high-quality performance has been demonstrated, and designs were presented for manifolding four drivers in various frequency ranges. The merits of manifolding include improved directivity control and audience coverage (mid-bass through high frequencies), increased efficiency (low frequencies), and extremely high power-to-enclosure-volume ratios (all ranges). Applications such as concert sound reinforcement, which require extremely high acoustic power output, should benefit greatly from these new forms of manifold technology.

The devices mentioned in this paper are the subjects of a number of patents currently pending or approved.

ACKNOWLEDGEMENTS

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REFERENCES

[1] Clifford A. Henricksen, "Sound System Designs Using Mechanical Specifications of Drivers", Synaudcon Tech Topics, Vol. 11, No. 2 (1983).

[2] Lawrence E. Kinsler and Austin R. Frey, Fundamentals of Acoustics, Wiley, 1966.

[3] D.B. Keele, "An Efficiency Constant Comparison Between Low-Frequency Horns and Direct Radiators", presented at the 54th Convention of The Audio Eng. Soc., Preprint no. 1127, May 1976.

[4] R. L. Pritchard, "Mutual Acoustic Impedance Between Radiators in an Infinite Baffle", J. Acoustical Soc. of America, Vol. 32, No. 6, pp. 730-737 (June 1960).

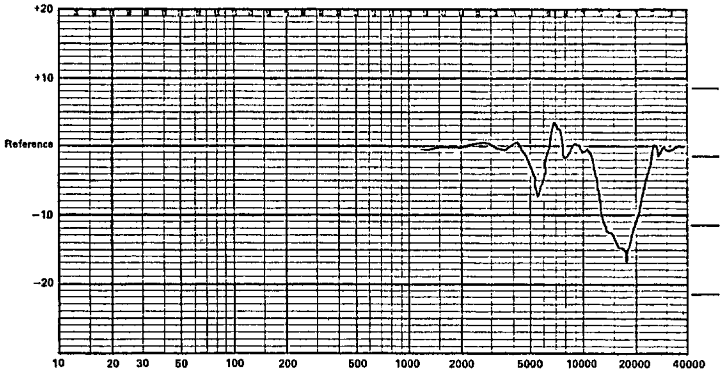


Figure 1. Summation Response of Large Y-Throat

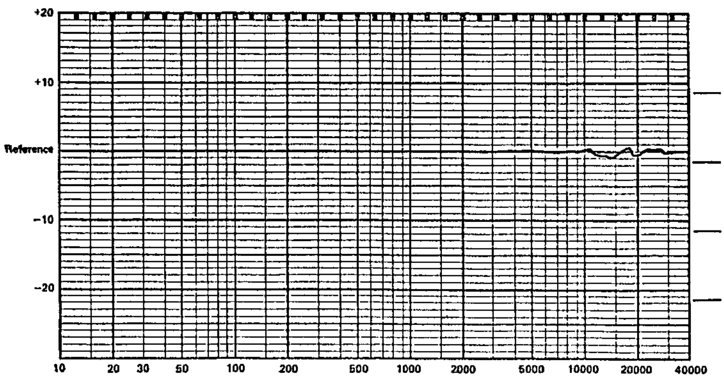


Figure 2. Right-Angle Reflective Bend; Difference Between Horn with Straight Tube, Horn with Reflective Bend.

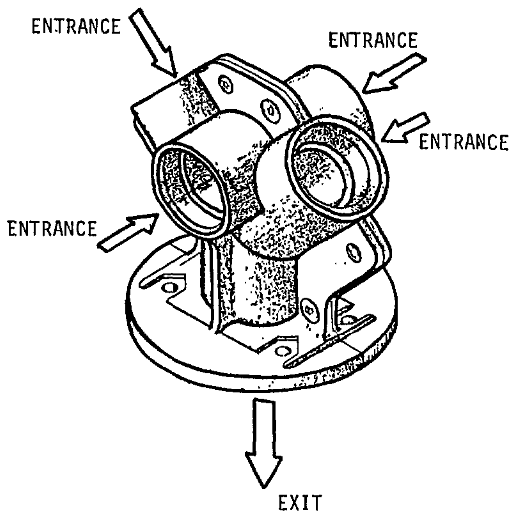
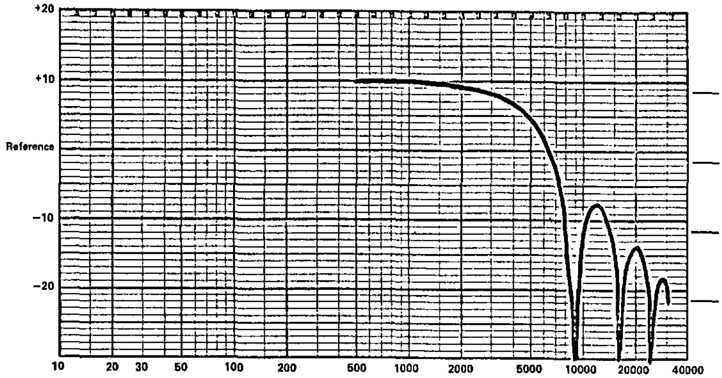
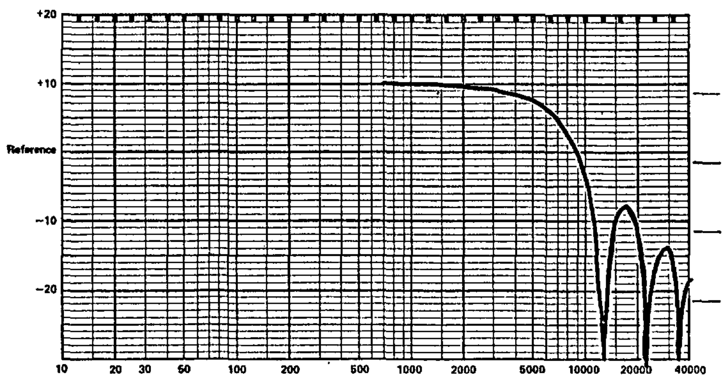


Figure 3. Manifold for Four Compression Drivers.



a)



b)

Figure 4. Off-Axis Response, Piston Source; a) 90 Degrees. b) 45 Degrees.

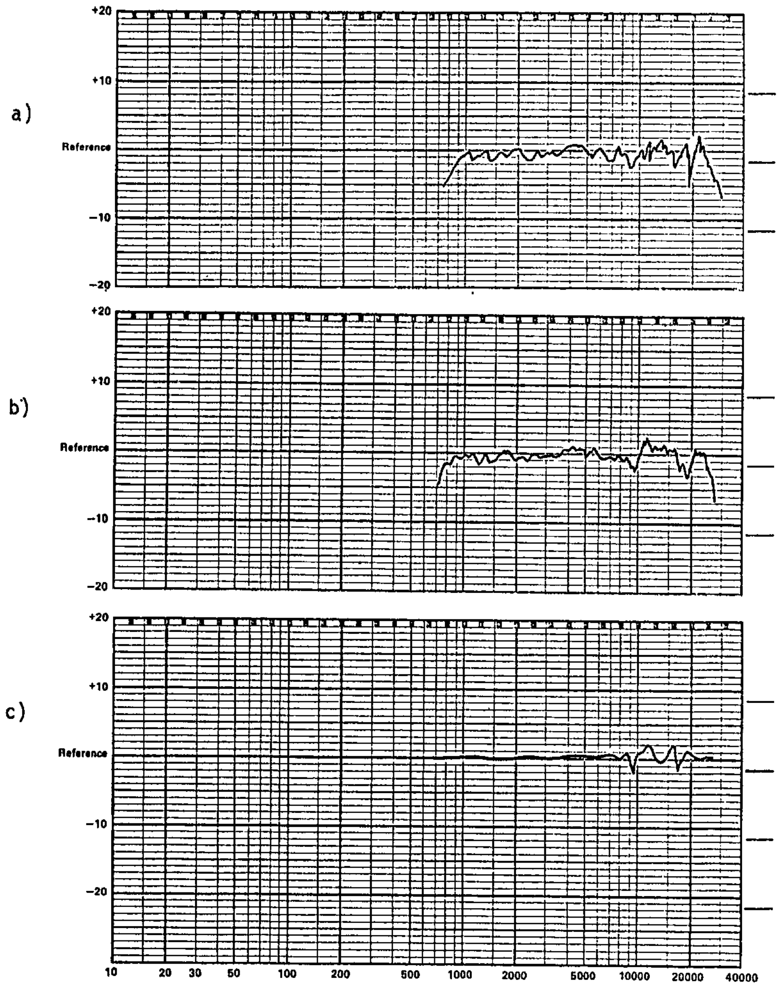


Figure 5. Four-Driver Manifold Summation; a) Single Driver on Horn, equalized. b) Manifold on Same Horn with Same Equalization. c) Summation Response.

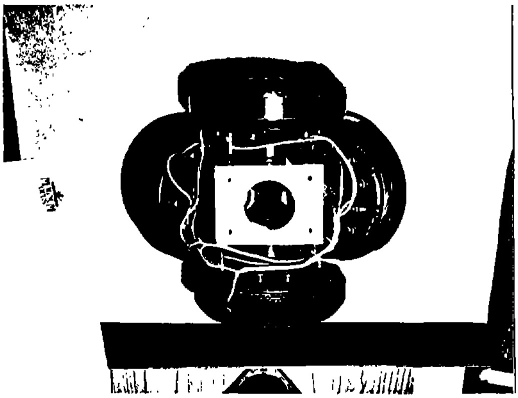


Figure 6. Photograph of Prototype Manifold with Four Compression Drivers.

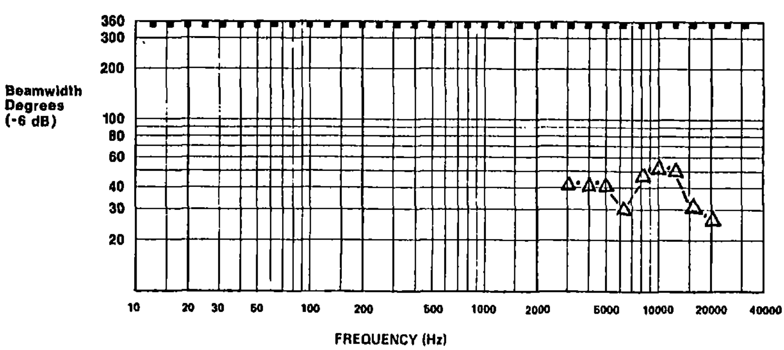


Figure 7. Beamwidth Plot of Typical 49mm Throat Feeding 40-degree Horn

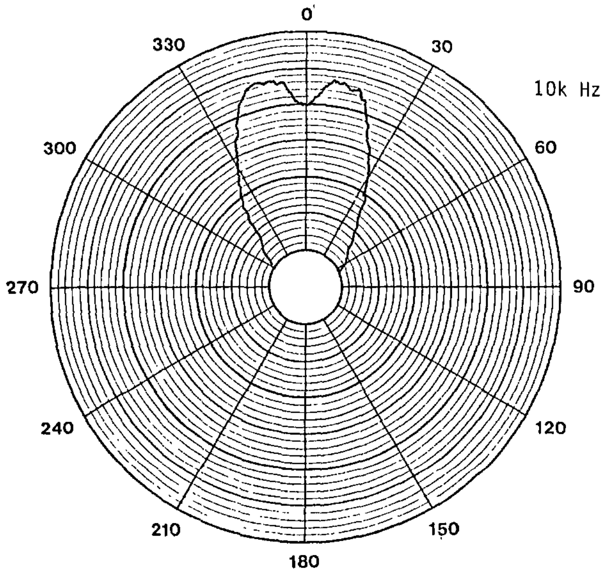


Figure 8. Polar Response, Typical 49mm Throat Feeding 40-degree Horn

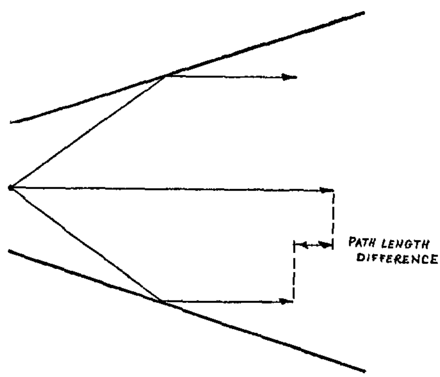


Figure 9. Path of rays in Horn Throat

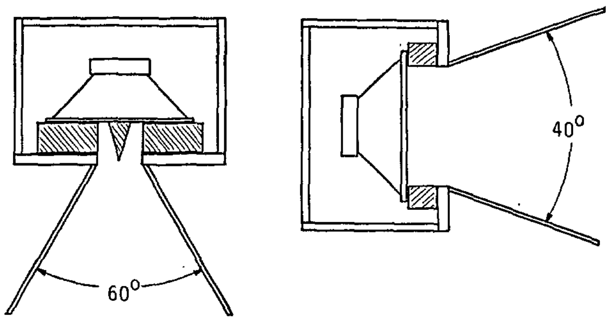


Figure 10. Single Mid-Bass Driver on a 60° X 40° Horn.

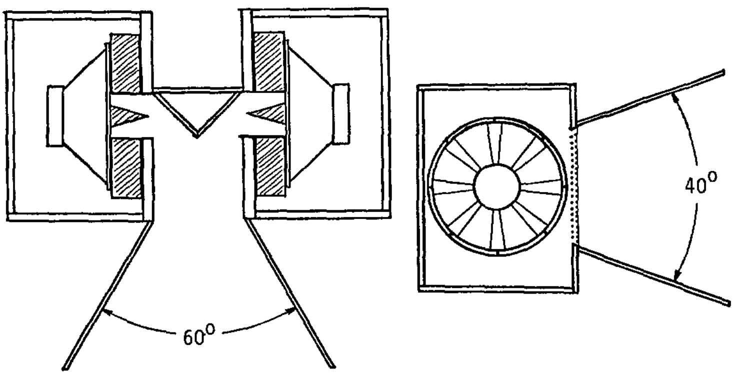


Figure 11. Two Mid-Bass Drivers Manifolded on a 60° X 40° Horn.

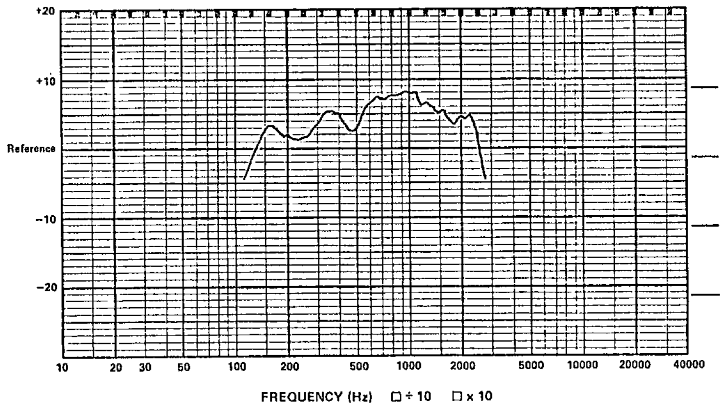


Figure 12. Frequency Response of Single Mid-Bass Driver (1 w @ 1 m; 100-dB Reference).

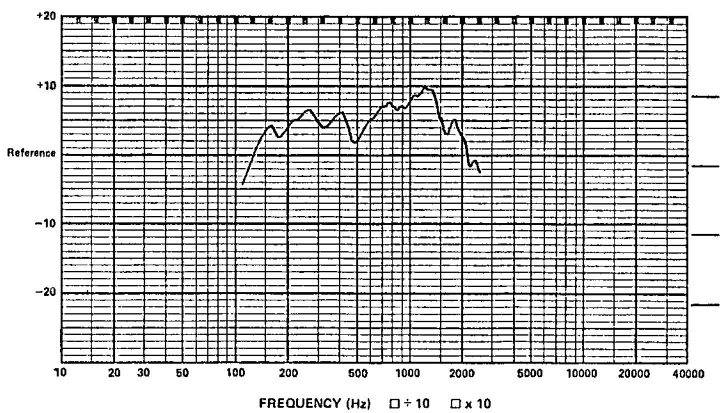


Figure 13. Frequency Response of Double-Manifolded Mid-Bass Drivers (1 w @ 1 m; 100-dB Reference).

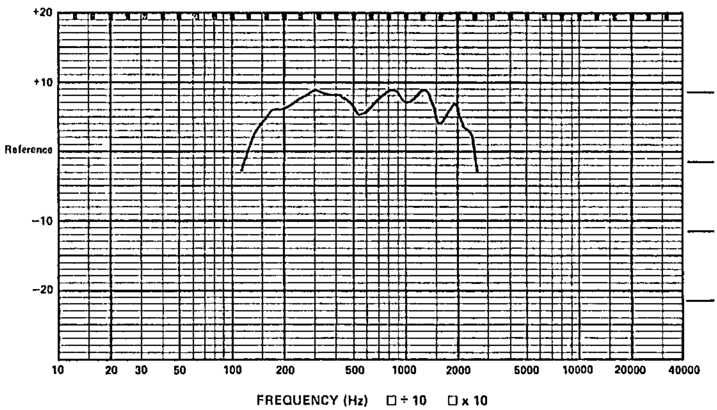


Figure 14. Frequency Response of Quad-Manifolded Mid-Bass Drivers (1 w @ 1 m; 100-dB Reference).

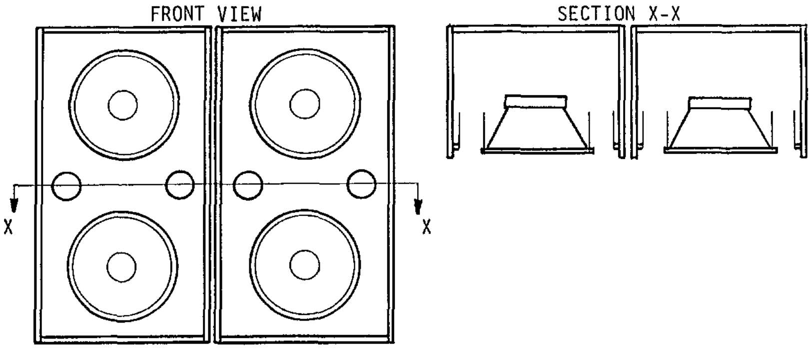


Figure 15. Four 46 cm Woofers Direct Radiating.

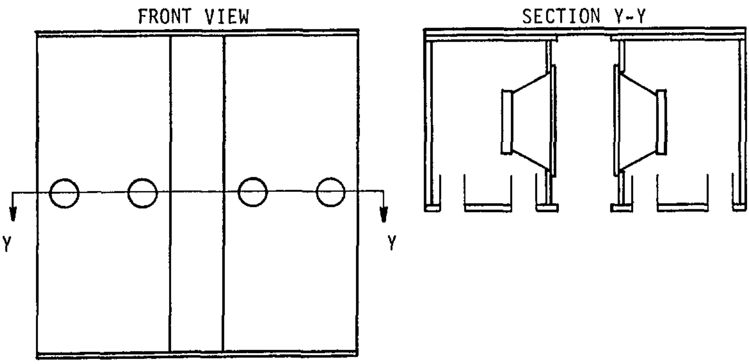


Figure 16. Four 46 cm Woofers Manifolded.

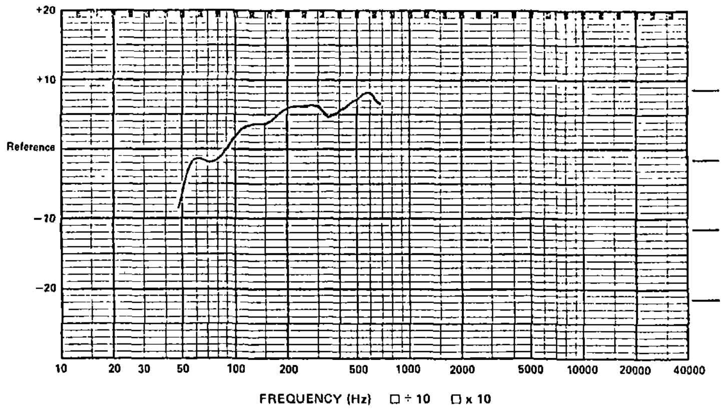


Figure 17. Frequency Response of Four Direct-Radiating Woofers. (1 w @ 1 m; 100-dB Reference).

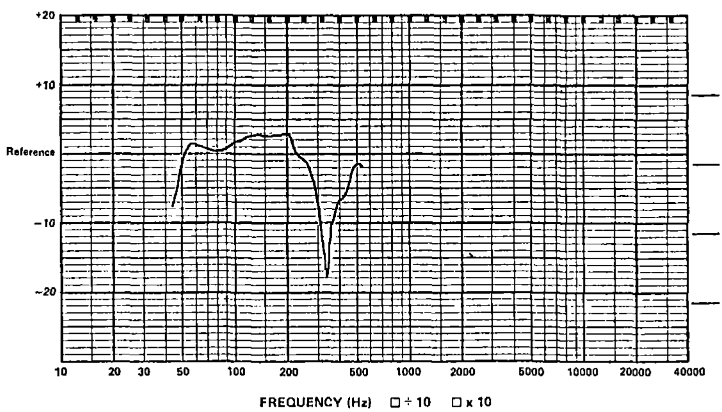


Figure 18. Frequency Response of Four Manifolded Woofers (1 w @ 1 m; 100-dB Reference).

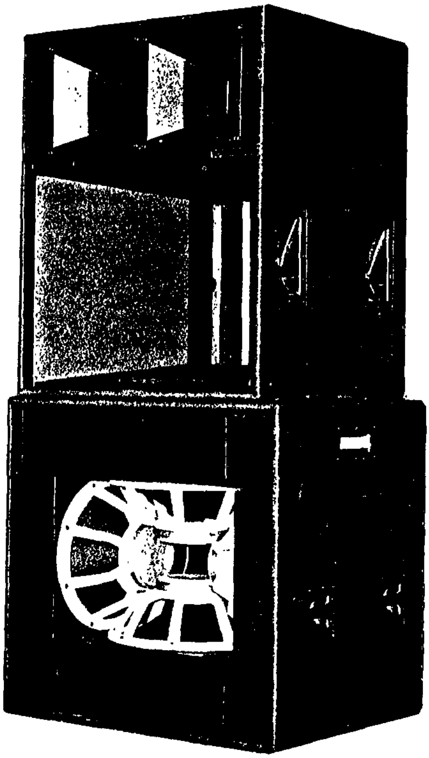


Figure 19. Photograph of Prototype MT-4 Four-Way Manifolded Loudspeaker System

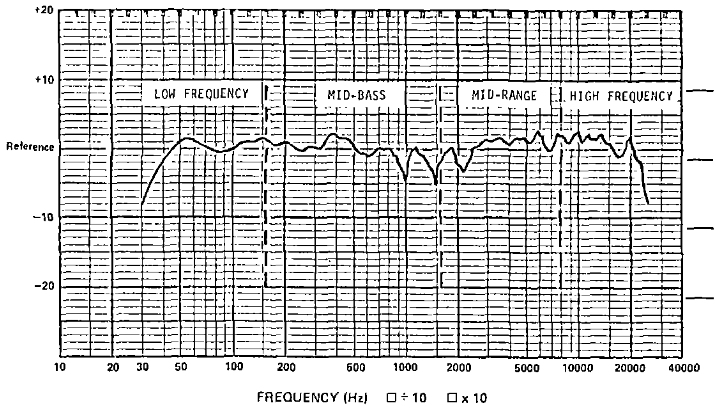


Figure 20. Frequency Response of MT-4 Loudspeaker System with Recommended Crossover, Equalization and Time Delay (10 w @ 3 m into Low-Frequency Section; 100-dB Reference).

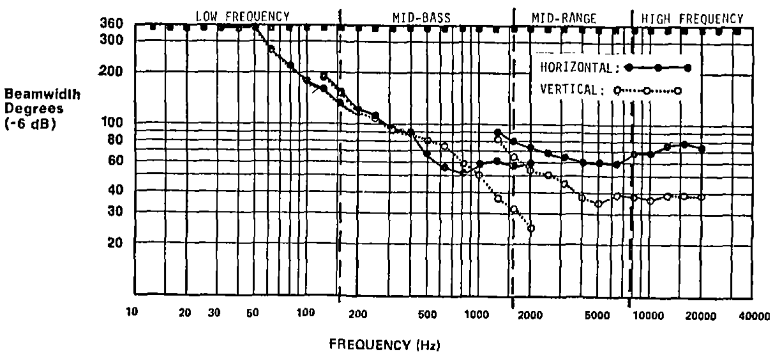


Figure 21. Beamwidth of MT-4 Loudspeaker System.