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Perkins Electro-Acoustic Research Laboratory, Inc. ❦ Engineering and Intuition Serving the Soul of Music

The Perkins Precision Reference Loudspeaker

IN 1974 PEARL UNDERTOOK the development of what was intended to become an exceptional loudspeaker. From the outset we realized that long-term survival in the wilds of the marketplace meant that we had to build a speaker that was truly special—a classic.

Best intentions aside, the design of a high-performance speaker suited to the domestic environment presents a tremendous challenge. The finished product must fit into a normally furnished environment without dominating the room and its available range of finish must present a sufficiently broad scope of options to suit the tone of various decors.

While the speaker must present itself in a visually pleasing manner, the design is bounded from the beginning by the laws of acoustical-wave propagation. Mother Nature is not about to bend the rules for any speaker designer much less redefine them yearly in lock-step conformance to the latest trends in the world of fashionable interior design.

Many difficult problems in diverse areas of scientific and artistic endeavour must be faced and resolved. The designer is required to have pertinent, working-knowledge in the fields of electronics, mechanics, acoustics, vibration, physics, and materials science. He must develop a musical ear as well as a keen intuitive sense for the things he hears and he must learn to relate his measurements in the lab to the things he hears in the listening room and vice-versa. He must either be, become or hire a woodworker capable of piano-grade construction, a competent industrial/domestic designer and finally, a business man sufficiently self-possessed to divine and act on the real needs of his customers. All this must be accomplished amidst the profit-driven clamour of an intensely competitive marketplace!

Although these requirements may appear to be eased in larger organizations, there is no substitute for competence in a wide variety of disciplines.

By our deliberate choice PEARL has remained a small company and has focused its efforts on the development of only a few outstanding products. We first developed our present loudspeaker and over the last few years have been engaged in the creation of a series of companion tube-type electronics that seamlessly integrate to form a *sympatico* system.

Through its 15 year developmental history we have refined the PR Two to an unprecedented level of design sophistication and musical elegance. Truly non-fatiguing musical performance and real value for money have always ranked at the top of the design and producablilty goals. Determined to remain faithful to these ideals, we began serious loudspeaker development in 1975.

As a starting point, we chose a generic three-way design using cone-type bass and mid-range units and a conventional soft-dome tweeter and relatively conventional 18dB/octave crossover sections. Much effort produced several prototype designs that were acceptable by the standards of the day, yet we were never particularly enraptured by their musical performance. The low bass was judged to be inarticulate, the upper bass somewhat turgid and the lower mid-range marred by peculiar problems. The mid-band, upper mid-range and lower treble all suffered serious colorations while the treble proper tended towards a hot, spitty quality. In spite of these failings, our prototypes were substantially easier on the ears than the great majority of mid-seventies offerings.

Through the course of a long research project to discover the reasons for this basically disappointing performance, we discovered a myriad of interesting and sometimes surprising things:

- of the five commonly used methods of bass loading—sealed box, vented box, passive radiator, transmission line or horn loading—none provides accurate bass reproduction within the confines of the typical domestic situation
- bass driver induced excitation of the enclosure walls produces severe coloration in the two octave band centered on middle C
- the outputs of adjacent ELMAC filters (the acronym denotes the ELectro-Mechano Acoustic Composite filter formed by the combination of drive units and crossover filters) were substantially out of phase with each

other. This causes the two drive units being "crossed-over" to radiate their respective acoustic energies from points in space that are not only non-coincident but in relative motion with changes in frequency

- placement of the bass to mid-range crossover in the 400Hz. region, nearly a standard procedure, presents an almost intractable problem in terms of passive crossover network design. The action of the bass unit stimulates the enclosure walls which form a mechanical "peaking network" of non-minimum phase characteristic. This causes the acoustic phase of the bass unit's output to be effected so as to make the correct recombination of the acoustic phase of the bass and mid-range units a near impossibility. The results of a research program carried out by Mssrs. Bang and Olufsen of Denmark show that in the 200Hz. to 600Hz. region the hearing system has its greatest sensitivity to phase anomalies of the sort commonly created by crossover networks. Seemingly the ear is not distressed by certain distortion forms but improper phase summation through this frequency band is not numbered among the forgiven. For this reason the PR Two is a two-way loudspeaker.
- high levels of coloration are produced when the upper frequency reproduction is handled by a cone/dome combination crossed over in the 3kHz. region. Investigation revealed several problems. All cone mid-ranges seen to date operate in an essentially uncontrolled "cone-breakup" mode above 1.0–1.5kHz. As a body, the motion of the cone is not under the direct control of the voice coil assembly.

  Conventional dome tweeters almost universally employ a sealed rear-cavity loading. Such methodology provides only minimal rear-wave damping thereby allowing a considerable amount of the energy radiated from the back of the dome to "talk through" the diaphragm for a period of time determined by the acoustic transparency of the tweeter dome and the marginal effectiveness of the small volume of damping material placed behind the dome.
- drive units still contribute audible information when their outputs have been rolled down 30dB by the action of the crossover. For this reason, it was decided to consider the effective *operating bandwidth* of a drive unit to lie within the –30dB points of its "crossed-over" frequency response.
- if a tweeter is mounted so as to vibrate in sympathy with an enclosure panel, significant amounts of upper-frequency intermodulation distortion are generated. IM distortion is among the most fatiguing and amusical of all distortion forms.
- the creation of a believable stereo image is severely impaired by the spurious sources of radiation formed when sound waves encounter abrupt discontinuities on the surfaces across which they travel. The most commonly experienced of these discontinuities are sharp cabinet-edges or grille-frame recesses.
- output level adjustments on the order of .2dB (200 millibels) produce marked changes in musical balance if they appear over a band as narrow as two octaves. Normal piece-to-piece drive unit variations produce irregularities in frequency response that are five to ten times that amount. This means that the unit-to-unit sound of untested production speakers can vary widely and very accurate drive-unit assembly and matching techniques must be implemented to correct for this.
- ultimately, we identified over fifty resonant mechanisms—acoustical, mechanical and electrical—that can act in a typical "high-fidelity" speaker. Some of these are plainly heard while others are of such a subtle nature that their contributions are not apparent until more significant sources of spurious output are identified and reduced in level. Progress is always marked by the discovery of new, previously buried problems.

Although the PR Two embodies several innovative solutions to decades-old problems in loudspeaker design, discoveries were only sometimes of the, "My God!, Come and listen to *this!*" variety. More usually, our understanding grew in small steps and as we steadily refined the speaker we found there was virtually no alteration we could make that was not audible.

Contrary to what the ad-copy writer would sometimes have you believe, there is no such thing as a breakthrough technology that works straight off the design table. Those products embodying genuinely innovative solutions are invariably the result of much diligent and expensive developmental effort.

So it is with the present incarnation of the PR Two. We first invested 5 years in the development of the basic technology then lavished another decade of R&D on this exceptional loudspeaker. Every detail, no matter how small, has been worked and reworked until we were satisfied. Our faithful dealers repeatedly accused us of meddling with a good thing as we developed one revision after another. The point being, we knew what we wanted and we weren't about to stop short of the mark. The gorgeous woodwork is one of the more readily apparent examples of our thoroughgoing dedication.

Rather than settle on a conventional hardwood such as walnut or oak, we chose koa for the standard finish. A seldom used species from Hawaii, koa's figure and color show splendid diversity from log to log while its density and toughness—*koa* means rough and tough in Hawaiian—make it ideal for loudspeaker applications. Being somewhat oily, koa is very difficult to sand or polish and is easily burned by normal finishing machines. We designed and built a multi-speed sander and a special, variable-speed polishing lathe that enable us to efficently achieve the silky smooth wood surfaces upon which high-grade finishing work depends.

Over the years, we developed a lengthy three-layer finishing process that involves seven separate stages. The wood surfaces are first oiled with a special mixture that brings up their natural color and depth of figure. The oil is allowed to dry and harden, then several coats of catalyzed lacquer are applied and these are likewise given ample time to harden. These initial stages are succeeded by a series of machine-aided polishing and hand waxing operations. Top quality oil, lacquer and carnauba wax are used and every step in this process has been indulgently refined to produce a surface like no other.

The result is an enduring finish of incredible beauty. Looking at it, your eye is drawn into the depths of koa's luminous, richly varying figure and fine, gold-auburn colors; running a hand over it, the woodwork touches back.

Specifics of the Design

In the following sections the PR Two is described in detail. Our solutions to the problems described in the introduction are presented along with much useful information and pertinent supporting data.

1. The Stand

The loudspeaker sits on a pedestal as shown in Fig. 1. The rectangular foot-platen is made of Koa and is fixed to a fabric covered, sand-filled riser block that is 7½″ (18cm.) high. The stand's construction incorporates a pivot mechanism that allows the speaker to be tilted slightly forwards or significantly backwards. This enables the listener to adjust the plane of correct acoustical summation of the drive units' outputs—the listening plane—to ear level. There is enough range of adjustment to accommodate a wide range of seated listening-height and -distance. A single jackscrew, accessible from the top of the riser block, provides the means of adjustment.

Four individually adjustable feet allow the speaker to be placed so that it sits squarely on slightly uneven floor surfaces. Where they contact the floor, the feet are approximately ¾″ in diameter allowing the speaker to be set on finished hardwood floors without fear of damaging the floor surface. If the speaker is to be placed on a concrete floor, the feet can be unscrewed from the bolts that pass through the foot platen from the top. Those bolts will then be seen to protrude from the bottom of the foot platen approximately ½″.

We do not provide, nor do we recommend the use of spiked feet for some very good reasons. The best being that the PR Two—which weighs 95 lbs—will, if spiked onto a conventional joist-construction floor, set the the floor and walls of the listening room into lower-band resonant modes, the negative effects of which must be heard to be believed.

The use of spiked feet is intended to reduce the speaker enclosure's bodily rocking to and fro in response to the fro and to motions of the bass unit's cone/voice coil assembly. While this action/reaction is real (Newton's observations) and can have a very deleterious effect on the sound produced by lightweight speakers, its effect on the PR Two is minimal. This is simply because the bass enclosure is so heavy

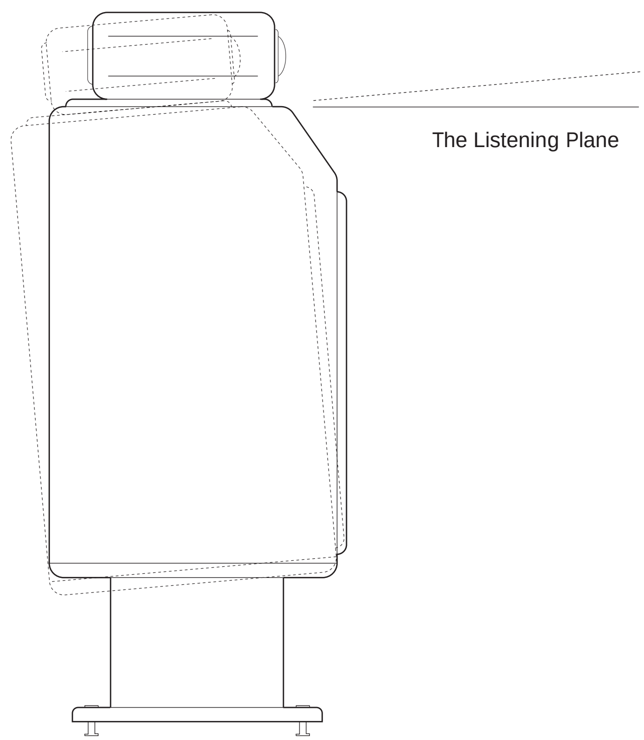

Fig. 1. The range of adjustment afforded by the PR Two's unique tilting stand is shown above. The speaker can be tilted forwards 1° or backwards 5°. By careful design of the crossover network, a plane of correct summation of the acoustical outputs of the tweeter and bass drivers is created that "lays" squarely on top of the bass enclosure and remains in place through the crossover region.

and rigid that any reactive motions it is driven to produce are tiny in comparison with the motion of the bass unit's cone/voice coil assembly. Mass and rigidity however, don't result in the dissipation of vibrational energy.

Because of the very tight mechanical coupling of all parts of the bass enclosure/stand, vibrational energy sees a very clear path to the feet upon which the stand sits. If spikes are used for feet then this energy will be transmitted into the entire structure of a joist-construction listening room, stimulating all of its surfaces into motion. The combined area of the walls of the room being many ft.^{2}, only tiny motions are required to produce significant sound pressures.†

Spurious acoustical output from the room's walls can seriously impair the perceived performance of the PR Two. The presented depth of field collapses from yards to feet, the width and height information, otherwise so superbly rendered, are heard in spatially truncated form. Dynamics are compressed while the entire lower half of the spectrum can become turgid and ill defined with a substantial loss in low-bass information. The upper octaves can sound recessed, forward, glassy or "phasey." In short, the speaker doesn't sound very good.

The PR Two is not alone in this regard, any high-performance speaker weighing more than about 15 lbs. will create a similar set of sonic problems when spiked onto a suspended, joist-construction floor.

Room-wall problems not withstanding, the enclosure-rocking issue needed to be addressed by some means as it can cause a slight but significant loss in the low bass output from the PR Two. A little research resulted in the "carpet compressing foot" now used. See Fig. 2.

Briefly, the interaction between the mass of the speaker and the compliance (springiness) of the combination of good carpet and firm underlay produces a mechanical low-pass filter that allows the speaker to "sit" solidly on the floor at low frequencies while progressively isolating it from the floor against increasing frequency.

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The foregoing is the first part of a complete rewrite of the original PR Two literature. As time permits, more information will be added.

† An 8′ x 12′ x 20′ room has 852 ft.^{2} of combined wall and ceiling area. If this entire area moves in-phase at 100Hz. over a peak-to-peak distance of .0004″—1⁄10 the thickness of this page—a sound pressure of 82 dB at 1 meter is produced into half-space. A peak-to-peak excursion of .000004″—that's 4 *millionths* of an inch—produces a 40dB SPL into half-space at 1 meter. Into the enclosed volume of a room and/or at higher frequencies these sound pressures are substantially higher for a given excursion.

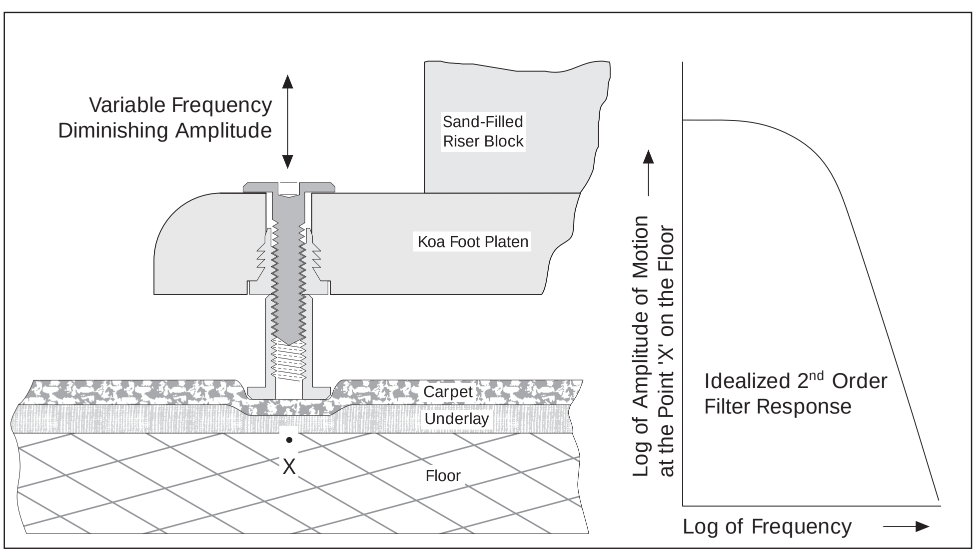


Fig. 2. The physical configuration and an idealized frequency response of the PR Two's carpet-compressing foot is shown above. At low frequencies, the stiffness of the compressed portion of carpet/underlay allows the speaker to effectively couple to the floor, thereby creating a solid footing upon which it can sit in resistance to the rocking forces applied to it. As the driving frequency rises, the mass of the speaker becomes effective in preventing rocking motions of any significant amplitude while the "springiness" of the carpet/underlay prevents the reduced-amplitude vibrations from reaching the floor.

INTRODUCTION

From our beginning in 1974 the company goal has been to produce and successfully market a moving coil loudspeaker which would provide fatigue-free musical enjoyment. The product had to be of domestically acceptable proportions, of a construction easily allowing changes in the tone of the finish to accommodate differing decor, and styled in a manner pleasing to the eye. The speaker had to give exceptional performance and value for money.

With these basic goals established, serious loudspeaker development began in 1975. At that time we chose as a starting point, a conventional three way design using cone type bass and mid-range units in conjunction with a dome tweeter. Much effort produced several prototype designs which, although acceptable by the standards of the day, were not musically satisfying.

Embarking on something of an odyssey to discover the reasons for this it was found that:

- None of the five commonly used methods of bass loading (sealed box, vented box, passive radiator, transmission line, or horn loading) offers accurate bass performance within the confines of the typical domestic situation.
- Bass driver induced excitation of the enclosure walls, however stoutly braced and/or damped, produces severe coloration in a two octave band centered on middle C.
- Placement of the bass to mid-range crossover in the 400hz region, nearly a standard procedure, presents an almost intractable problem in terms of passive crossover network design. The action of the bass unit stimulates the enclosure walls which form a mechanical 'peaking network' of non-minimum phase characteristic. This causes the acoustic phase of the bass unit's output to be effected so as to make the correct recombination of the acoustic phase of the bass and mid-range units a near impossibility.

  The results of a research program carried out by Messrs. Bang and Olufsen of Denmark show that in the 200hz to 600hz region the hearing system has its greatest sensitivity to phase anomalies of the sort commonly created by crossover networks. Although the ear seems to overlook certain distortion forms without undue distress, improper phase summation in this frequency band is not numbered among the forgiven. There is no crossover in this region in the PR Two.
- When upper frequency reproduction is handled by a cone-dome combination crossing over in the 3khz region, unacceptable levels of coloration are produced. Investigation of this difficulty indicates two main problems.

  The first is that above 1.0 to 1.5khz all cone mid-ranges seen to date are operating in an essentially uncontrolled 'cone breakup' mode, ie. the motion of the periphery of the cone is no longer under the direct control of the voice coil assembly.

  The second is that conventional dome tweeters almost universally employ the sealed cavity loading method. This procedure offers minimal rear wave damping allowing a considerable amount of energy radiated from the back of the dome to 'talk through' the diaphragm for a period of time determined by the acoustic transparency of the dome and the effectiveness of what damping material is available.
- The outputs of adjacent **ELMAC** filters (the acronym denotes the **EL**ectro-**M**echano-**A**coustic **C**omposite filter formed by the combination of drive unit and crossover filter) were substantially out of phase with each other.
- When a tweeter is mounted in such a way as to vibrate in sympathy with an enclosure panel, significant amounts of intermodulation distortion are generated. IM distortion is among the most fatiguing and amusical of all distortion forms.
- Creation of the phantom stereo image is drastically impaired by the presence of second sources of radiation formed when sound waves encounter abrupt discontinuities on the surfaces across which they travel.

As research toward solutions to these problems continued answers gradually appeared. Discoveries were seldom of the 'Eureka' variety, rather understanding grew in small steps.

One particularly illuminating discovery was that the output of a drive unit which the filter network has reduced by 30 to 50db in the crossover region is still contributing audible information. It was decided therefore to define the operating bandwidth of a drive unit as lying between the frequencies which coincide with the -30db points of the ELMAC filter's response.

Also discovered was the fact that output level adjustments in the order of .2db (200 millibels) produce emphatic changes in musical balance when applied over a several octave band.

As the PR Two grew to a state of refinement we found that there was virtually no alteration we could make which was not audible. The change to the mechanically isolated tweeter head was a literal quantum leap; high quality capacitors in the crossover network wrought amazing improvements. The change to Signal Path Green (a cable developed by us specifically for loudspeaker internal wiring runs) showed us once again that the human hearing system possesses capabilities which are seldom appreciated.

Given the almost unknowable complexity of music and given the ultimate unwillingness of a coil of wire, a piece of plastic and a magnet to produce musical sounds it is a wonder that we can experience such enjoyment from the audiophile's ménage à trois — a listener and two loudspeakers.

SPECIFICS OF THE DESIGN

In this section our present loudspeaker is described in detail. Our solutions for the problems previously described are presented along with pertinent supporting data.

In order to dispense with the least important aspects first we shall start from the bottom of the speaker and describe our way to the top.

1. THE STAND

The loudspeaker sits on a pedestal as shown in Fig. 1.1. The rectangular platen is fabricated from solid hardwood and is fixed to a fabric-covered riser piece 18cm (7") high. The stand is unique in that its construction provides the means for tilting the speaker backwards. This enables the listener to place the plane of correct drive unit summation, the listening plane, at ear level regardless of seating height or listening distance. Adjustment is accomplished by a single screw accessible from the bottom of the stand.

2. THE BASS ENCLOSURE

The low frequency enclosure is shown in Fig 1.1. Constructed of 3/4" Medite, a high density composite material, the various panels form an interlocking structure similar to a jigsaw puzzle. After the application of an adequate amount of high strength glue to all the joints, the box is placed on its side and pressed together in a purpose-built pneumatic press which exerts a total force on the box of 4550kgs. (10,000 lbs.). The box is kept under pressure for approximately 1 hour to provide ample curing time for the glue.

As shown in Fig. 4.1 the box is well braced using what we call the shelf method. This involves the placement of two dividers or shelves in the box which partition it into three acoustically definable cavities. As the intention is to break up vibrational modes in the enclosure walls, the shelves have circular openings cut into them, and are of an area equal to the piston area of the bass unit.

3. THE BASS UNIT

The bass driver is a nominal 200mm (8") dia. damped bextrene cone unit employing a long throw 37mm ($1\frac{1}{2}$") dia. voice coil wound on a high temperature KAPTON former. The coil works in a 1.1 Tesla (11,000 gauss) gap energized by a .8kg. (28oz.) ceramic magnet.

The unit uses no form of dust excluder cap mounted on the cone for two reasons:

First, due to the variabilities in the dust cap glueing procedure, unit-to-unit variation in frequency response is increased from ±.5db without a cap to ±1.2db with a cap fitted. This variation occurs in the 500hz. to 4khz. region.

Second, all dust caps tested thus far, introduce coloration in one form or another. This applies even to dust caps made of an open weave material.

4. BASS LOADING USING THE DAMPS METHOD

Of the five common bass loading methods mentioned in the introduction none offers any means of effectively lowering back EMF generation in the region of the fundamental system resonance. Rather than seeking control of the resonating elements, most of the

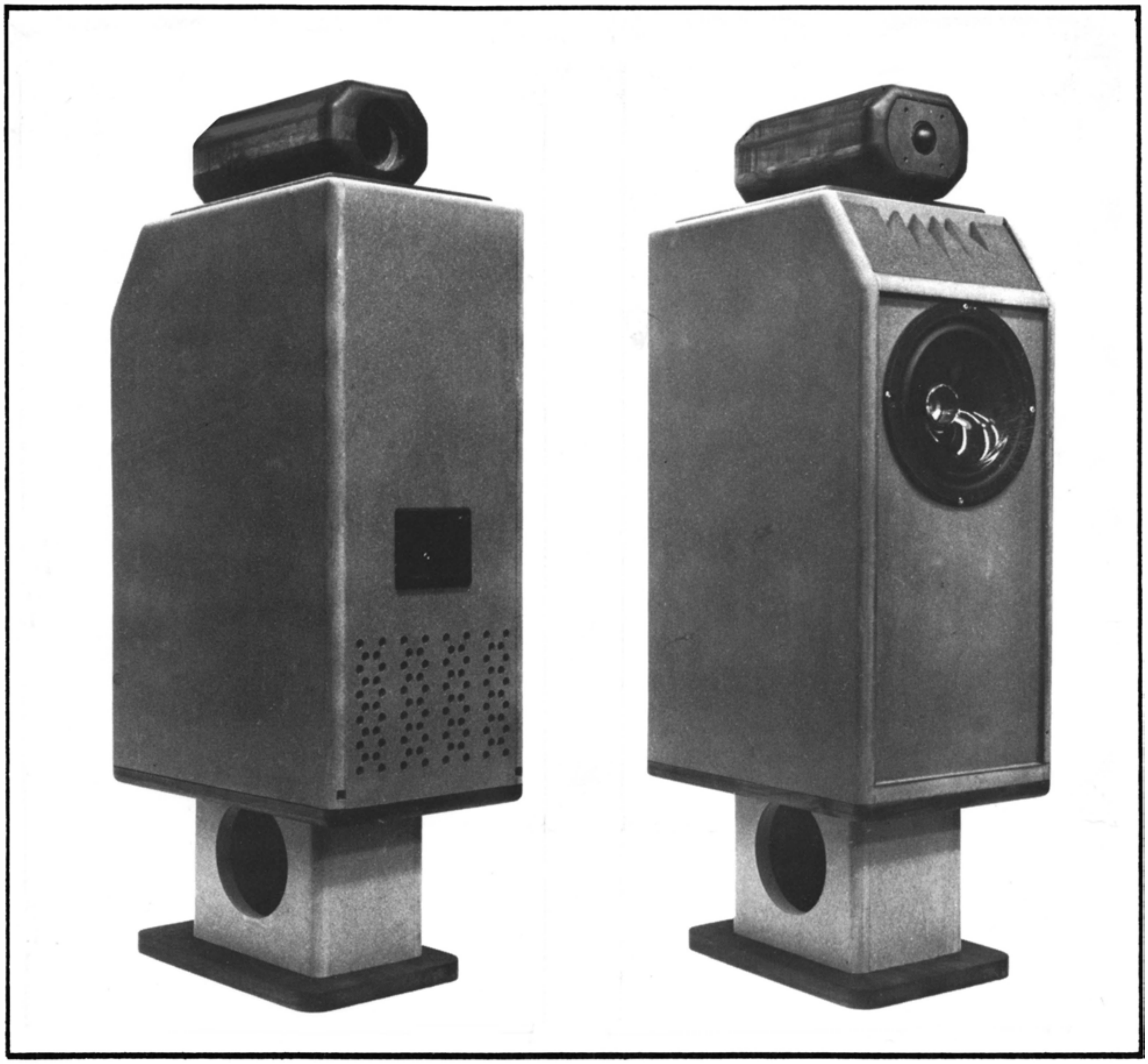


FIG. 1.1 The PR Two 'sans fini'. Clearly seen in the rear view is the distrubuted vent which forms part of the **DAMPS** bass loading system. Illustrated in the front view is an anechoic foam wedge placed so as to prevent reflections from the step in the cabinet necessary to 'time align' the output of the tweeter with that of the bass unit.
The mechanical details of the stand tilting mechanism are not shown as the exact construction is considered proprietary at this time.

FIG 3.1 The 200mm bass unit showing the large magnet structure. Note the vent holes in the back plate and in the pole piece. These vent a cavity in the magnet structure and prevent pressure build up from expelling the FERROFLUID.

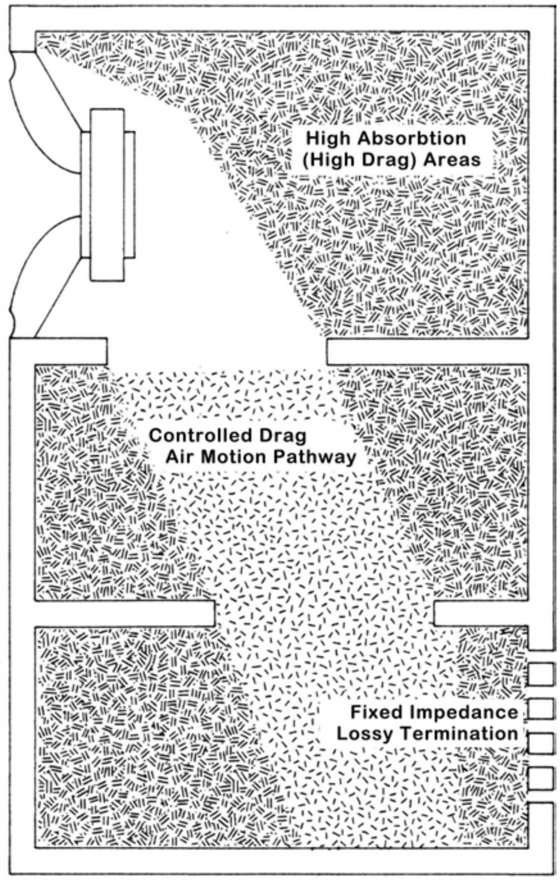


FIG. 4.1 The DAMPS loading method is schematically shown. Various materials provide the necessary acoustically absorbtive characteristics.

methods endeavor to exploit the increasing energy storage of the system at resonance by converting it to sound pressure with little regard for decay characteristics. This approach is at odds with methods employed to generate sound pressure throughout the rest of the audio spectrum. Who has ever seen a deliberately executed tweeter reflex enclosure?

It was with this in mind that we developed the **D**istributed **A**coustic i**MP**edance**S** bass loading technique.

The basic idea is to closely couple the bass diaphragm to a volume of air whose Q and aerodynamic flow impedance are closely controlled. The effect of this is the achievement of a worthwhile degree of energy dissipation through Coulomb or friction damping. As the enclosure is open to the atmosphere little pressure wave energy remains in the enclosed air volume.

Fig. 4.1 shows constructional details of the enclosure while Figs. 4.2, 4.3, and 4.4 illustrate the degree of control exerted over the resonant system.

5. THE PANEL RESONANCE PROBLEM

When a bass unit is rigidly fixed to the wall of a wooden bass enclosure and driven from a sine wave source, measurements taken on the panels themselves reveal the presence of a multitude of strong (high Q) resonances. These are due largely to the bending forces exerted on the structure by the reaction of the bass unit magnet and chassis assembly to the motion of the cone. Due to the low internal damping of all commonly used enclosure materials, energy is readily stored and slowly released by the enclosure structure. sequently a substantial amount of acoustic radiation is generated by the reaction of the enclosure wall to mechanical stimulation. Investigation reveals that in the two-octave band

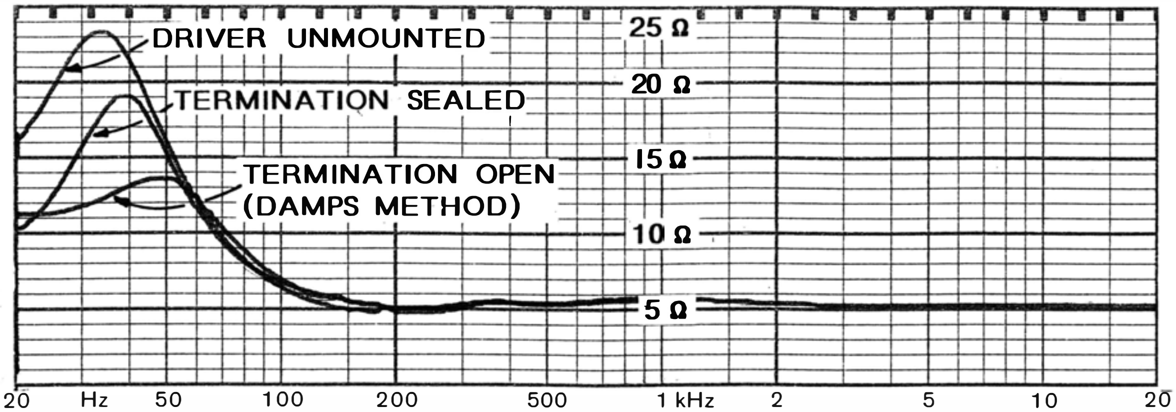


FIG 4.2 200mm bass unit impedance curves shown with the driver unmounted (measured in free air) and mounted in the 42 liter bass enclosure. In the 'termination sealed' curve the system is shown operating in the sealed box mode. The 'termination open' curve shows the DAMPS system dissipating a substantial portion of the energy otherwise stored in the reactive components of the system.

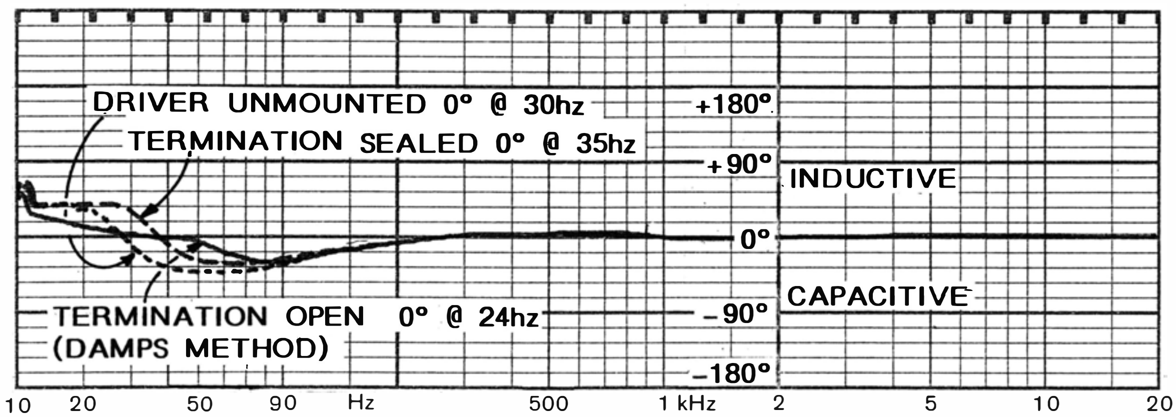


FIG 4.3 200mm bass unit electrical phase angle curves shown with the driver unmounted (measured in free air) and mounted in the 42 liter bass enclosure. For accuracy in comparsion the exact frequency of system resonance, ie. 0°, is noted for each case. The rate at which the phase angle changes from inductive to capacitive is a reliable indicator of the energy storage factor, the Q, of the system. The greater the rate of change the higher the Q. It is seen that in the 10hz. to 60hz. region, a band $2\frac{1}{2}$ octaves wide, the DAMPS system shows the least energy storage.

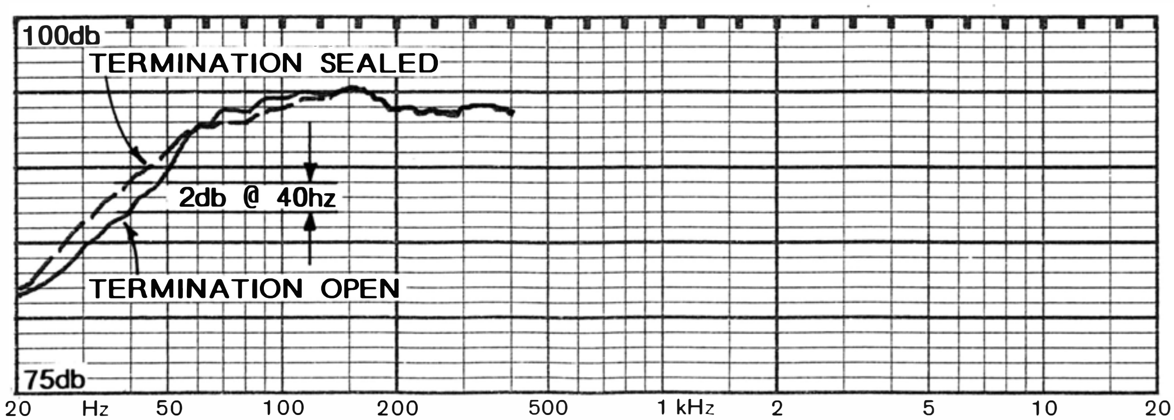


FIG 4.4 200mm bass unit frequency response when mounted in the 42 liter enclosure. As should be expected from the information given in Figs. 4.1 and 4.2 the DAMPS system shows less acoustic output over its region of operation than the sealed enclosure.

centered roughly on middle C (125hz to 500hz) the acoustic power radiated by the enclosure is often equal to that radiated by the bass unit itself.

This is an exceptionally difficult problem and one which we have researched in depth.

Presently several British manufacturers attempt to prevent the driver from exciting the enclosure walls by the placement of rubber grommets between the drive unit chassis and the baffle to which it mounts. Our work shows that while the concept of isolating the drive unit from the enclosure is perfectly valid (and one of the few solutions to the problem) the rubber grommet execution does not effectively arrest panel excitation. The compliance (springiness) of the grommet reacts with the mass of the drive unit to produce a resonant mechanical circuit in the 200hz. to 600hz. region. The exact resonant point is determined by the degree to which the grommet's compliance is effected by the crushing action of the drive unit fixing screw passing through its center. The object of the exercise being the elimination of resonances in essentially this region, this method seems almost counter-productive. However, some mechanical filtering action does come into effect above the resonant point, producing a beneficial reduction in coloration at higher frequencies.

As a result of one of the most determined research projects ever undertaken by this company, we have developed a method of virtually eliminating panel excitation. This involves the creation of a true 12db/octave mechanical filter with a turnover frequency of approximately 10hz. While this system is supremely effective, it has been deemed too costly for production.

We have opted instead to place a passive equalizer circuit in the low-pass section of the crossover to bring the frequency response into correct balance. This mid-bass correction filter does not materially effect the time domain response of the system, it is merely an amplitude correction. Fig. 5.1 details the degree of improvement in frequency response obtained.

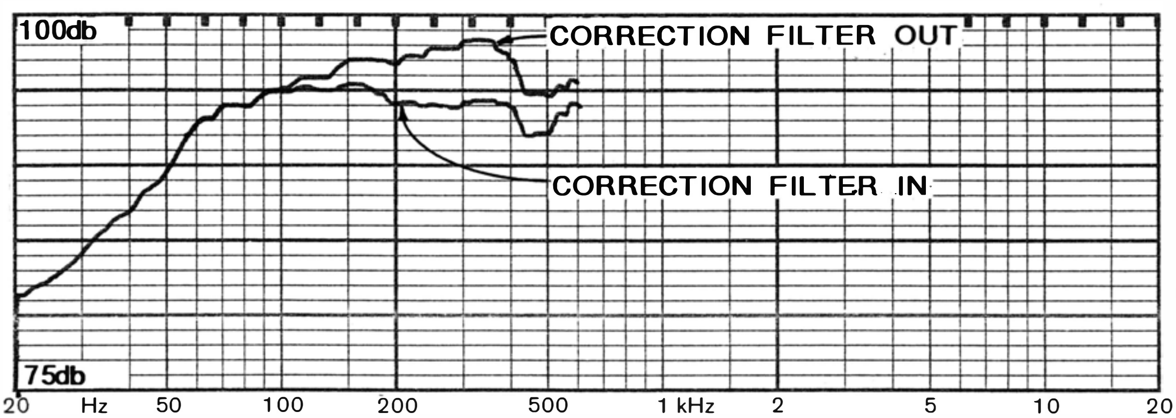


FIG. 5.1 Frequency response of the 200mm bass unit fed via the mid-bass correction filter. The three octave wide 1db dip in response starting at 150hz is created to offset the 'hang-over effect' from the uncontrolled panel motion. If the response is left flat the perceived frequency balance is adversely effected.

6. THE TWEETER, THE LOADING METHOD, AND MECHANICAL ISOLATION

The tweeter is of a construction which at this point in time is unique. Rather than being of the common sealed rear cavity type, it is rear vented so as to radiate in a di-polar fashion.

The unit is mounted in a block of solid hardwood as shown in Fig. 6.1. Bored into this block, called a tweeter head, is a tapered hole which extends the length of the piece. At the tweeter mounting end, this cavity has a cross-sectional area equal to that of the hole in the tweeter center pole piece. This area increases
hole from the tweeter end.

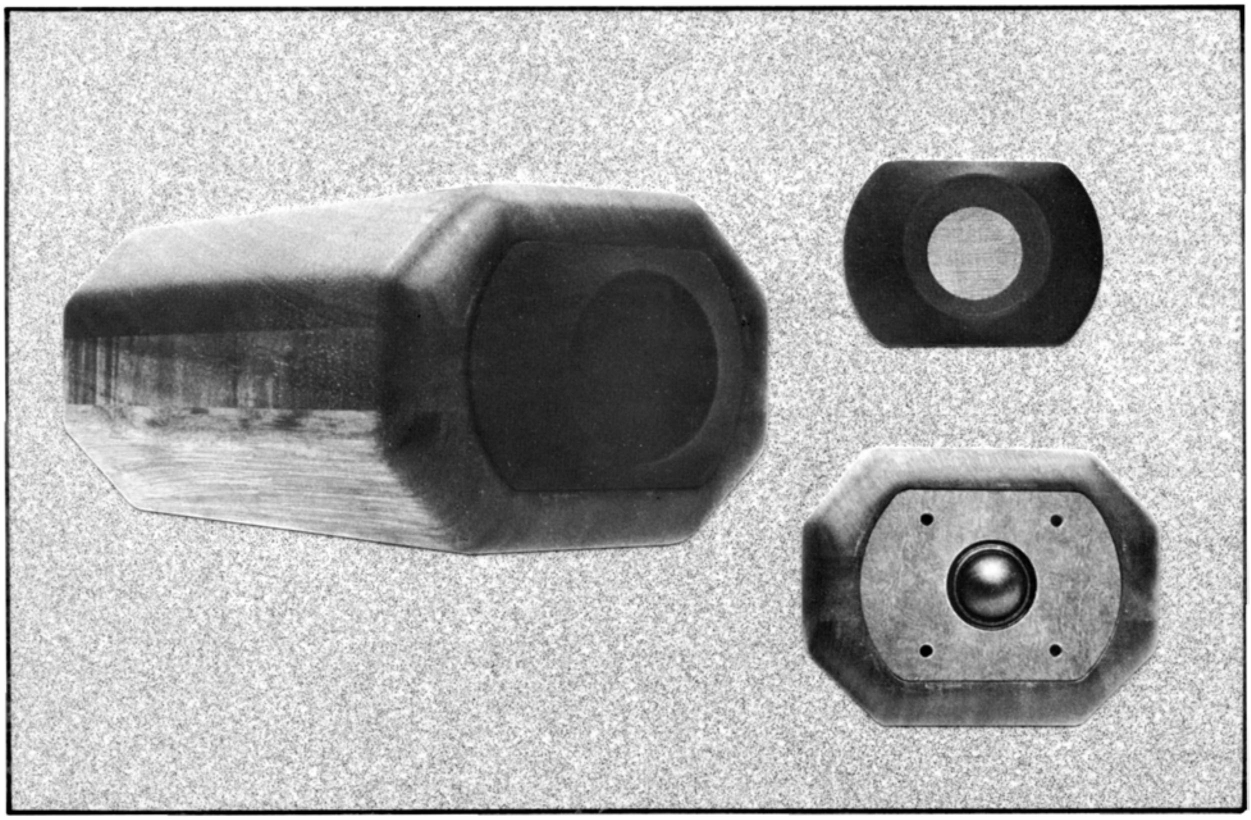


FIG 6.1 The tweeter head assembly is shown with the removable protection cover. The cover is fitted only to prevent mechanical damage to the tweeter and should be removed for serious listening.

The tweeter is a 25mm soft dome unit using a single layer voice coil wound on an aluminum former. Mechanical damping and power handling are improved through the use of FERROFLUID injected into the voice coil gap in the magnet structure. The fundamental resonance of the unit is at 750hz and is extremely well damped as shown in Figs. 6.2 and 6.3.

A further unusual construction feature is the heavy copper plating applied to all the mild steel parts of the magnet assembly. This has the effect of reducing impedance rise at high frequencies with its attendant impairment of the rise time capabilities of the unit. Sonically the copper plating has a sweetening effect on the upper registers. Transient material is more accurately rendered with percussive sounds taking a very life-like hardness. The reduction in impedance rise is shown in Fig. 6.4.

Mounted on the pole piece directly behind the dome is a piece of selected density felt which acts to absorb highly directional high frequency rear radiation which otherwise reflects off the mild steel pole piece and talks through the dome. Fitted in the hole through the pole piece is a cylinder of acoustical foam which, through aerodynamic drag, helps control the motion of the dome around the resonant frequency. Wool is placed in varying density down the length of the tapered cavity to provide the necessary match in acoustic impedance between the drive unit and the termination line formed by the cavity. See Fig. 6.5.

The improvements in performance as a result of these developments are dramatic. There is a great reduction in the amount of coloration generated by the unit. Depth information is rendered superbly. Instead of the 1 to 3 foot depth of field normally presented, there seems to be no limit to the depth of field which may be reproduced. The operating bandwidth is some 5.5 octaves compared with barely 4 octaves for a similar sealed back unit.

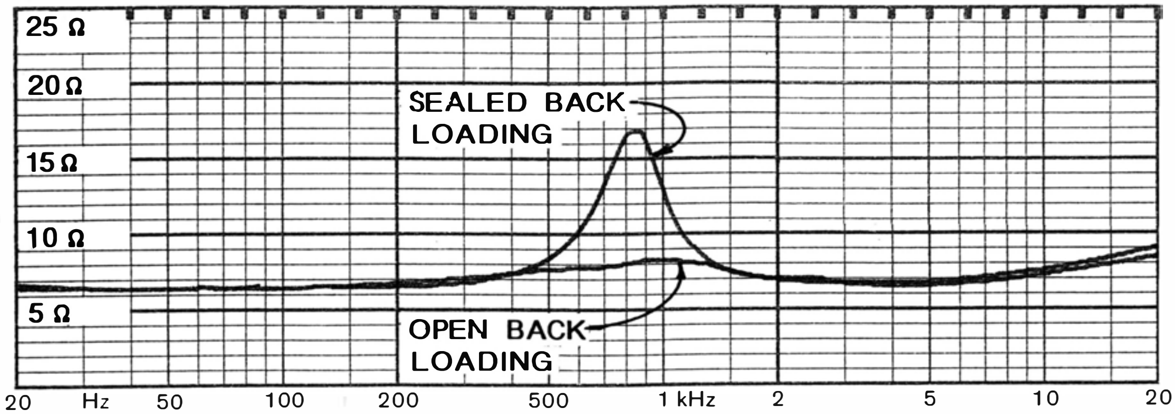


FIG 6.2 25mm tweeter impedance curves showing the higher Q of the sealed back loading method. The open back curve was taken with the tweeter mounted in the tweeter head.

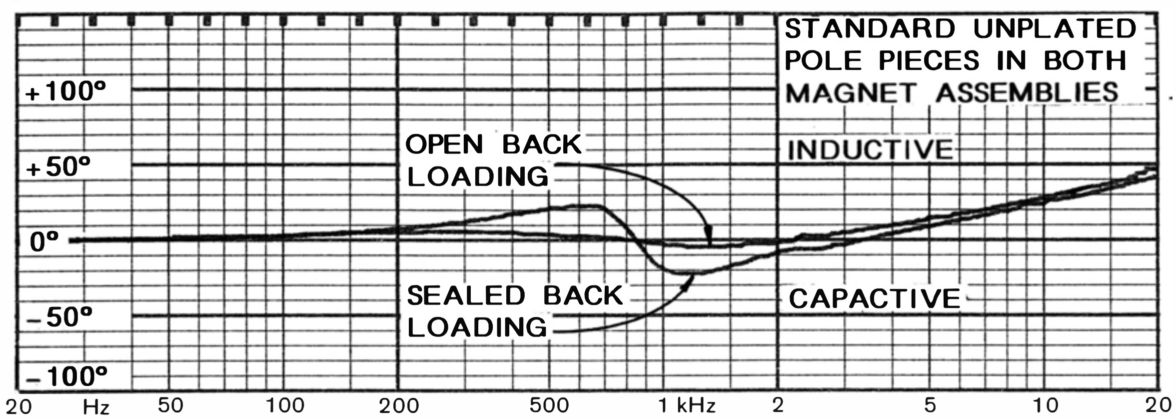


FIG 6.3 25mm tweeter electrical phase angle curves shown for the sealed back and the open back loading method. The open back curve was taken with the tweeter mounted in the tweeter head.

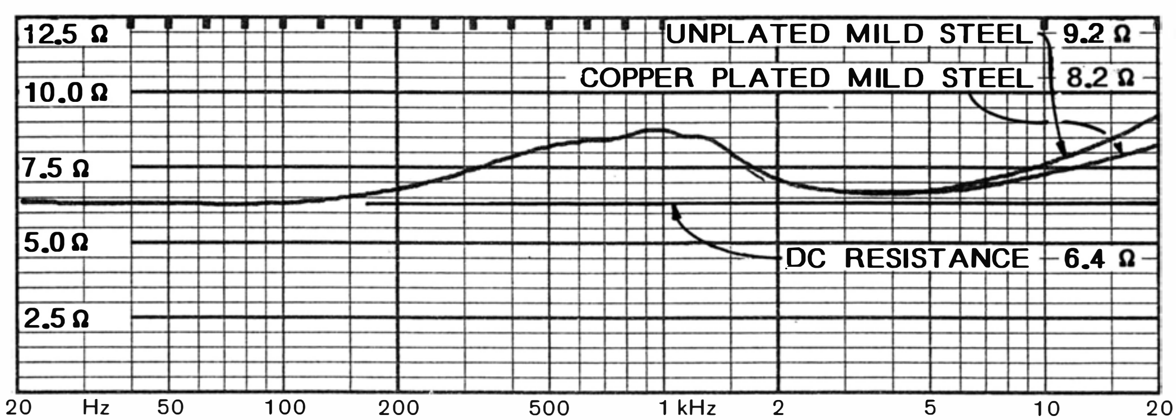


FIG 6.4 25mm tweeter impedance curves showing the inductance reducing effect of copper plating the pole pieces in the magnet assembly. For the standard mild steel piece the impedance rise due to increasing voice coil inductance is 2.8 Ω while the copper plating reduces this to 1.8 Ω, an improvement of 36% relative to the case of unplated mild steel.

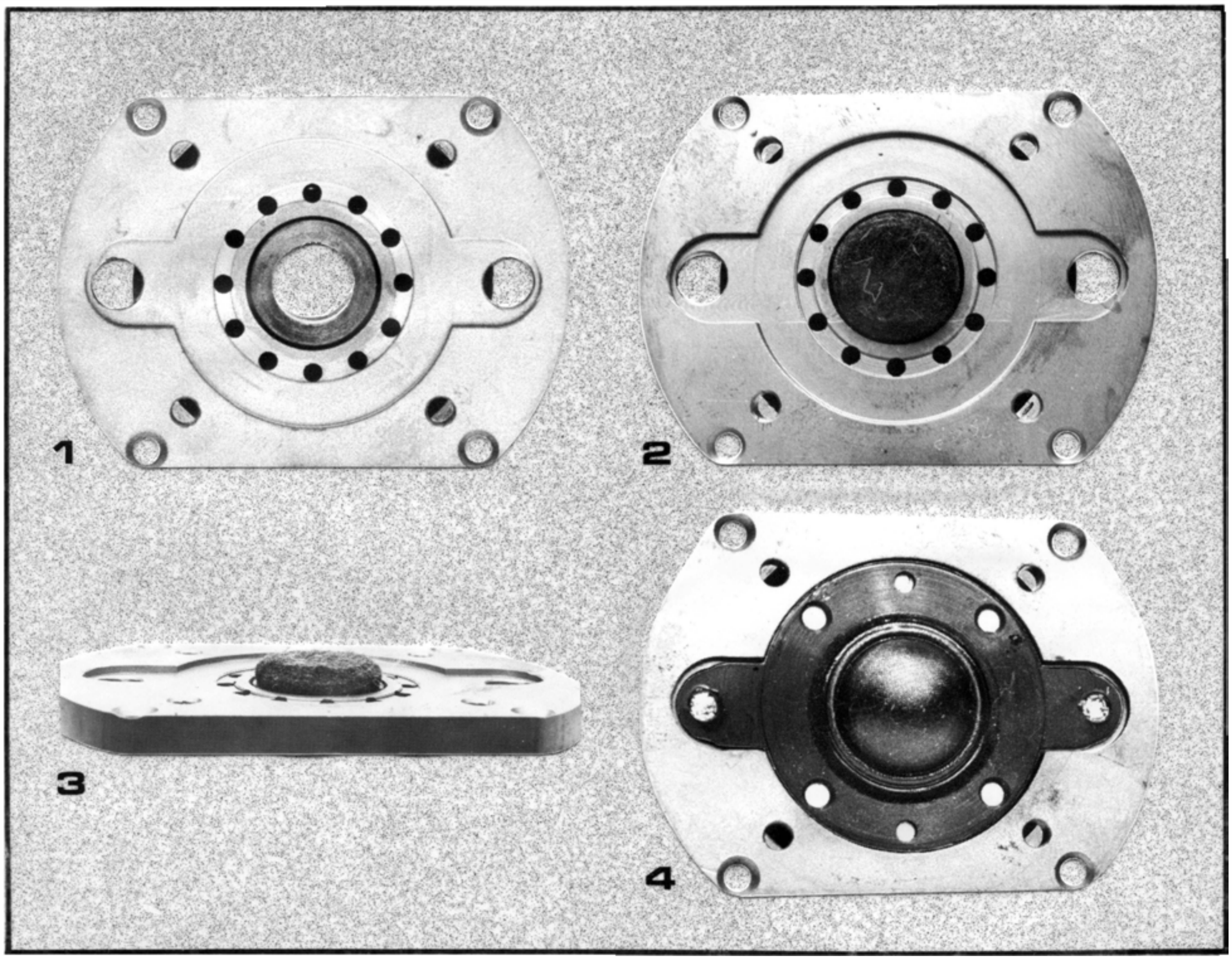


FIG 6.5 The tweeter assembly is shown in three stages of assembly. Illustration 1 shows the magnet assembly only. Note the large hole extending through the center pole piece. Illustrations 2 and 3 show the felt damping plug in place. Also seen is the milled recess in the magnet front plate. As shown in 4, this recess allows the dome/voice coil assembly to be fitted flush with the periphery of the front plate. When mounted in the tweeter head, the assembly presents no discontinuitites in the form of mounting screws or other hardware which would act as a 'secondary radiation source' because of diffraction effects. In this regard, the complete assembly is representative of the current state-of-the-art. Finally, note the vent holes behind the dome's surround. These provide the same sort of benefit as the hole though the center pole, and are similarly felt-loaded.

Whereas the conventional tweeter always presents something of an acoustic 'hot spot' the dipolar device is usually sonically invisible. To futher explain: listening to a normal tweeter, it is no difficult task to close ones eyes and concentrate on the point in space from which the high frequency sounds appear to be emanating. Once this point has been clearly established, and still with eyes shut, point to that area in space. Usually you will find you are pointing directly at the tweeter. This experiment repeated with the PR Two positioned in a good listening spot, nearly always results in pointing everywhere but at the tweeter.

We feel that the development of this tweeter represents a major advance in the low-cost, high-quality reproduction of music in the home. Before dome tweeters were introduced, a high frequency driver was usually like a small woofer in construction. It used a cone, magnet and frame assembly which were simply scaled down replicas of a low frequency unit. Being of this construction, radiation from both sides of the cone travelled freely into space. With the advent of the sealed back dome tweeter, many enthusiasts complained of a lack of openness in the treble produced by these new units despite their much touted improved dispersion pattern and potentially faster transient response.

It is this very openness we number among the most musically satisfying aspects of the performance of the PR Two.

Mentioned in the introduction is the problem arising from the motion of the entire tweeter as a function of being mounted solidly to a vibrating enclosure panel. Fig. 6.6 shows our solution which is in the form of a 12db/octave mechanical filter of 10hz turnover frequeney in the vertical plane. The arrangement to secure the isolated system to the top of the enclosure uses large headed screws through oversize holes in the platform. Tightened for transit, these are merely loosened to free the suspension, and decorative plastic caps supplied with the speaker are push fit onto the screw heads.

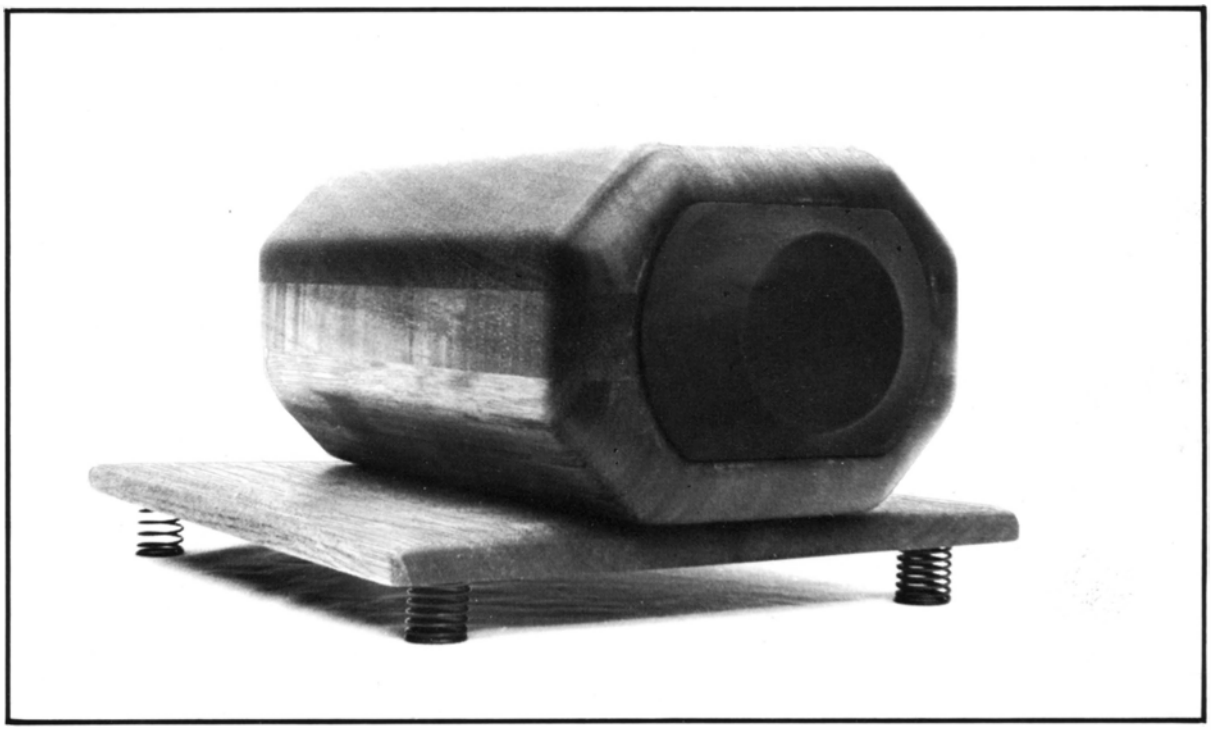


FIG 6.6 The tweeter head assembly is shown mounted on its isolated platform. Research into the causes of phantom stereo image 'defocusing' shows that when the tweeter moves as a body with a vibrating panel, an enclosure front baffle for instance, significant intermodulation distortion products are generated. The effect of these products is the appearance of non-harmonically related side bands above and below the frequency of a single note. Through this mechanism a form of 'sonic resolution interference' is set-up, no note duced without the addition of amusical mis-information. Phantom imagery suffers due to the masking effect of this distortion on the low level information present in well recorded pieces through which the ear-mind-brain recreates a sense of musical reality under artificial conditions.

Isolation of the tweeter assembly from sources of mechanical excitation results in a great reduction of this particular distortion form and allows width and depth information to be properly perceived.

The sonic result is a dynamic loudspeaker capable of detail resolution previously encountered only in the best electrostatic loudspeakers.

7. THE CROSSOVER

There are several problematic characteristics inherent in passive crossover networks which must be kept clearly in mind during the design stages.

The first and most often overlooked of these is that the passive multi-element crossover is essentially a resonant circuit whose principal damping is provided by the load ie. the drive unit. The function of any damping device is to act as an 'energy sink'. In order to perform this function well, it is necessary for the device to be completely non-reactive, a pure resistance in other words. There are few drivers which meet this specification. As a consequence the designer, if he wishes to control the performance of his network in the time domain, must be able to apply damping in the correct amount and location.

The second is that generally speaking, filter networks are designed to 'see' a constant impedance at their output port. Again, the typical dynamic driver with its fluctuating impedance does not satisfy this condition and measures must be taken to make the driver appear to the filter section as if it were a resistor. Approximating this requirement is not too difficult through the use of the well known Zobel network placed across the driver terminals. Figs. 7.1 and 7.2 show the results which can be achieved with this simple, two component network.

Further complicating the problem is the fact that one is dealing with the composite ELMAC filter described earlier. Currently in vogue among crossover designers in North America is the concept that using a first order (6db/octave) network eliminates most of the evils of passive networks and the pitfalls in their design.

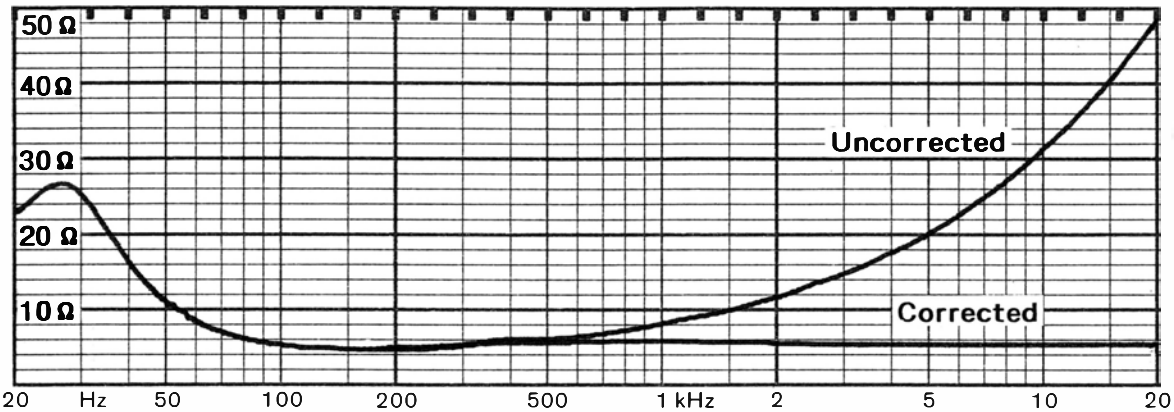


FIG 7.1 200mm bass unit impedance shown with and without Zobel network correction.

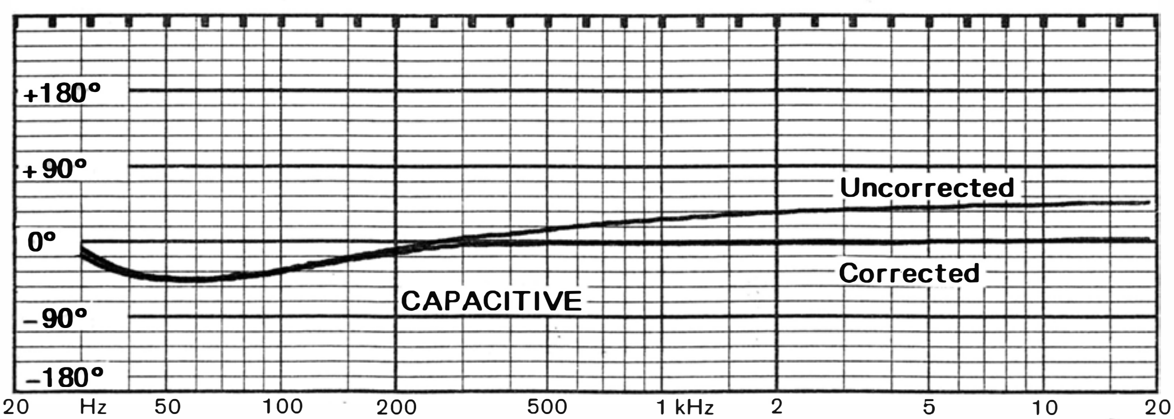


FIG 7.2 200mm bass unit electrical phase angle with and without Zobel network correction. Simply stated, a +90° phase angle corresponds to pure inductance and -90° corresponds to pure capacitance, while 0° represents pure resistance.

The basis for this thinking seems to be that because purely electrical first order filters will sum to unity in the amplitude and phase domains all that is required to achieve this highly desireable performance using real world drive units is to utilize first order networks. Unfortunately, this is not the case.

The system we are dealing with is a composite made up of two filter types. The first is the electrical filter represented by the crossover and the second, represented by the drive unit, is a much more complex filter which is an electro-mechano-acoustic composite. It is the characteristic of the net output resulting from the combination of these two which is the determinant of success in design.

To clarify the situation arising from the use of first order filter sections, it must be stated that most loudspeakers act as 'minimum phase' devices. A minimum phase device is one whose phase response closely matches that of an electrical network of identical frequency response. Because purely electrical networks can be described mathematically with good accuracy, we say that if a non-electrical filter has amplitude and phase responses which are close to those predictable for its electrical analogue, then it is acting in a manner which could be described as 'minimum phase deviation' from that which is calculated. Minimum phase for short.

To use the example of the typical sealed back tweeter, a frequency response curve will show that at some point below its fundamental resonant frequency, the rate of output roll-off approaches 12db/octave. This is the same characteristic of frequency response shown by the second order, or 12db/octave, electrical filter. Being a minim
the tweeter in question will show an acoustic phase 'lead' of 90° at resonance becoming asymptotic to 180° as frequency progresses downwards. Now when we feed this unit from the output of a first order filter with a -3db point (and 45° of phase shift), at, for instance, the fundamental resonance of the tweeter, the resultant acoustic phase lead is equal to the sum of the phase shifts of both filters, or 135°. This 135° phase shift is approximately the degree of shift created by a third order filter. Thus it becomes apparent that the use of first order electrical filters does not make first order ELMAC filters.

Sonically the result of poor phase performance is a loss in musical definition. Transient attacks are blurred, decay information is smeared, and music is rendered lifeless by the loudspeaker.

With passive crossover networks feeding moving coil drivers (also called cone and dome or dynamic drivers) there is no perfect solution to the problem of achieving flawless recombination of a musical signal split for reproduction between two or more drive units.

There is, in the form of the 24db/octave filter, a very good compromise available to the astute designer. First proposed by Siegfried Linkwitz, a chief design engineer with Hewlett-Packard, the fourth order filter allows the output of complementary filter sections to be rendered in phase but one cycle out of step. Alarming though this sounds, the one cycle time shift seems to be among the errors which pass unnoticed and do not cause listening fatigue. Our experimentation has shown that the phase errors generated by other crossover configurations are far more objectionable in long term listening.

A decision must be made early in the design stages of any loudspeaker as to the placement of the crossover point(s). In order to make this decision intelligently it is necessary to establish the range of frequency within which the speaker is expected to operate.

Considerable research has shown that the majority of musically necessary information lies within the nine octave span from 35hz to 18khz. Further, it has been shown that a gently rolling response beginning as much as an octave into the operating area is musically preferable to a flat response to the frequency extremes followed by a sharp roll-off. See Fig. 7.3.

Given bass and high frequency drivers capable of good performance over a wide operating bandwidth (o/b) the reasonable approach is to split the total o/b of the system equally between them. This places the transition point 4.5 octaves up from 35hz at 790hz.

To get a clear idea of the required driver o/b for such an arrangement, it is necessary to know the rate at which electrical input to the drivers is varying throughout the crossover region.

Due to the nature of the acoustic phase response of the drive units we have developed, the use of highly damped third order filter sections provides the required fourth order ELMAC response. These filters give an unusually gentle initial roll - off which becomes asymptotic to 18db/octave. It is this moderate initial slope which yields transient response superior to the undamped networks of similar design.

With this information reasonable approximations of the required o/b-s can be made. Fig. 7.4 provides details for the ideal crossover point placement.

In the majority of contemporary two-way loudspeakers, the crossover point is placed in the 3khz region. However, such a transition point results in the generation of unacceptable levels of coloration. It is the inability of the tweeter to provide satisfactory output over a wide o/b which dictates this rather poor compromise. See Fig 7.5.

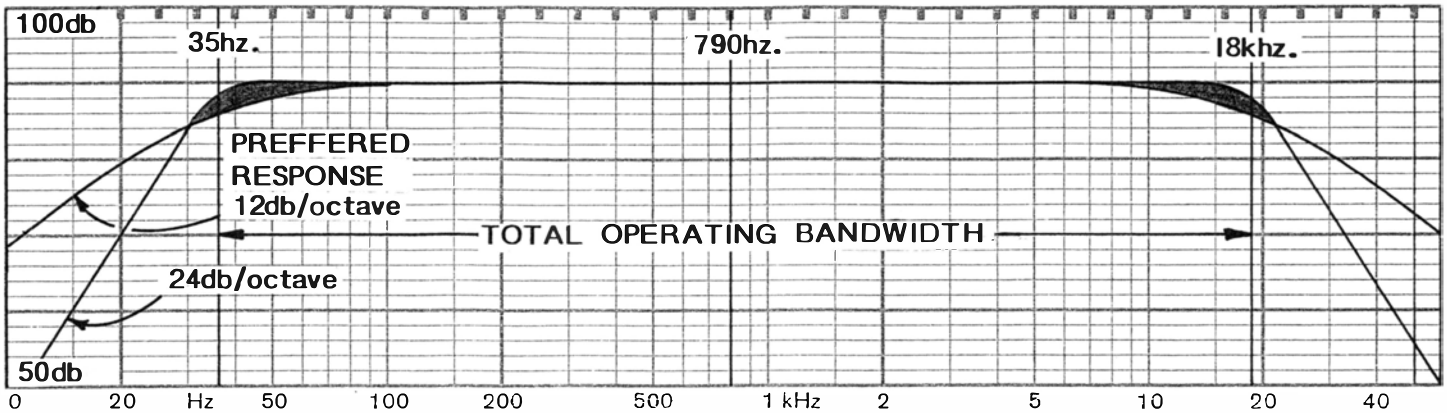


FIG 7.3 Note that outside the total operating bandwidth, the system having slower roll-off maintains higher output levels. The output in the grey areas is usually the result of resonating elements in the speaker system which by definition have poor transient response.

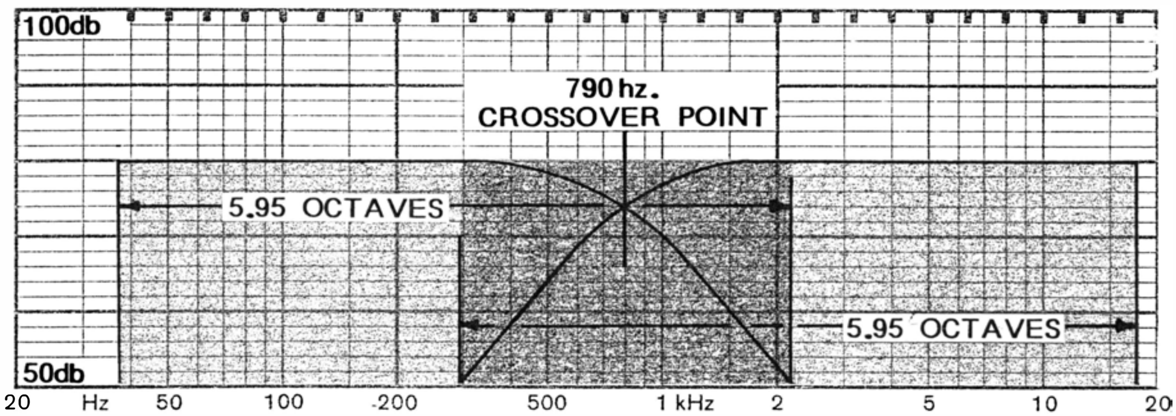


FIG 7.4 Shown in light grey is the total o/b of nine octaves with the heavily shaded area representing the crossover region. Also shown is the o/b for each drive unit when driven by complementary damped third order filters.

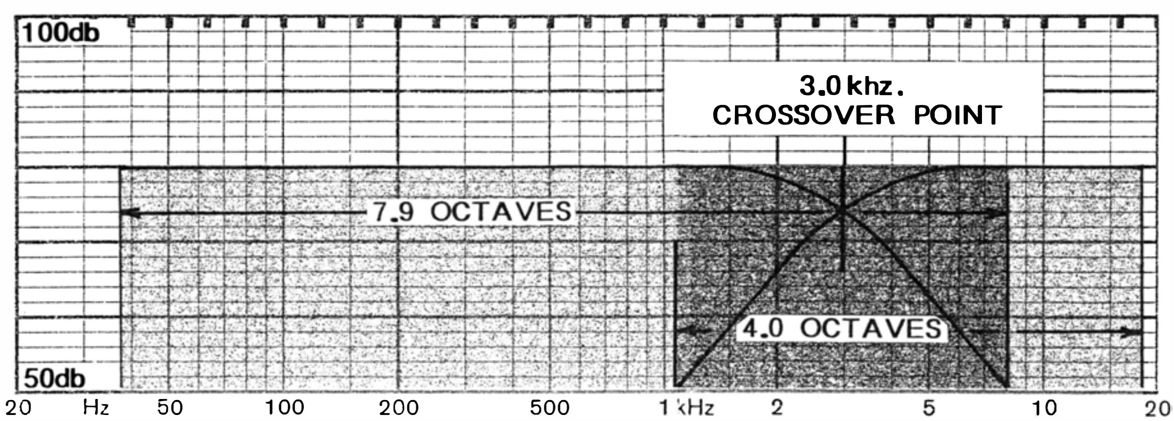


FIG 7.5 The respective o/b for a bass unit and tweeter with a crossing over at 3khz. are shown. Note the excessively wide o/b required from the bass unit.

Given that all cone units seen to date operate poorly above 1.5khz and given the unreasonably wide o/b expected of any bass-mid unit crossed over in the 3khz region, the obvious solution is to lower the crossover point as close 800hz. as possible. This requires the development of a tweeter capable of an o/b of 5 to 6 octaves. Such a unit has been developed and allows the crossover point placement shown in Fig. 7.6

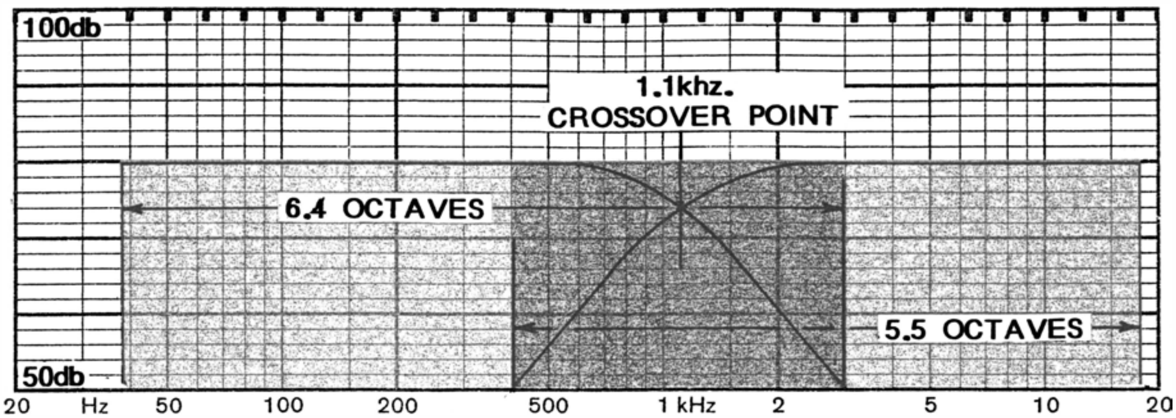


FIG. 7.6 The crossover point and the operating bandwidths used in the PR Two are shown. Note the close approximation to ideal crossover point placement.

Having selected an appropriate crossover point and electrical filter configuration, the designer must now surmount the considerable problems of achieving accurate phase summation over the crossover region in concert with musically accurate sonic balance.

This requires an iterative process involving frequency response (f/r) measurement, a round of listening assessment, then f/r correction followed by further listening to assess the value of the changes. Subsequent laboratory measurement and correction of the system's acoustic phase response completes one cycle of the process.

For a given balance the crossover elements are adjusted to correct phase errors with as little effect on frequency response as possible. Since the phase and frequency response of an electrical network are inextricably meshed, a change in one response always causes a change in the other. Thankfully there are adjustments which primarily effect phase response as well as those which principally effect frequency response.

Through adjusting f/r and correcting phase then readjusting f/r and re-correcting phase, one eventually arrives at a point of optimum system performance.

This requires a major effort on the part of any designer and we are justifiably proud of the very good phase curve which appears in Fig. 7.7. The frequency response of the PR Two is equally impressive with particulars heading the Specifications section.

So far discussion has centered on the output of the crossover to the drive unit, the output port of the network. There are however the characteristics of the input port to be considered as these determine the load seen by the driving amplifier.

Best amplifier performance is realized when the unit is terminated into a pure resistance and a goal of crossover network design is to approximate this ideal as closely as possible. Practically stated, the amplifier should 'see' a fairly constant impedance of moderate phase angle variation. Generally speaking, most present-day amplifiers of any merit will tolerate phase angles ±60° providing the angle does not swing suddenly from one extreme to the other. and impedances within the 3.5 Ω to 100 Ω region.

Impedance and electrical phase angle curves are given for the PR Two in Figs. 7.8 and 7.9.

FIG 7.7 (See top of facing page) The difference in acoustic phase angle between the outputs of the bass unit and the tweeter is shown. It should be noted that while the phase error approaches 20° at 1.1khz this is minor indeed. The majority of present-day loudspeakers show phase errors of 50° to 200° over a similarly wide band centered about a crossover frequency.

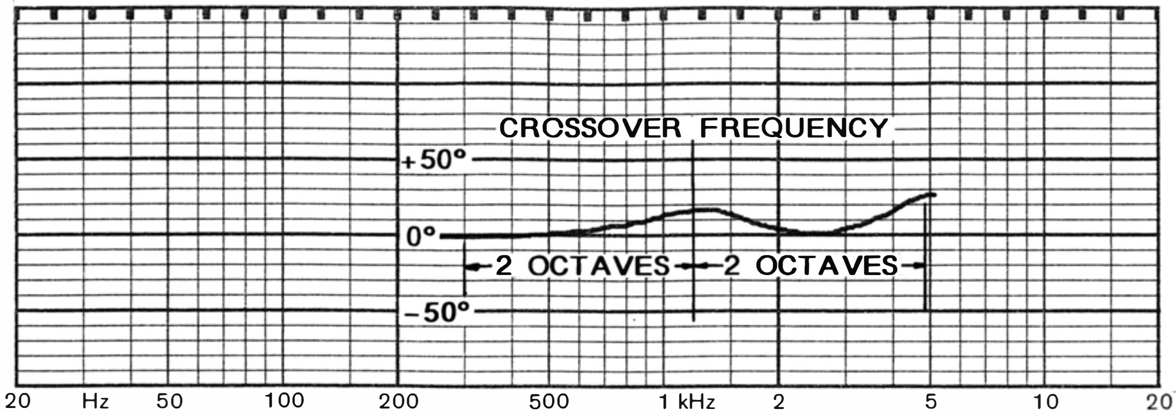


FIG 7.7 See caption on the bottom of the preceeding page.

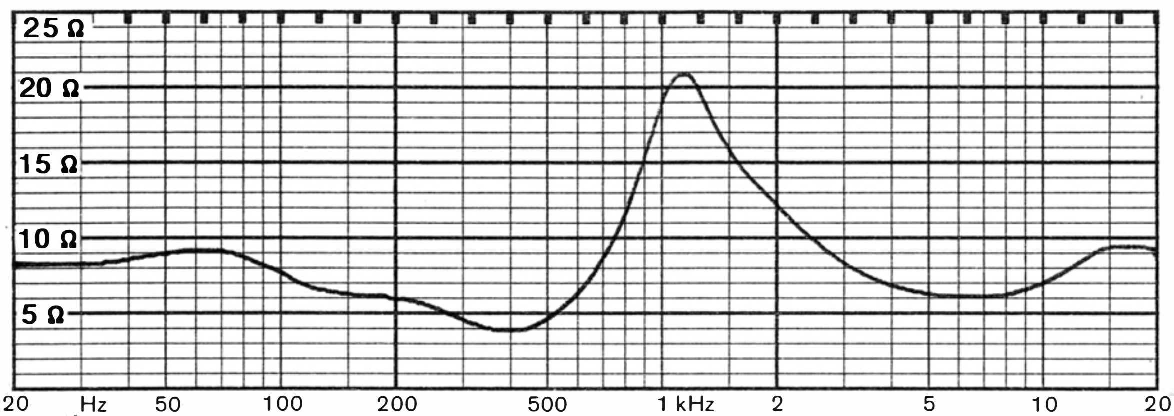


FIG 7.8 Input impedance vs. frequency for the PR TWO, Series 'C' loudspeaker.

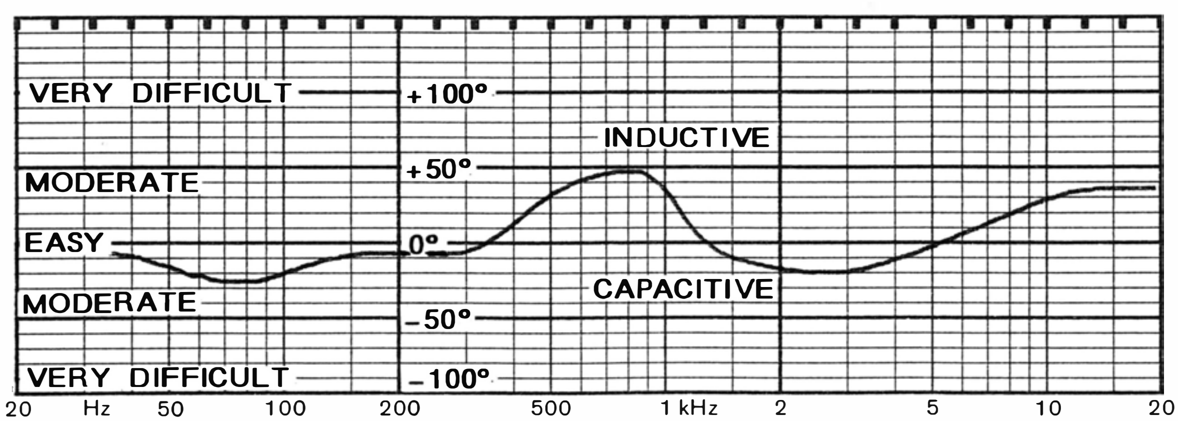


FIG 7.9 Input phase angle vs. frequency for the PR TWO, Series 'C' loudspeaker. The plot shows the degree of difficulty encountered by the driving amplifier. Highly reactive loads (those closely approaching + or - 90°) are seen to be the most difficult, while the largely resistive load (in the ± 15° range) is seen to be the least difficult.

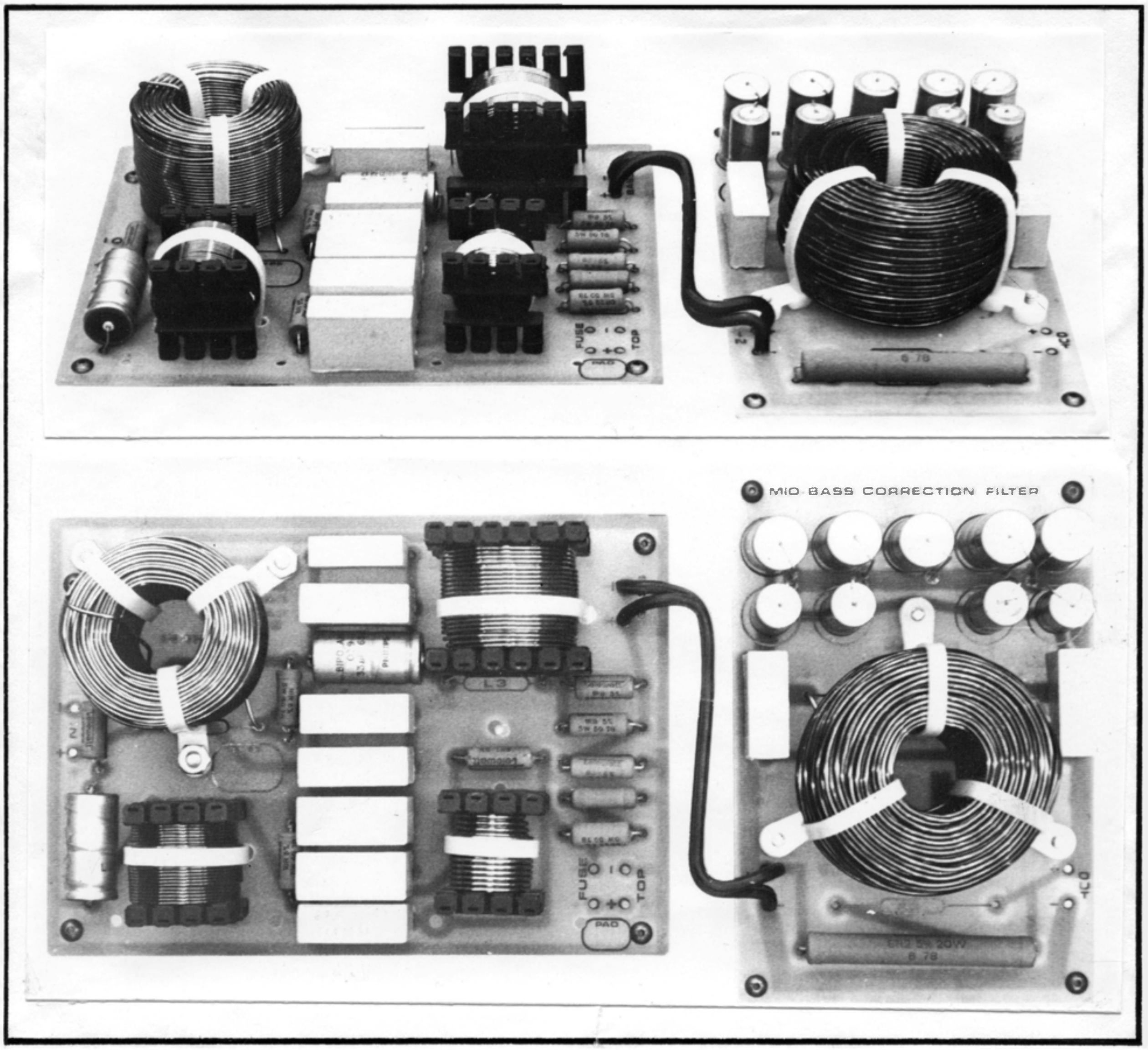


FIG 7.10 The crossover used in the PR Two. Note the use of large air core inductors and metallised polyester foil capacitors throughout. Wiring to the crossover from the input block and to the drivers from the crossover is via a special inductance cancelling cable developed by Perkins Acoustics specifically for loudspeaker internal wiring runs.

To conclude this section on the crossover, a few details of the physical construction of the unit follow:

- all inductors are of the air core type wound with #15 or larger AWG copper wire with winding tolerance of ±.5%.
- all capacitors appearing in the signal path are of a metallised foil type and self-healing in the case of moderate over voltage conditions. These are sorted into ±1% batches and all capacitance values in the network must fall within this limit.
- each crossover is individually adjusted to provide an acoustic output response variation of no more than ±1/4db and ±5° variation as compared with a reference standard.

8. CROSSOVER/DRIVE UNIT MATCHING

Through the foregoing sections, the construction of a loudspeaker giving very good musical balance and presenting an accurate, stable stereo image has been described. The creation of a hand built prototype however does not ensure that subsequent production copies will yield the desired performance.

The ear is very sensitive to small errors in musical balance, particularly those extending over a several octave bandwidth. Further compliciting matters, the ear is even more sensitive to the relative variations in frequency response invariably existing between the left and right speakers.

If it is desired to replicate a given balance, subsequent loudspeakers must show frequency response deviations from the prototype no greater than ±.25db measured in octave bands. While maintenance of a particular musical balance required the ±.5db adherence to an absolute standard, good stereo phantom image generation requires the relative left to right (and vice versa) frequency responses to lie within ±.25db limits measured in one third octave bands.

It is necessary therefore that all drive units used for high performance loudspeaker production be accurately measured for frequency response. While the unit-to-unit f/r variation of PR Two bass drivers is quite small (±.5db in octave bands), tweeter f/r-s present a wider variation. Consequently it is necessary to match crossovers to tweeters and to adjust the crossover output so as to give a tweeter f/r falling within the stated absolute limits. This accomplished, a bass unit of suitable efficiency can be selected to give the desired total system frequency response and corresponding musical balance.

To meet the demands of stereo image generation crossover/drive unit combinations must be matched into pairs which show a relative deviation falling within ±.25db measured in 1/3 octave bands. This selection completes the neccessary matching procedures.

As a final quality control all PR Two-s are subject to a listening test wherein any further fine adjustments (±.1 to .25db) required to ensure correct musical balance are concluded.

FIG 8.1 Tweeter frequency response measurement in the production anechoic chamber. All drive units used in PR Two production are subjected to rigorous inspection and measurement.

Fig. 8.2 The free-field measuring lift shown above was constructed by ourselves in 1984 and is the only such dedicated lift known to us to exist in Canada. It is presently on site at the University of Alberta's Mechanical Engineering Acoustics and Noise Unit, MEANU, in Edmonton, Alberta.

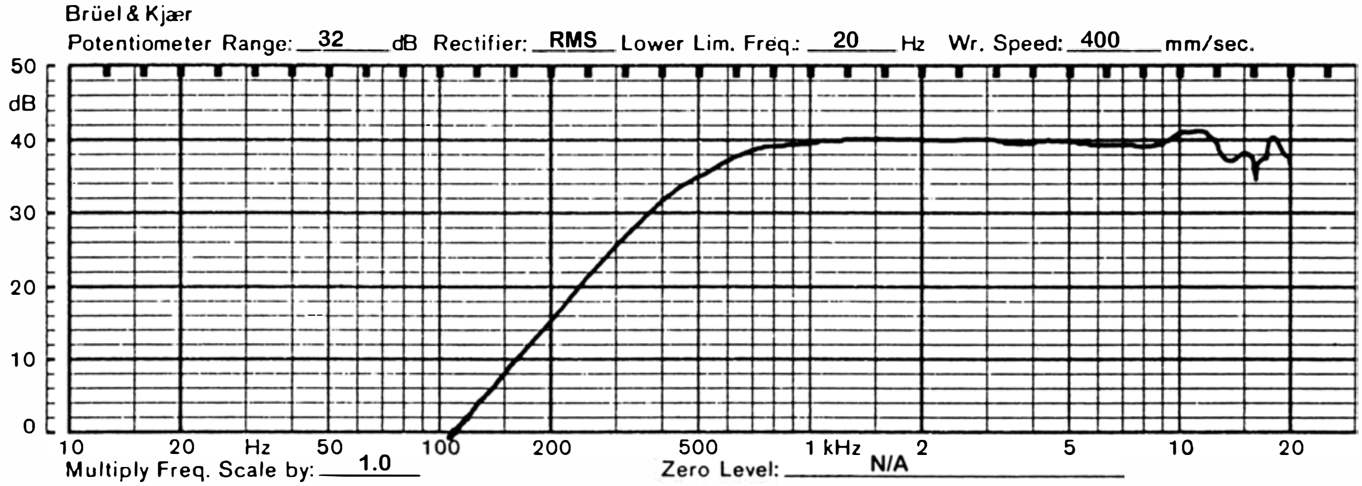


Fig. 8.3 This curve was run with a Brüel & Kjaer Type 1022 Beat Frequency Oscillator and plotted with a Brüel & Kjaer Type 2305 Level Recorder. The signal was old school swept sine with a slow paper speed, a fast writing speed and a low averaging time constant.

Conditions in the PEARL designed and built anechoic chamber seen in Fig. 8.1 were anechoic above approximately 300Hz, the measuring microphone was a 12mm Brüel & Kjaer Type 4133 with its protective front grille removed.

The PEARL-developed, dipole-radiating soft dome tweeter was mounted in the solid koa tweeter head seen here and was driven full range at low level.

The dome itself was a mesh-fabric soft dome supplied undoped by Audax in France; I developed a three stage coating technique using a heat-cured PVC blend.

Although the frequency response is remarkable, the later polymer-graphite hard domes far, far surpassed this unit in what I came to call “Resolving Power”; both in listening and measurement. Realization of the test signal for this unique measurement resulted from careful study of Siegfried Linkwitz 1980 AES paper, “Shaped Tone-Burst Testing”.

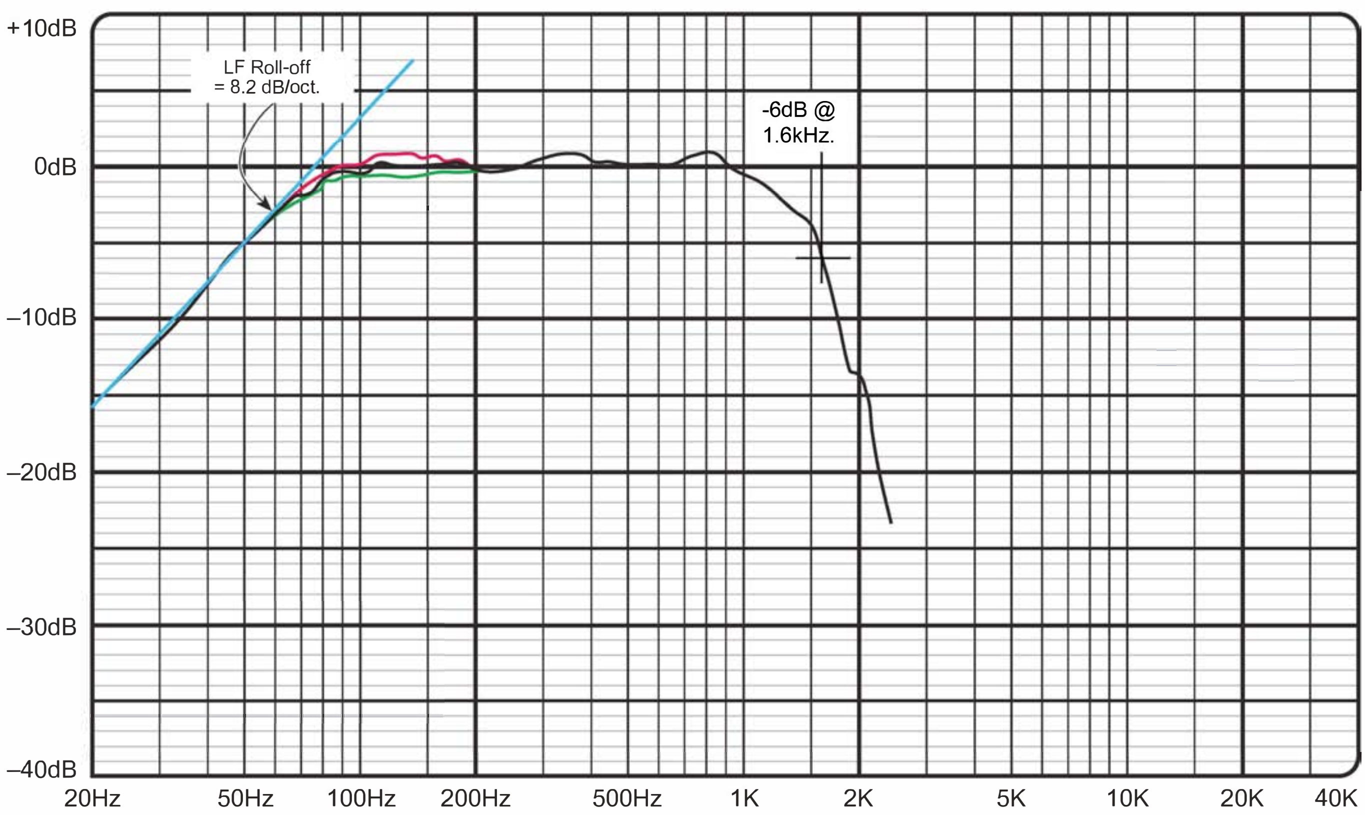


Fig. 8.4 Captured in free-space 25ft. above ground level at 1M distance using the measuring lift seen in Fig. 8.2. The enclosure loading is what I call ‘DAMPS’, complete information on which is here. The gentle rolloff rate of 8.2dB/oct is noteworthy and is due to the DAMPS loading method. The red and green curves are the ±1dB positions of a correction filter used in earlier models.

9. PLAYING MUSIC

To realize the full capacity of this exceptional loudspeaker, the following suggestions should prove helpful.

- Correct phono cartridge loading and tone arm/turntable set-up is essential for optimum functioning of the disc playback system. It is recommended that due consideration be given the numerous parameters involved.
- With any good loudspeaker the use of extreme tone control or octave band equalizer setting is neither necessary nor recommended. Parametric equalizers may be set to give sharp (narrow band) notches in frequency response to correct for room resonances in the lower registers.
- The use of pre-amplifier low frequency roll-off (sub-sonic) filters is recommended.
- The use of a low inductance type speaker cable such as 'Perkins Signal Path Green' is necessary to achieve correct sonic balance. The use of large, parallel conductor 'zip' cords is emphatically not recommended. While improving the bass definition compared with smaller wire gauge cables, all zip cables seriously impair resolution at high frequencies due to noise generation caused by strand-to-strand copper-oxide rectification effects.
- Nearly all high performance loudspeakers give their best performance when placed well away from the walls of the listening room. Approximately one quarter to one third of the way along the room diagonals is a good initial location.
- The listening plane should be at ear level at the listener's seated position. To facilitate this, there is a screw adjustment on the underside of the base which tilts the listening plane upwards when turned clockwise.
- The speakers should not be pointed directly at the listener. Rather, their axes should cross at a point in space 4.5 to 10 meters behind the listening position.
- The speakers should be placed approximately 2 meters apart.

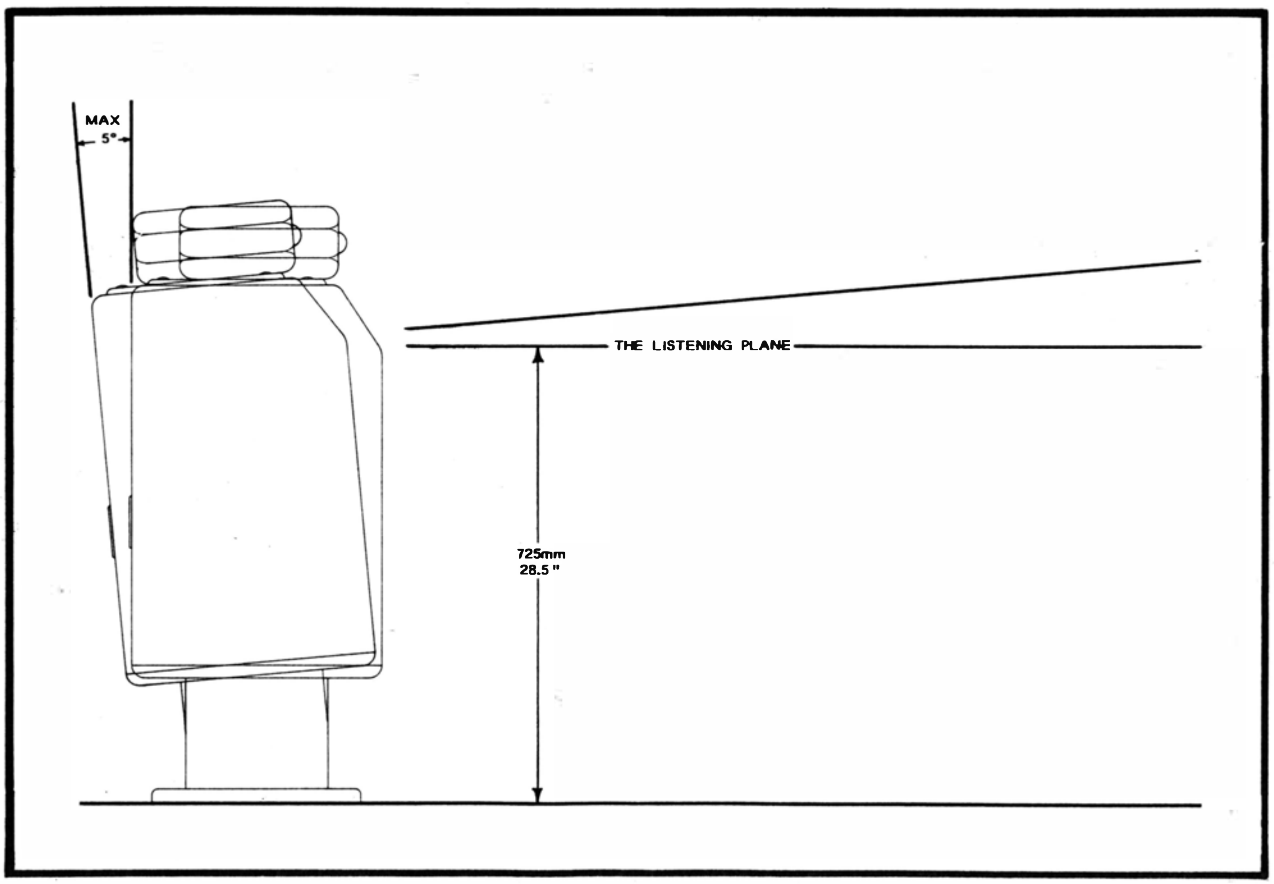


FIG 9.1 The effect on placement of the listening plane as a function of operating the tilting stand is shown.

10. SPECIFICATIONS

FREQUENCY RESPONSE

Measured in 1/3 octave bands using pink noise excitation - ±1.5db from 63hz to 16.0khz.
- minus 10db at 30hz.
- minus 4db at 21khz.

Measured in 1/1 octave bands using pink noise excitation - ±.4db 100hz to 12.8khz.

DISPERSION

Vertical - ±1db over a ±5° arc with 0° being the 'listening plane'
Horizontal - +0db, -2.5db for a ±30° arc on the listening plane with 0° being the normal radiating axis

LOW FREQUENCY SYSTEM

DAMPS bass loading method with a system resonance of approximately 25hz

MID-HIGH FREQUENCY SYSTEM

Dipolar dome tweeter with rear radiation terminated into a tapered absorbtive cavity.

DRIVE UNIT ARRANGEMENT

The drive units are placed in a vertical line. The tweeter mounting plane is set back from that of the bass unit to provide correct relative placement of the acoustic centers of the units.

THE DRIVE UNIT COMPLEMENT

BASS-MID UNIT

- 200mm nominal diameter.
- Bextrene thermo-plastic diaphragm with uniform coating of visco-elastic damping.
- Suspension - high linearity spider.
  - PVC surround 'soft terminated' to the drive unit frame.
- Voice coil - 2 layer, long throw, 37mm dia. coil wound on a high temperature KAPTON former.
- Magnet system - 2.1kg, ceramic.
  - 1.1 Tesla (11,000 gauss) gap strength.

MID-HIGH FREQUENCY UNIT

- 25mm nominal diameter.
- Close weave mono-filament fabric dome, soft setting damping uniformly applied.
- Suspension - integrated into the dome structure.
- Voice coil - 1 layer wound on an aluminum former.
- Damping and heat dissipation are improved through the use of FERROFLUID injected into the voice coil gap in the magnet assembly.
- Magnet system - .65kg, ceramic.
  - 1.0 Tesla (10,000 gauss) gap strength.
  - All mild steel parts heavily copper plated to reduce impedance rise due to voice coil inductance.
- Moving mass - 310 milligrams.

IMPEDANCE & INPUT PHASE ANGLE

See Figs. 7.8 and 7.9.

POWER HANDLING

On program material of wide dynamic range (direct discs and most 'serious music') the PR Two will accept the full unclipped output of an amplifier rated at 200 watts output into 8Ω. On electronic music, most pop music, and heavy rock music a limit of 100 watts is recommended. These limits assume that all tone controls are set to their 'flat' positions and that a sub-sonic filter is used. A minimum amplifier power of 40 watts RMS. into 8 Ω is suggested.

EFFICIENCY

At 1 watt RMS. input power the PR Two generates a sound pressure level (SPL) of 85db measured at 1 meter from the speaker grille on the listening plane.

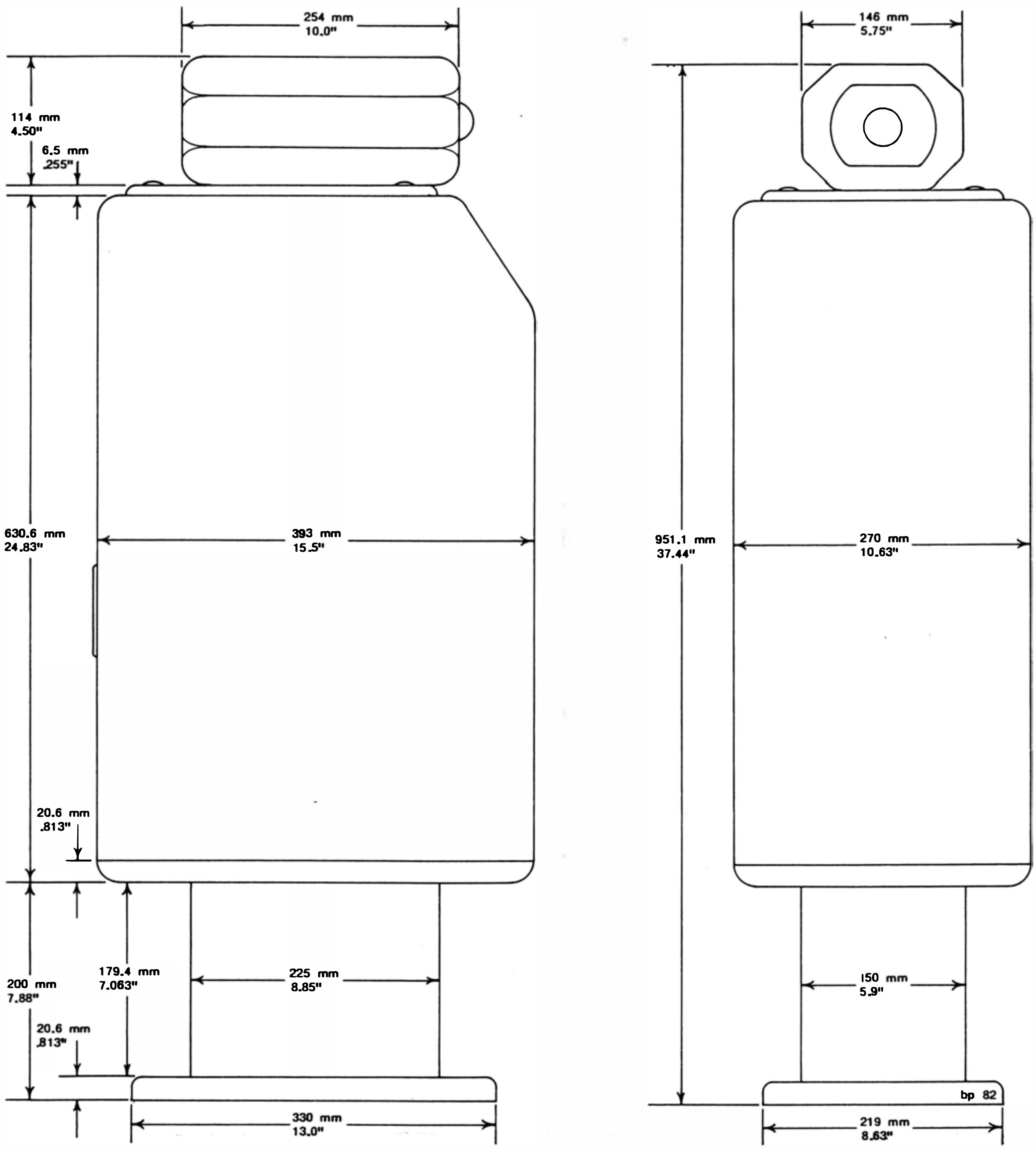
FINISH

The standard finish for the PR Two is a dark brown polyester grille material with all hardwood pieces fashioned from solid Koa (a Hawaiian species). Other hardwood finishes can be furnished at extra cost. Appalachian red oak is $65.00 extra while American walnut and Indonesian teak are $85.00 extra. Optional finishes are subject to a 6 to 8 week delay in delivery.

* *   * *   * *

All plots shown in this booklet were taken using **BRUEL & KJAER** precision microphones and acoustic analysis instrumentation. This equipment is recognized world-wide as the audio industry standard.

PERKINS
PRECISION REFERENCE LOUDSPEAKER
MODEL TWO, SERIES 'C'.



SHIPPING WEIGHT PER PAIR - 66 kgs.
- 145 lbs

PACKING CARTON DIMENSIONS - 104cm high, 46cm deep, 33.5cm wide; .16 m^3
- 41" high, 18.8" deep, 13.2" wide; 5.9 ft^3

PR Two Series 'D' Addendum

The foregoing booklet describes the Perkins PR Two, Series 'C'; we have upgraded to the Series 'D' with the following improvements:

- polypropelene capacitors of the **IAR trademarked 'Wonder Cap'** variety are used throughout the tweeter section of the crossover network.
- the bass section of the crossover network has been redesigned for better transient response, flatter input impedance and more resistive input phase angle.
- all of the inductors in the bass section of the network are wound with #15 rather than the smaller #17 AWG copper wire for lower power loss at low frequencies.
- the damping methods and materials used to absorb the rear output from the tweeter have been improved yielding a yet more open and spacious sound stage.
- the riser pedestal of the base is now sand filled resulting in a lower center of gravity and greater resistance to tipping as the result of being bumped by a child or pet.
- all internal surfaces of the bass enclosure have been damped with a dense tar based compound increasing the weight of the enclosure alone from 39# to 54#.
- a removable front is now fitted to facilitate access to both the bass unit and the crossover in the unusual circumstance that service is required.
- the hold-down screws for the spring isolation system of the tweeter head are of a more elegant style and provision is made to prevent the screws from rattling in the threaded inserts when backed out to free the head assembly on its suspension.

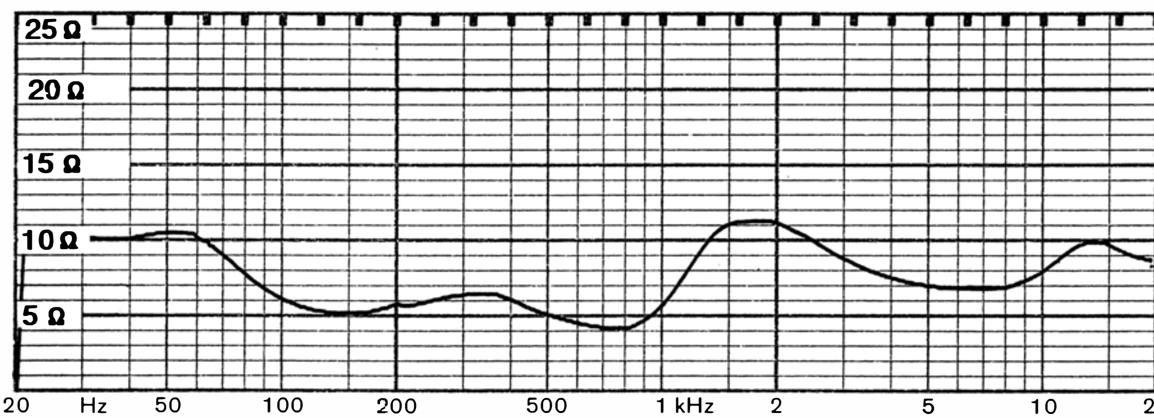


FIG A.1 Input impedance vs. frequency for the Series 'D', compare with Fig 7.8 on page 15

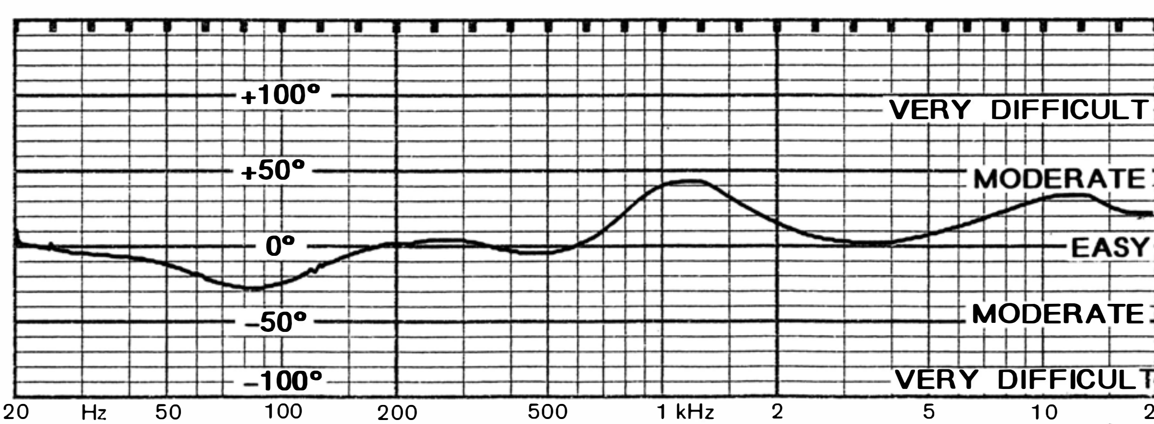


FIG A.2 Input phase angle vs. frequency for the Series 'D', compare with Fig 7.9 on page 15.

- a new printed circuit has been designed using the 'star ground' principal. In this application it eliminates the modulation of the tweeter ground by signals from the bass section of the network. The result is less IM distortion, greater upper octave openess, improved dynamic linearity and contrast.
- the entire crossover assembly is covered with a 3/4" layer of damping compound applied to the component side of the printed circuit board. This is done to prevent the assembly from generating coloration due to its otherwise undamped vibrations stimulated by sound waves within the bass enclosure.
- both the bass and the tweeter are fused and provision is made for 'solder bridging' past the fuses in critical listening situations. It is recommended that this be done only by experienced listeners as the bypassing of the fuses voids all warranty on the drive units effected.
- the tweeter is connected to the crossover with an improved low inductance wire developed specifically for tweeter circuitry.
- a die-cut felt piece now surrounds the tweeter dome on the front of the tweeter head. This replaces a non-removable plastic piece making field service of the tweeter a simple matter. Sonically, there is a reduction in the amount of high frequency 'splash' from area immediately adjacent the tweeter dome.
- a four point foot assembly has been developed and is fitted to the existing base platen of the speaker. This provides direct contact between the four support points and the floor below despite the presence of any carpeting up to 3/4" thick. The result is greater clarity in the warmth region (100hz. to 400hz.), improved low bass response and an overall improvement definition and sound stage accuracy.
- a major reduction in enclosure generated coloration has been achieved. The method is strictly proprietary.
- an equalizer circuit has been fitted which allows the frequency response in the two octave band centered at 100hz. to be tailored to suit various room acoustics. Three switch selectable positions are provided giving reponses at 100hz of +.75db; flat; -.75db. See Fig. A.3 on page 27 for free-field response curves showing the filter's effect.

PRELIMINARY SET-UP INSTRUCTIONS FOR THE PERKINS PRECISION REFERENCE LOUDSPEAKER, MODEL TWO, SERIES 'D'

- Uncarton the speaker and lay it on its side on the floor. Remove the transit block fixed between the tilting base and the enclosure. Save this piece for re-use whenever the speakers must be shipped. Also save the packing cartons. The warranty is voided by shipment of the speakers in any but these cartons.

- Note the jackscrew in the top of the riser pedestal of the base. This is the tilt adjustment about which more detail is provided further along.

- Stand the speaker upright and back out (approx. 1/2") the four large plastic headed screws which hold the tweeter assy. to the top of the enclosure. Hold the head down until all four have been loosened and then release the head to float on its spring suspension. The screws should now be adjusted for 1/16" clearance relative to the tweeter head platform.

- The speakers can now be placed in their intended listening positions. These should be approx. 3-4' from the (preferably absorbtive) rear wall and 2-3' from the side walls of the listening room. In very large rooms the speakers should probably be centered left to right approx. 7-8' apart.

- Connexion to your power amp should be via PERKINS SIGNAL PATH CLEAR speaker cable. Connect the end of the cable with the small protrusion between the red and black terminations to the power amp the other cable end has no such protrusion and obviously connects to the 5 way binding posts on the speaker. This should be done by inserting the cable end through the holes drilled through the gold plated post of the connector and then tightening with a nut driver or small wrench the plastic head of the binding post to crush the wire into firm contact.

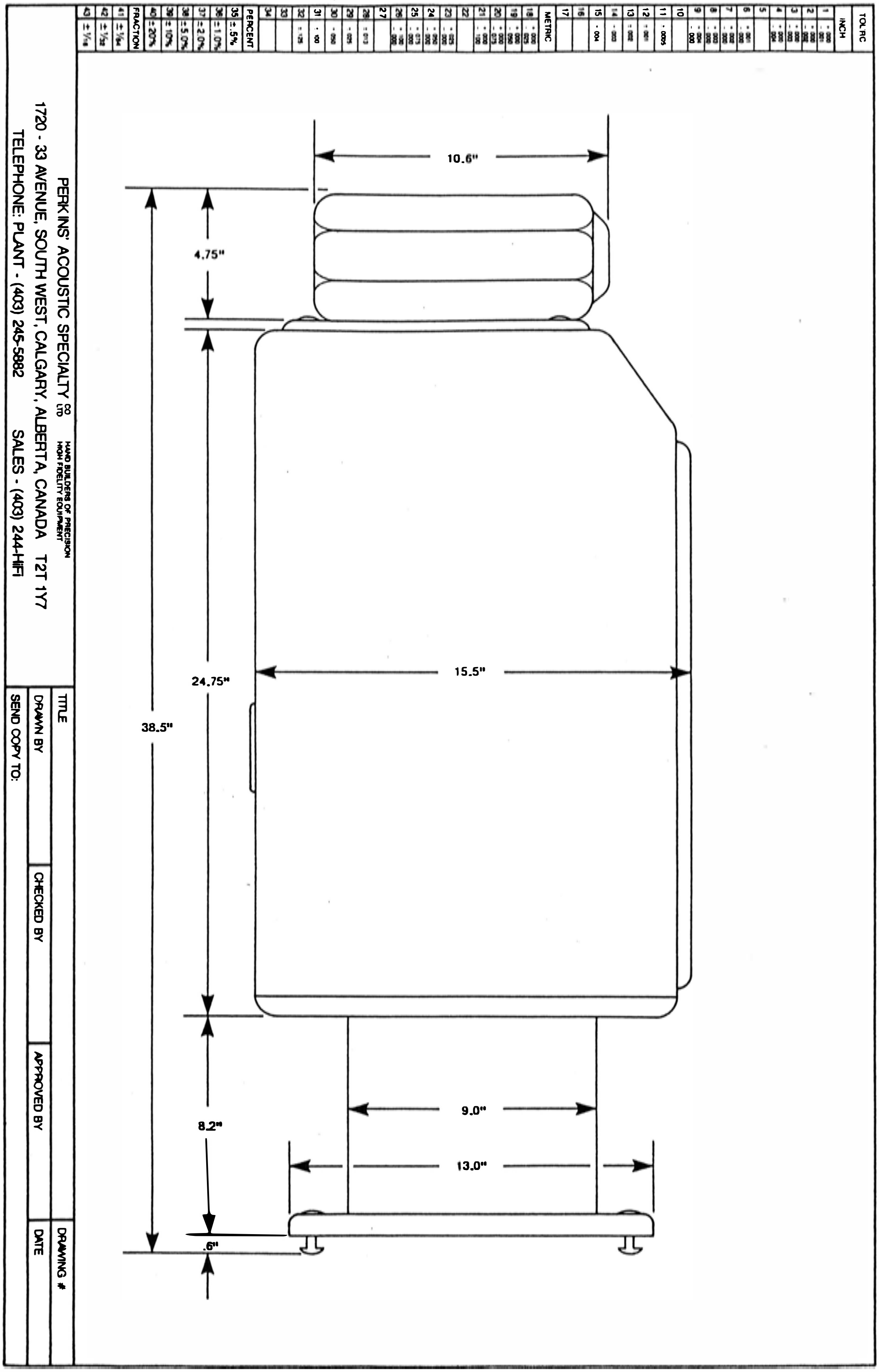
- Behind the binding terminals is a three position toggle switch through which contouring of the two octave span centered on 125hz is achieved. The three positions are as follows;

- left, -.75db
- center, +.75db
- right, flat

Note that in your room best stereo and sonic balance are not necessarily achieved by identical switch settings. The is due the vagaries in standing wave patterns due to room asymmetry caused deviations in room proportions from regular rectangularity, open doorways, adjacent hallways and open spaces, placement of large pieces of furniture, etc

- Provision has been made for the fitment of various feet to the platen of the speaker stand by way of four threaded plastic inserts pressed into each of the corners. Two foot styles are currently being developed, a true carpet piercing foot intended **only** for use on carpeted concrete floors poured directly on Mother Earth and a ball type foot for use on conventional floors of joist and beam construction. Availability should be Dec. '83

- The speaker is supplied to allow the listening plane to be varied so as to lie at the same height as your ears at whatever distance and elevation you are seated relative to the speaker. An excerpt from the technical literature on the PR Two appears overleaf and details the position of the listening plane


10.6"
4.75"
24.75"
15.5"
38.5"
8.2"
9.0"
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.6"
PERKINS' ACOUSTIC SPECIALTY CO LTD
HAND BUILDERS OF PRECISION HIGH FIDELITY EQUIPMENT
1720 - 33 AVENUE, SOUTH WEST, CALGARY, ALBERTA, CANADA T2T 1Y7
TELEPHONE: PLANT - (403) 245-5882 SALES - (403) 244-HIFI
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IAR HOTLINE! #24

CES Winter 1983

IAR Engineering Achievement Awards

Music reproduction in our homes is chiefly advanced by audio products that explore new scientific paths to truly better sound — not by new products that merely boast more cosmetic glitter, fancier convenience gimmicks, or slicker ad campaigns than last year's models.

These prestigious IAR awards, given after each CES, are bestowed in recognition of innovative product engineering that truly advances the science of the music reproducing art. IAR is perhaps uniquely qualified among publications to evaluate the technical merit of new products and new design ideas. So give the following award winners serious attention; even if you don't buy these products now, their pioneering influence will surely affect your system in the near future.

Special Merit Awards

Perkins PR Two D Loudspeaker

Though it's 'simply' a two way system, the Perkins reflects more thorough and complex engineering accomplishment than most four or five way monster 'reference' speakers. Many factors explicitly considered in this design were discovered by Perkins' original research, and have been utterly overlooked by other speaker designers.

The Perkins PR Two D is the first speaker released by this company. But it's been under development for 8 years, as Bill Perkins the designer empirically researched many new, subtle mechanical and acoustical factors that can affect the final sonic performance of a speaker system. The PR Two D is a medium size floor standing two way system, reminiscent in appearance of some B & W models with a separate tweeter enclosure on top. It's made in Canada, and the U.S. selling price is $1850 per pair.

We overheard several people at CES saying that they thought the Perkins was the best sounding speaker anywhere at the show. We wouldn't go that far based on what we heard — but the exhibit room, listening setup and program source had shortcomings when we visited the exhibit, so we surely didn't hear the speaker to full advantage. In any case, it's noteworthy that others had such a high opinion of the Perkins' sound, and this alone makes it worth your while to listen to it for yourself and decide what you think.

What impresses us as much as the system's sound is the care and thoroughness of research that went into its design. This is apparent from a comprehensive 21 page booklet you can get from Perkins. You should really write to Perkins to get one for yourself, but here are some of the highlights.

Note that most of these factors haven't even been thought of by other speaker system designers, much less optimized in their products.

The tweeter enclosure of the Perkins is floated on springs, almost as in a turntable suspension. Why? In all conventional speaker systems, the tweeter is hard mounted to the same enclosure housing the woofer and/or midrange driver. Even if the enclosure walls are made very stiff, they will still vibrate at the higher frequencies put out by the woofer and/or midrange; indeed, the stiffer the wall panels and the better they're braced, the higher the frequency of this vibration and the longer it will ring (higher Q), unless the walls also have been given drastic deadening treatment. The amplitude of these panel vibrations might be minute compared to the woofer's excursions, but it is significant compared to the tweeter's much smaller excursions. If the whole tweeter body is moved by these panel vibrations, the acoustic output from the tweeter's diaphragm will be modulated by these panel vibrations.

The unwanted direct acoustic radiation from vibrating enclosure wall panels has been recognized for many years as an evil to be combatted. But here we have an additional problem: modulation distortion of the tweeter's output, if the tweeter is hard mounted to these vibrating enclosure walls. Perkins has recognized this problem and found a solution: floating the tweeter (and its enclosure) on a suspension. As with a turntable suspension, the idea is to make the resonant frequency of this mechanical filter be below the frequency of any external vibrational stimuli. In this case, Perkins can make his main enclosure walls stiff enough so that only higher frequencies of the woofer's output reach the top of the main enclosure, and then he can merely set the tweeter suspension's filter frequency below these stimulating vibration frequencies.

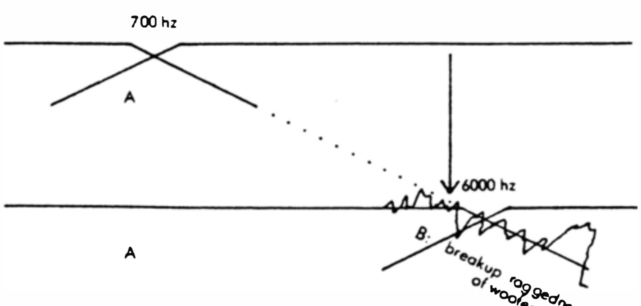
Perkins has drilled out the back of the tweeter; this eliminates the usual cavity resonance behind the dome. Perkins then resistively damps the dome's mechanical resonance and absorbs its back wave with felt and foam in a large tweeter enclosure. The graphs in the product information booklet show almost complete elimination of the tweeter's reactive peak in its impedance curve. This in turn has several benefits, including easier control of the tweeter's crossover and power handling in the critical lower region of its passband.

And this in turn helps Perkins to establish a much lower than usual crossover freqeuncy for this tweeter, 1.1 kc instead of the usual 3 to 5 kc. Then, as a result of this, the Perkins has the advantage of not using the woofer in its upper frequencies, where its cone would break up and its narrower dispersion would cause a coloration in the system's power response (see Hotline 18 for explanation). Such a low crossover frequency would impose severe power handling stress on the tweeter and hence constraints on the whole system, but for the fact that the PR Two D uses approximately fourth order (nominally Linkwitz) crossover filter slopes, with approximately third order electrical protection of the tweeter. This crossover configuration does not, however, offer flat phase (transient perfect) response (see the KEF 105 story in Hotline 6 for discussion).

Most of the tweeter assembly is custom constructed or modified by Perkins, to obtain improvements in other novel aspects. There's a specially milled front plate, with recesses to accept the tweeter's terminals and mounting plate with screws. This means that no hardware or discontinuity is projecting out into the tweeter dome's radiation to the side, where it might cause subtle diffraction problems. All the mild steel parts of the tweeter assembly are copper plated by Perkins. He reports a measured reduction of the usual tweeter impedance rise above 6 kc, and a sonic improvement in the sweetness of the trebles and the accuracy of hard transient attacks. The woofer of the Perkins has had its dust cap removed; the dust caps of most drivers are responsible for spurious radiations that give a ragged upper frequency response and peaky colorations (while the rest of the diaphragm has smoothly rolled off). Vent holes have been drilled in its pole piece too.

Other worthwhile aspects of Perkins' design have been employed previously in other speaker systems. Resistive damping, achieved both by padding inside the cabinet and by small holes drilled to vent pressure from the cabinet, flattens the woofer's impedance peak and phase shift at resonance, thus making the system's bass less affected by varying amplifier and speaker cable source impedance (see Hotlines 8-11). A Zobel type network in the crossover corrects for the rising woofer voice coil impedance at its upper frequencies. Another correction network pulls down the excess output of the woofer in the lower midrange (due we think to front baffle loading, though the booklet's explanation is different). And so on.

The Perkins PR Two D is reasonably efficient, and seems to play reasonably loud. Its bass goes impressively low for a system with an 8 inch woofer that's highly damped. At its price of $1850 per pair, it's not inexpensive for a two way system. But the buyer is getting a lot of custom crafted refinements, which in turn should make a difference in sonic refinement, for those who can appreciate refinement and low coloration in their music listening.

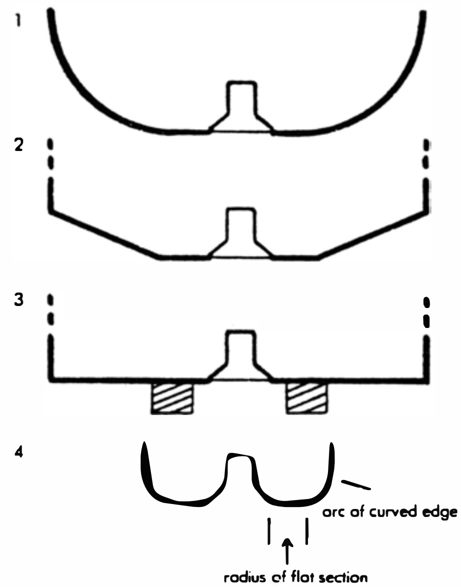


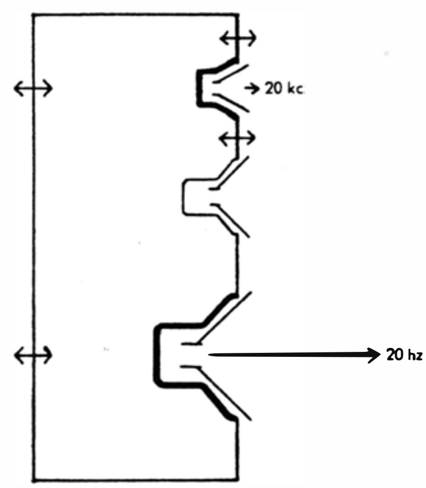
Low or High Crossover?

Setting a first order crossover at a low 700 hz or so (Perkins) eliminates the breakup raggedness of the woofer driver (region B, which starts around 2000 hz) from being heard as colorations (notice how far down from the top line this raggedness is located). But the small tweeter must handle a lot of power (region A), which limits the loudness capability of the system (unless the tweeter is ruggedized, as in the Perkins).

Diffraction can be combatted by:

1. severely rounding the baffle's own contours, so there are no sharp edges or corners as discontinuities
2. radically bevelling the baffle's contours
3. absorbing the acoustic energy travelling sideways along the baffle surface, before it gets to the edge
4. using very small baffle surfaces, so only the highest frequencies (smallest wavelengths) are even supported by the baffle in the first place. A small rounding of merely the edges then suffices to prevent these small wavelengths from seeing a discontinuity. It's sufficient if the arc of the curved edge is of the same order as the radius of the flat baffle section.

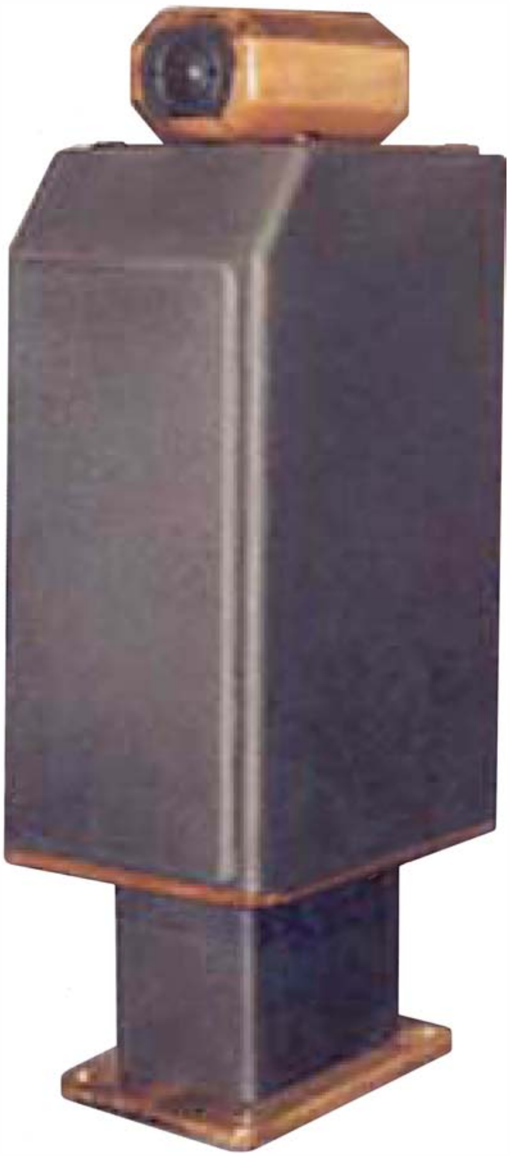


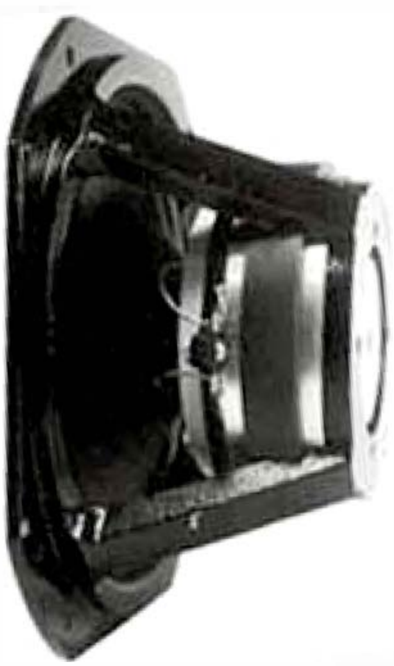


Rigid Cabinets

The woofer is being fed a strong signal at 20 hz, the tweeter an equally strong signal at 20 kc. Let's suppose the cabinet is a high quality rigid one, and vibrates only .001% (1/100,000) of what the woofer cone does. Then the tweeter frame, mounted to the same cabinet, is vibrating merely .001% of the woofer's excursions. But the tweeter cone's excursions for an equal strength signal at 20 kc are just 1/10 of this.

So the tweeter's frame, which is the reference point for the tweeter's own output, is vibrating 10 times the amplitude of the tweeter cone's own excursions. That means you're hearing 1000% modulation distortion of the tweeter's 20 kc signal by the 20 hz signal. Thus, music's treble will be severely blurred by any bass. That's why Perkins uses heavy, mechanically isolated separate cabinets for the tweeters.





PR-2 — Precision Reference Loudspeaker; E-series

The E-series embodies substantial improvements to every aspect of the design including:

— the tweeter is upgraded from the conventionally suspended soft dome to an ultra-compliantly suspended polymer-graphite hard dome of truly exceptional performance. The magnet structure is a non-resonant assembly that provides excellent rear wave damping in the large, absorbent filled cavity in the tweeter head.

Seeking reduce moving mass, we experimented with both aluminum and copper clad aluminum wire (CCAW) for the voice coil. Aluminum presented solder joint reliability issues, while CCAW is the worst sounding conductor ever experienced, unimaginably so...

— the bass unit features a post-machined, cast basket, a hard PVC Cobex cone with a welded surround and lager and much more linear magnet structure. The cast basket is far stiffer than the earlier stamped steel basket and being of much more open structure allows unimpeded rear wave egress into the DAMPS-loaded bass enclosure. The acoustic resistance unit (ARU) is improved for better damping and flatter complex impedance.

— the enclosure uses a first-in-industry constrained-layer material that essentially eliminates the decades long problem with enclosure wall resonances; allowing a 'detuning' of a band-reject mid-bass/bass step filter in the low-pass section of the crossover.

— the amplifier inputs are now bi-wired and use Cardas terminals. The internal wiring is upgraded.

— the stand is sand-filled with an attendant lowering of the speaker's center of gravity and an increase in all-up weight to 100lbs.

The solid Koa tweeter head with the tweeter I spent 10 years developing, the magnet structure for which is seen in the following pages. Although not seen in this brochure, the cover over the tweeter dome is easily removed, however mindfulness regarding replacement is paramount as the shiny, black "tweeter button" is a finger magnet with nearly "magical" properties. The need for 95% of tweeter repairs is the result of an inquisitve finger poked through the dome.

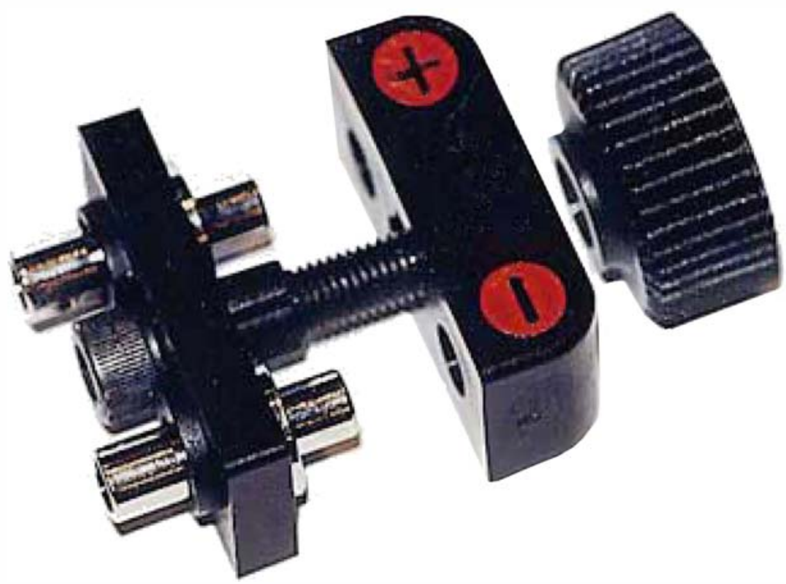
Koa is a remarkbly beautiful wood worthy of its own page, seen **here**.

The patented Cardas binding post system eliminates the terrible sounding screw threads on the current carrying components seen in other speaker connection terminals, with the result that this design sounds better. It is designed to be internally direct soldered. The connection posts accept 6.3mm spade lugs and can accommodate 9mm as well.

- Base metal: High purity copper
- Plating: Silver/rhodium
- Termination: Cardas Quad Eutectic Solder

Although not seen here, we replace the steel, center bolt with one of a non-magnetic, non-electrically conductive material. While it might seem unlikely to many, this effects a plainly audible improvement in sound quality.

A downside is that the material has a much lower tensile strength than steel so the he-men must go easy on tightening torque.



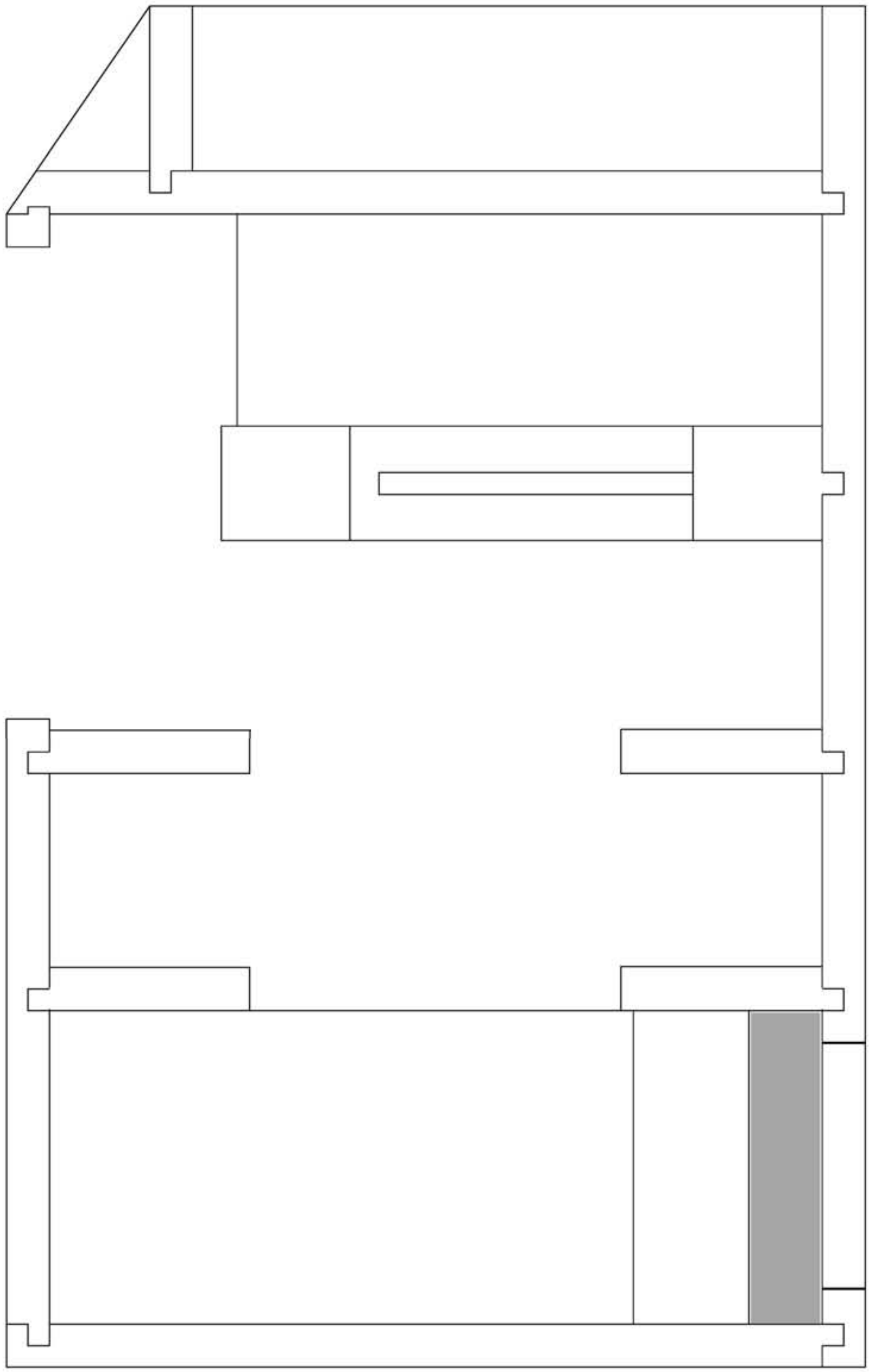
Reinforcements in all four corners behind the bass unit provide solid anchoring for the 'T' nuts into which the unit's eight, 3" long mounting bolts are threaded.

Embedded within the 2" thick enclosure wall brace seen here is a barrel nut into which is threaded a bolt that passes directly through the center pole of the bass unit's magnet structure.

By means of these fasteners the bass unit/magnet structure is made an integral part of the 65lb. enclosure, thereby ensuring a ratio of reactive bass unit/enclosure mass to driven-cone mass greater than 1,000:1 or 60dB.

Any rocking motion that might induced into the entire structure as a reaction to driven, cone mass is essentially eliminated.

A cross-section view of the contrained layer sheet material we custom fabricate for use in construction of the bass enclosure. Having high density, ρ, high Young's modulus, E, and very high internal damping, tan δ, it is the supremely effective result of a decade's developmental work and in 1986, the first such implementation of constrained-layer enclosure construction.



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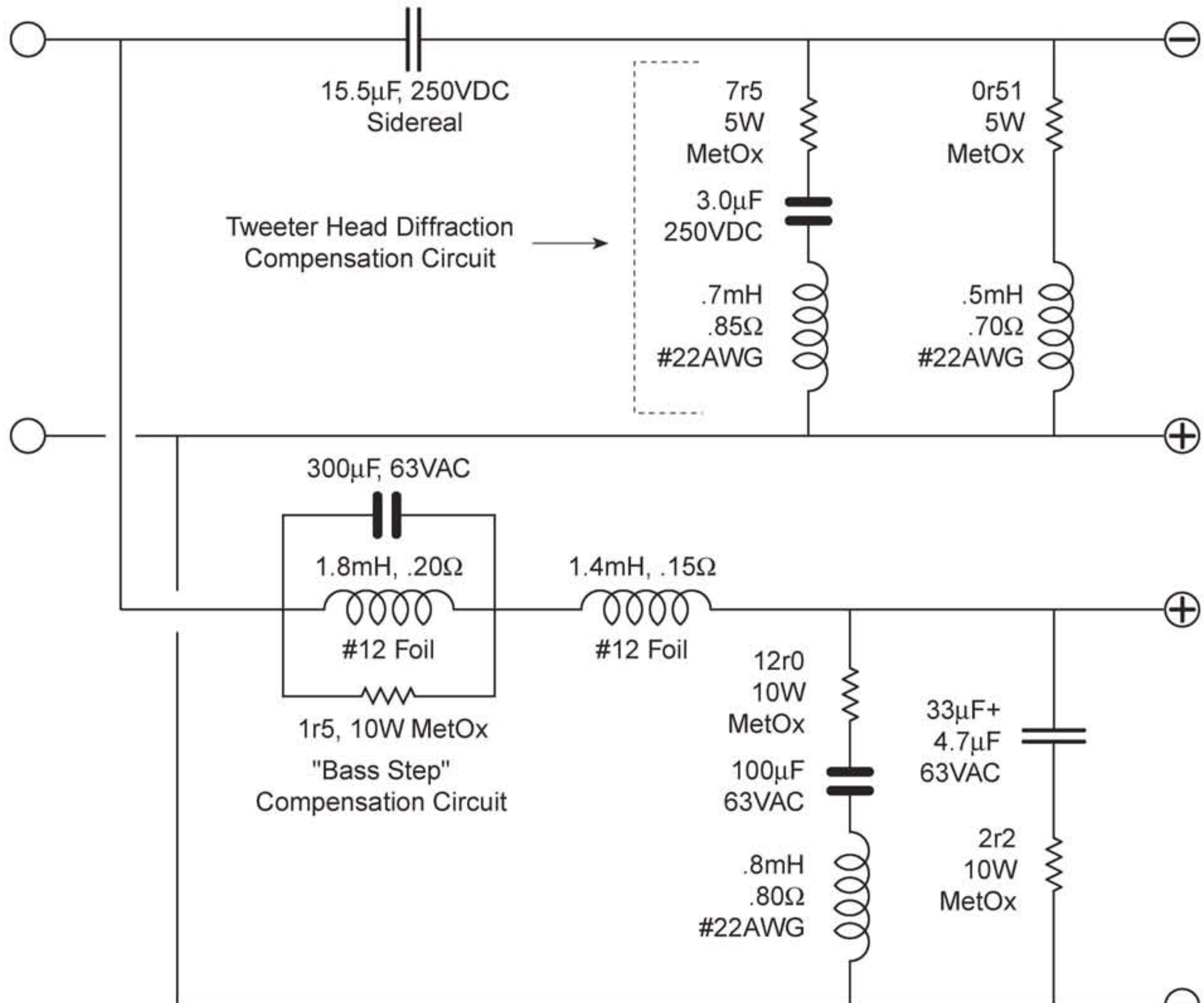
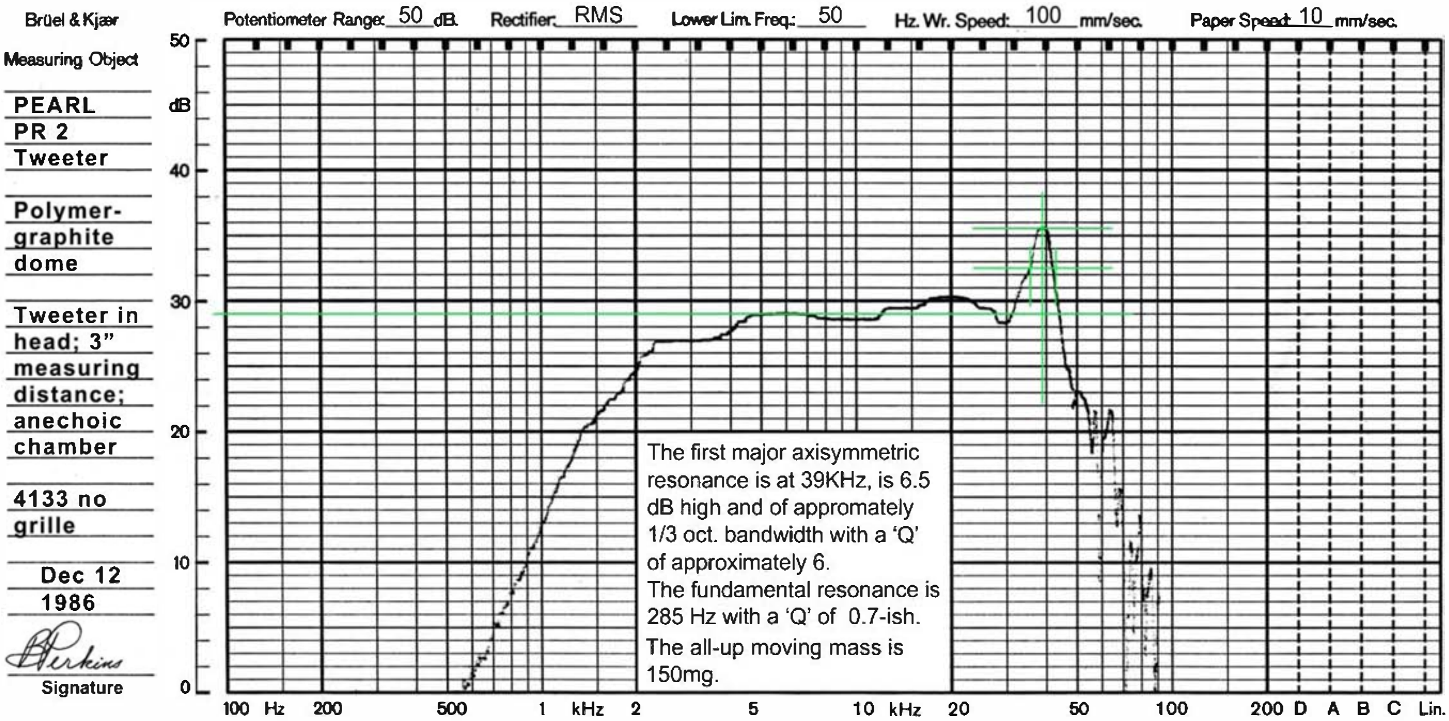
Date: 9/2/00	Ver No: 3.30	



Name: PR2 Crossover		Notes:
Date: 8/28/98	Ver No: 2.8	



This curve was run with a Brüel & Kjaer Type 2010 Hetrodyne Analyzer and plotted with a Brüel & Kjaer Type 2307 Level Recorder. The signal was old school swept sine with a slow paper speed, a fast writing speed and a low averaging time constant.

The conditions in the PEARL designed and constructed anechoic chamber seen here were anechoic above approximately 300Hz, the measuring microphone was a 12mm Brüel & Kjaer Type 4133 with its protective front grille removed.

The PEARL designed and built dipole radiating tweeter was mounted in the solid koa tweeter head seen here and was driven by the high-pass section of the passive crossover seen here.

The dome itself was formed on the PEARL designed and built hot cavity vacuum former seen here with the math to derive the range of blended radii, or torispherical, shapes investigated is here.

In the late '70s and early '80s scanning laser doppler vibrometers (SLDV) such as seen here and here were the province those with deep pockets so I did the development by the simple, empirical expedient of building tweeters with different profiles and finding the best performer.

TOL'RC	
INCH	
1	+.000 -.001
2	+.000 -.002
3	+.000 -.003
4	+.000 -.004
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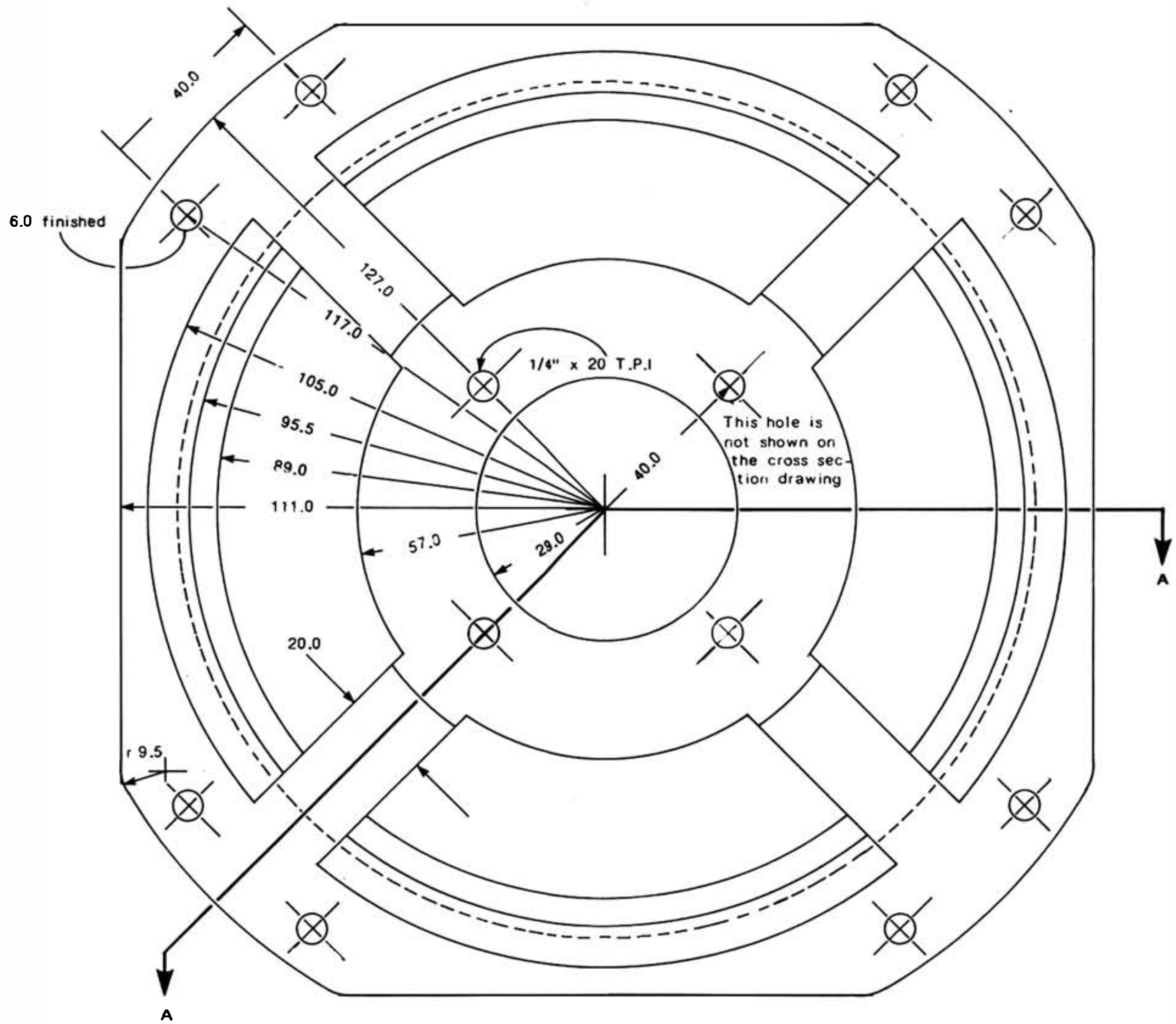
6	+.001 -.000
7	+.002 -.000
8	+.003 -.000
9	+.004 -.000
10	
11	±.0005
12	±.001
13	±.002
14	±.003
15	±.004
16	
17	
METRIC	
18	+.000 -.025
19	+.000 -.050
20	+.000 -.075
21	+.000 -.100
22	
23	+.025 -.000
24	+.050 -.000
25	+.075 -.000
26	+.100 -.000
27	
28	±.013
29	±.025
30	±.050
31	±.100
32	±.125
33	
34	
PERCENT	
35	±.5%
36	±1.0%
37	±2.0%
38	±5.0%
39	±10%
40	±20%
FRACTION	
41	±1/64
42	±1/32
43	±1/16



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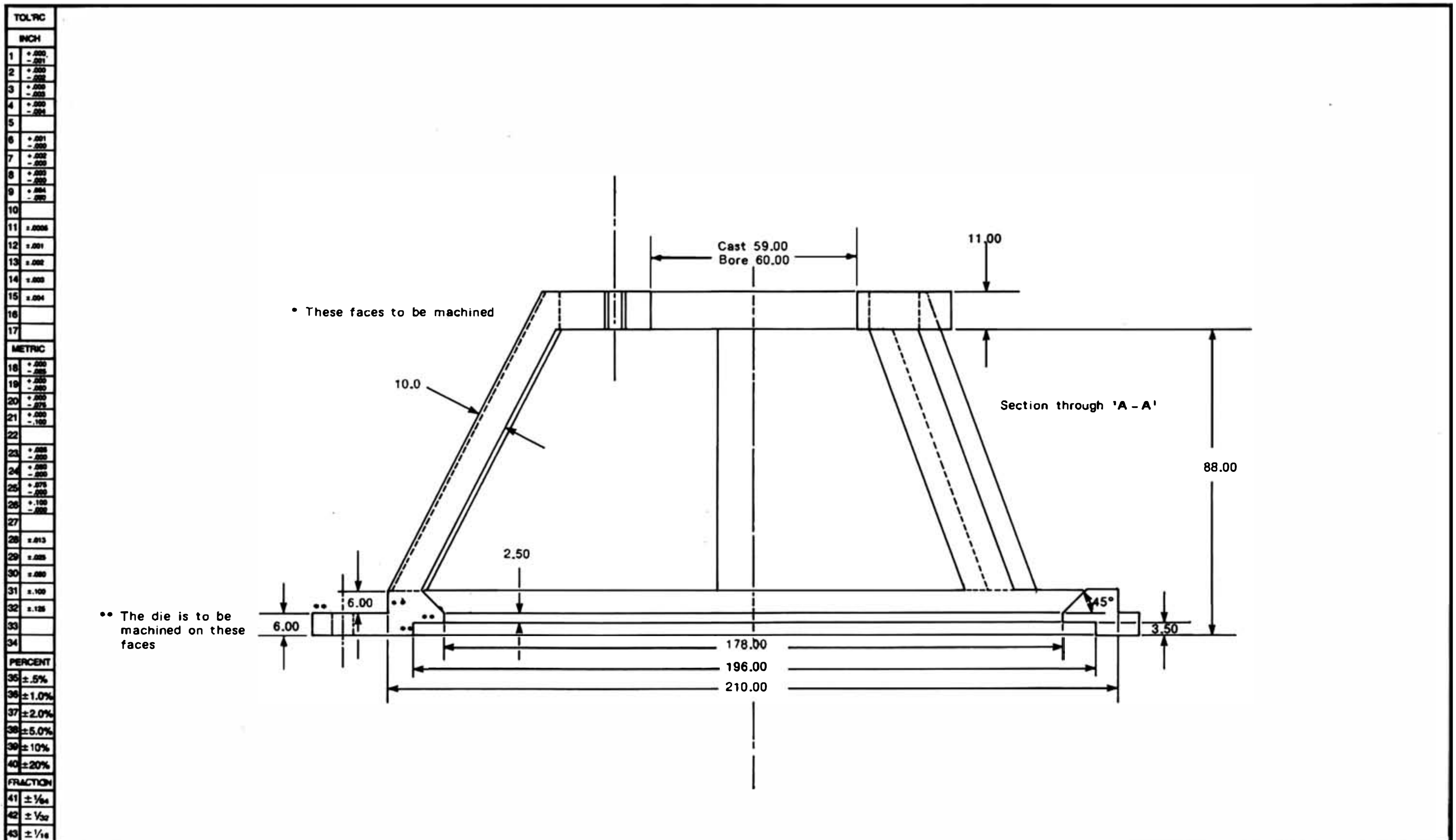
TITLE 200mm Cast Basket			DRAWING # 1078
DRAWN BY B. Perkins	CHECKED BY D. Connor	APPROVED BY B.P.	DATE: Mar 20, 1985
SEND COPY TO:			SCALE: 1:1



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TELEPHONE: PLANT - (403) 245-5882 SALES - (403) 244-HIFI

TITLE 200mm Cast Basket			DRAWING # 1077
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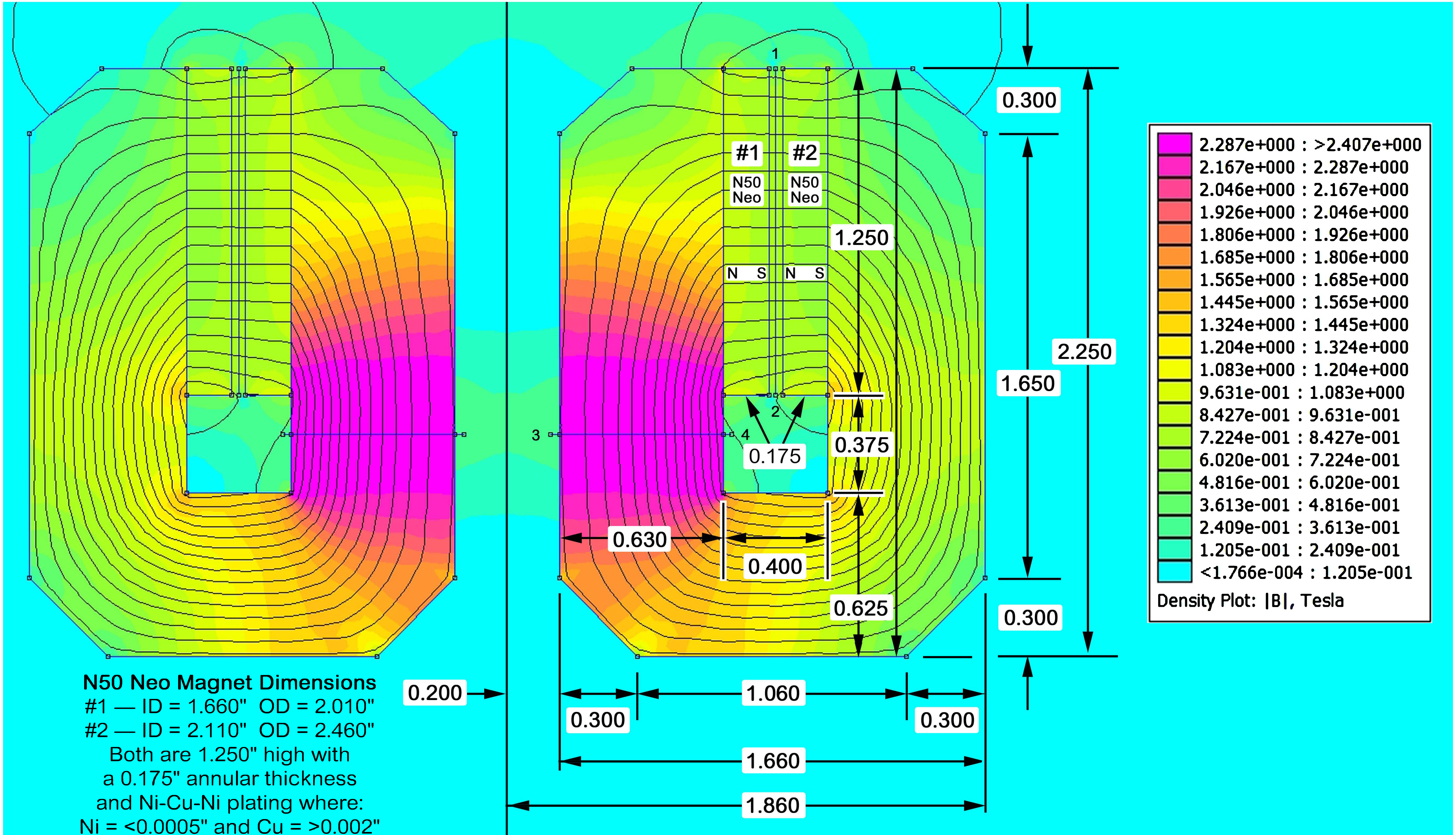

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N50 Neo
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1.650
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0.625
0.300
0.200
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1.060
0.300
1.660
1.860
2.287e+000 : >2.407e+000
2.167e+000 : 2.287e+000
2.046e+000 : 2.167e+000
1.926e+000 : 2.046e+000
1.806e+000 : 1.926e+000
1.685e+000 : 1.806e+000
1.565e+000 : 1.685e+000
1.445e+000 : 1.565e+000
1.324e+000 : 1.445e+000
1.204e+000 : 1.324e+000
1.083e+000 : 1.204e+000
9.631e-001 : 1.083e+000
8.427e-001 : 9.631e-001
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6.020e-001 : 7.224e-001
4.816e-001 : 6.020e-001
3.613e-001 : 4.816e-001
2.409e-001 : 3.613e-001
1.205e-001 : 2.409e-001
<1.766e-004 : 1.205e-001
Density Plot: |B|, Tesla
N50 Neo Magnet Dimensions
#1 — ID = 1.660" OD = 2.010"
#2 — ID = 2.110" OD = 2.460"
Both are 1.250" high with
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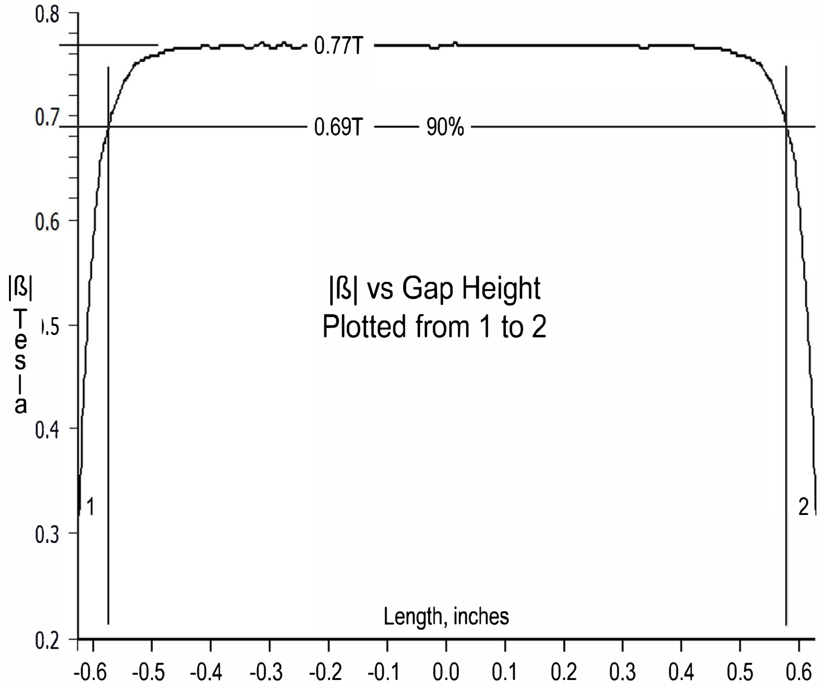
and Ni-Cu-Ni plating where:
Ni = <0.0005" and Cu = >0.002"


|ß| vs Gap Height
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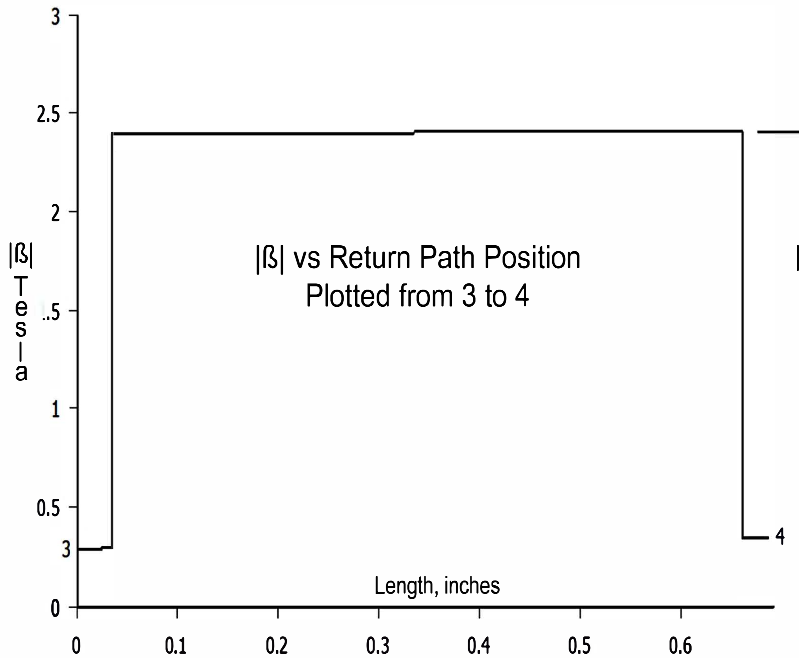
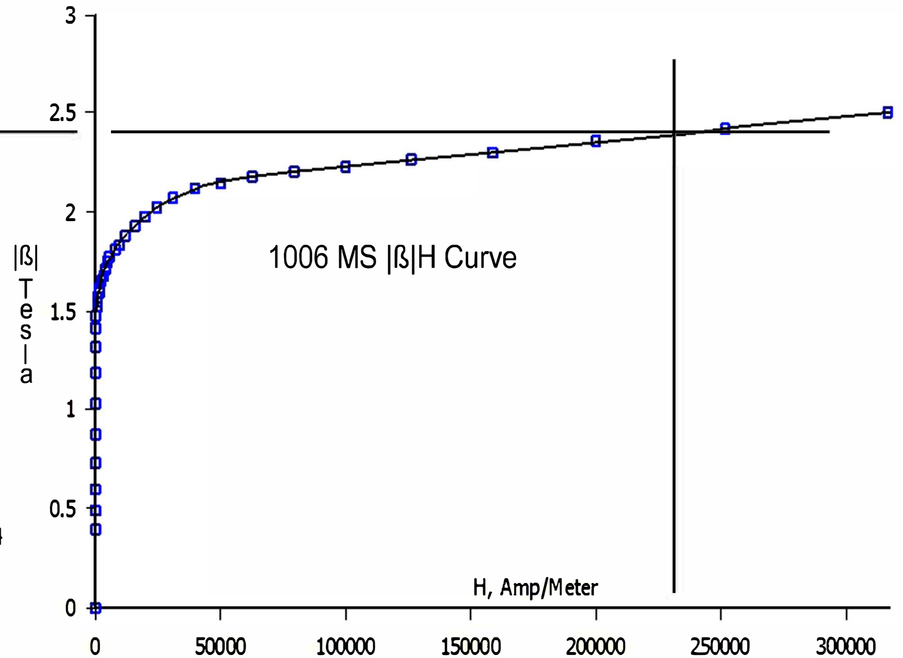
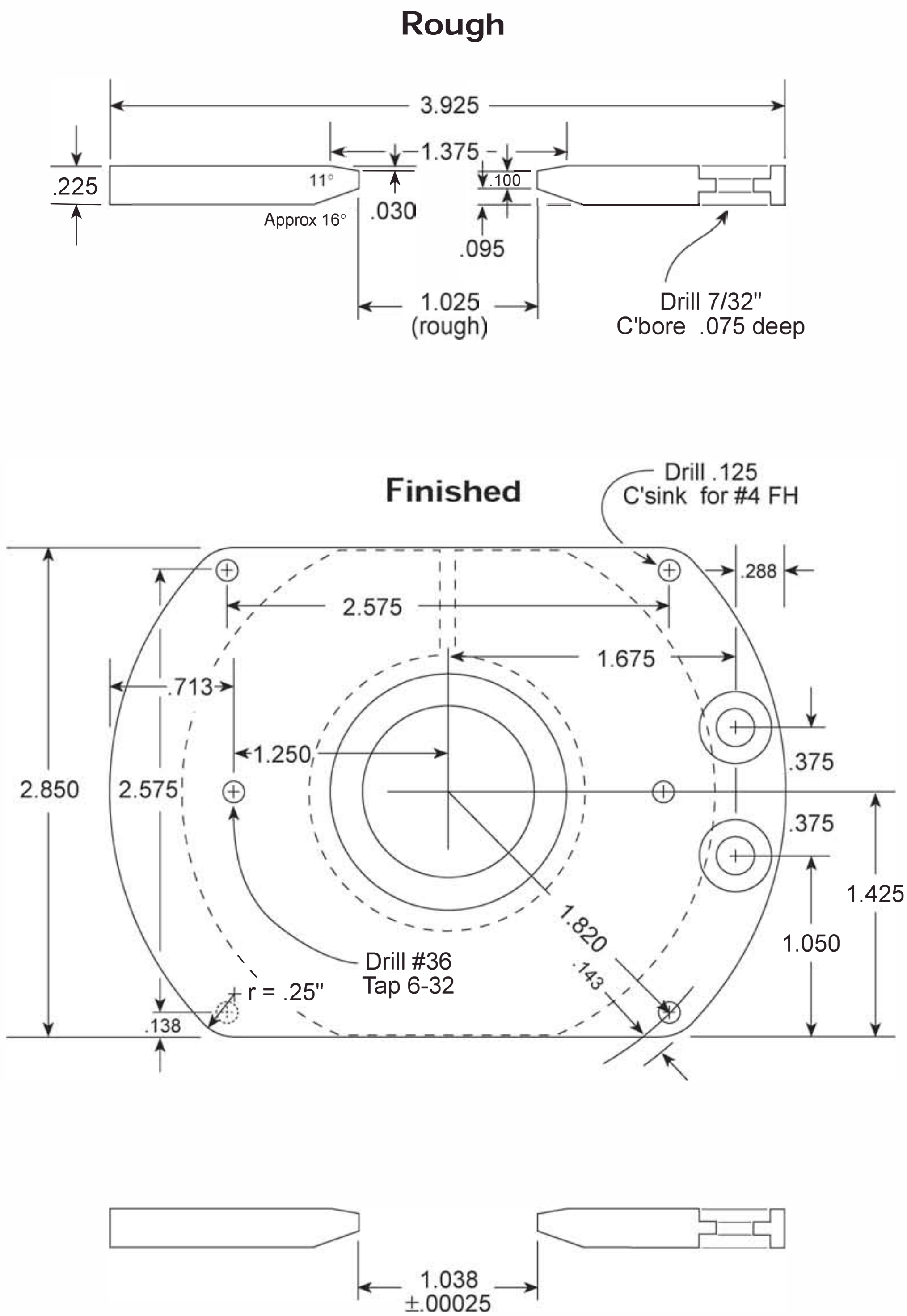
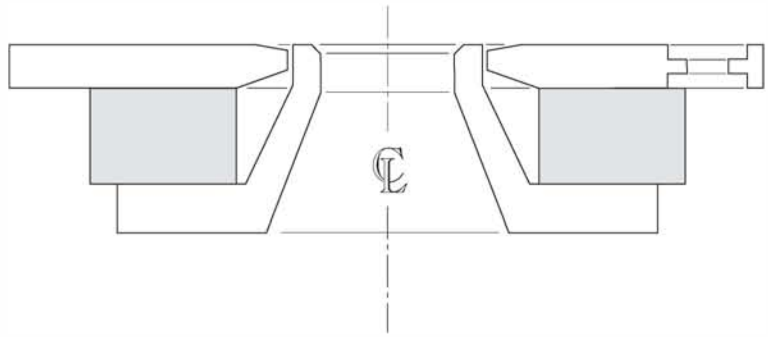

|ß| vs Return Path Position
Plotted from 3 to 4
|ß| Tesla
Length, inches
3
4
3
2.5
2
1
0.5
0
0 0.1 0.2 0.3 0.4 0.5 0.6

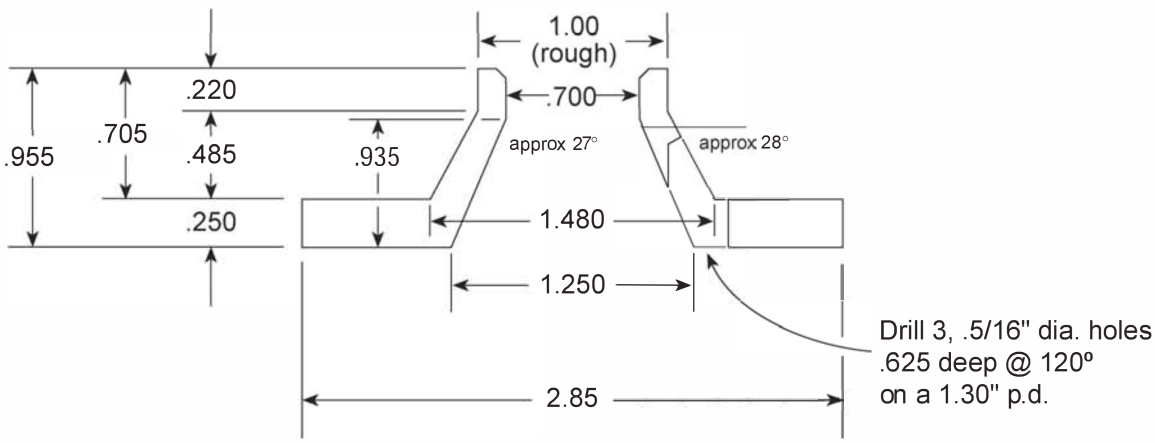

1006 MS |ß|H Curve
|ß| Tesla
H, Amp/Meter
3
2.5
2
1.5
1
0.5
0
0 50000 100000 150000 200000 250000 300000



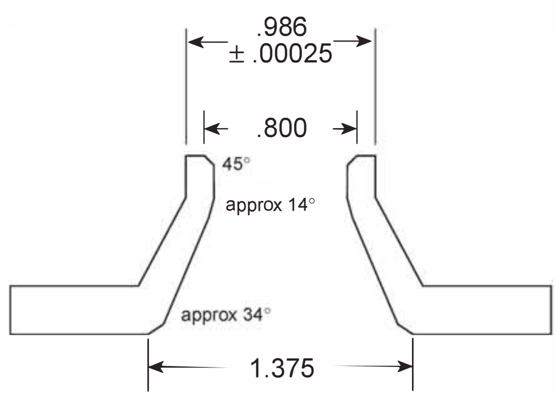
Drawing Name: PR 2.6.0 Tweeter Magnet Assy: Top Plate		Notes: The top plate is plated .0009" acid copper and .0001" nickle.
Drawing Date: Nov 12, 97	Version Number: 1.3	



Rough

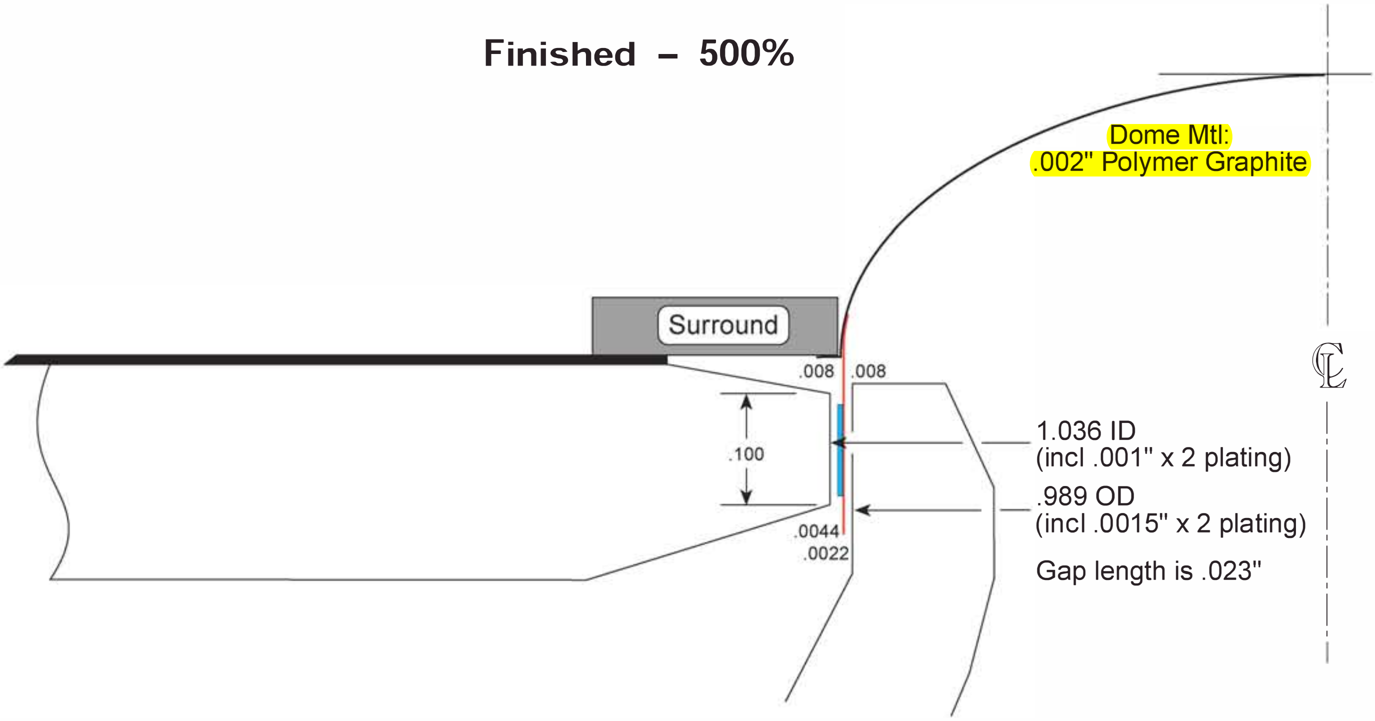


Finished



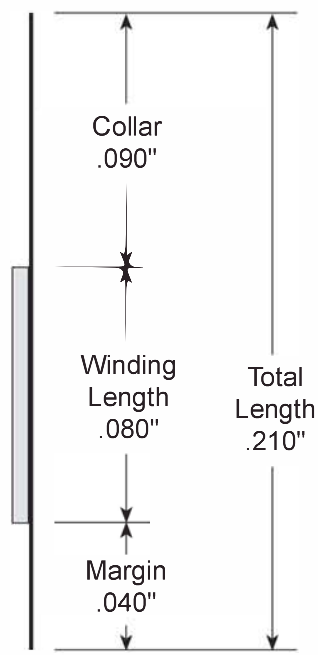
Drawing Name: PR 2.6.0 Tweeter Magnet Assy: section		Notes: The center pole is plated .0014" acid copper and .0001" nickle
Drawing Date: Oct 8, 97	Version Number: 1.2.0	

Finished – 500%



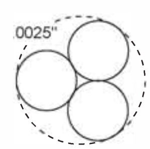
Additional Data

Hisco C200 former is 95gms/ sq. meter. Former is .645 sq"; 39.5 mG. Hisco .003" MTB is 81gms/sq. meter, former is 33.7mG. Hisco .0022 Ultratuf is 48gms/sq. meter, former is 20.0mG. Using C200, coil mass in 100mG



New Voice Coil Specs:
Rdc = 3.9Ω.
# of turns on all coils must be ***identical,*** ie. ±0.0 turns.
Former is gapped so that the bottom lead just fills the gap.
Leads break out @ 180°
Lead length is 5", untinned.

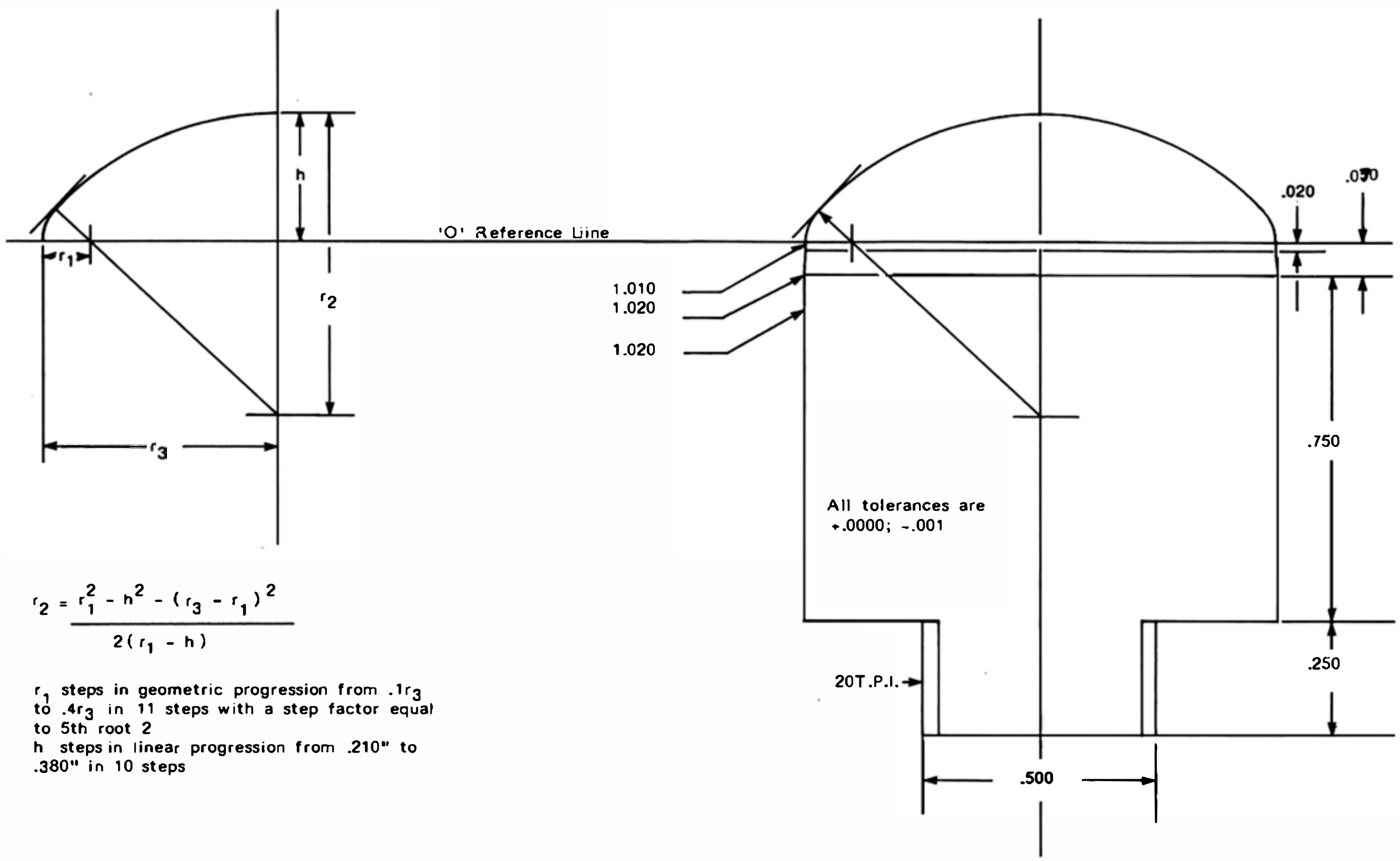
.0053" OD Nom
.0044" Actual



3 - # 44
.86Ω/ft.
17.8 mG/ft

Drawing Name: PR 2.6.0 Tweeter Voice Coil Assy		Notes: The top plate is plated .0009" acid copper and .0001" nickle. The center pole is plated .0014" acid copper and .0001" nickle
Drawing Date: Nov 24, 98	Version Number: 1.6	

TOL'RC	
INCH	
1	+.000 -.001
2	+.000 -.002
3	+.000 -.003
4	+.000 -.004
5	
6	+.001 -.000
7	+.002 -.000
8	+.003 -.000
9	+.004 -.000
10	
11	±.0005
12	±.001
13	±.002
14	±.003
15	±.004
16	
17	
METRIC	
18	+.000 -.025
19	+.000 -.050
20	+.000 -.075
21	+.000 -.100
22	
23	+.025 -.000
24	+.050 -.000
25	+.075 -.000
26	+.100 -.000
27	
28	±.013
29	±.025
30	±.050
31	±.100
32	±.125
33	
34	
PERCENT	
35	±.5%
36	±1.0%
37	±2.0%
38	±5.0%
39	±10%
40	±20%
FRACTION	
41	±1/64
42	±1/32
43	±1/16



$$r_2 = \frac{r_1^2 - h^2 - (r_3 - r_1)^2}{2(r_1 - h)}$$

r_1 steps in geometric progression from $.1r_3$ to $.4r_3$ in 11 steps with a step factor equal to 5th root 2
h steps in linear progression from .210" to .380" in 10 steps

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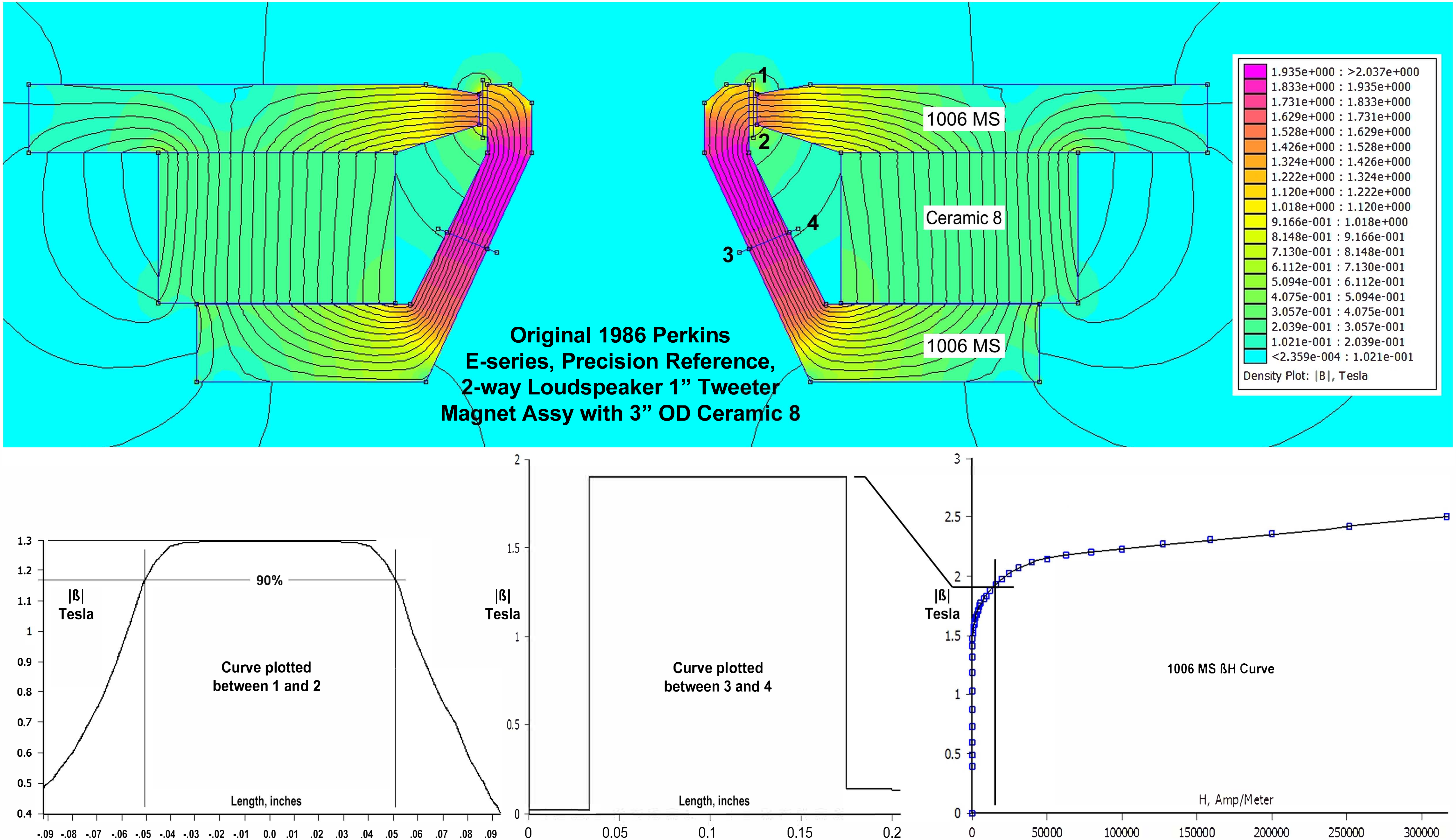
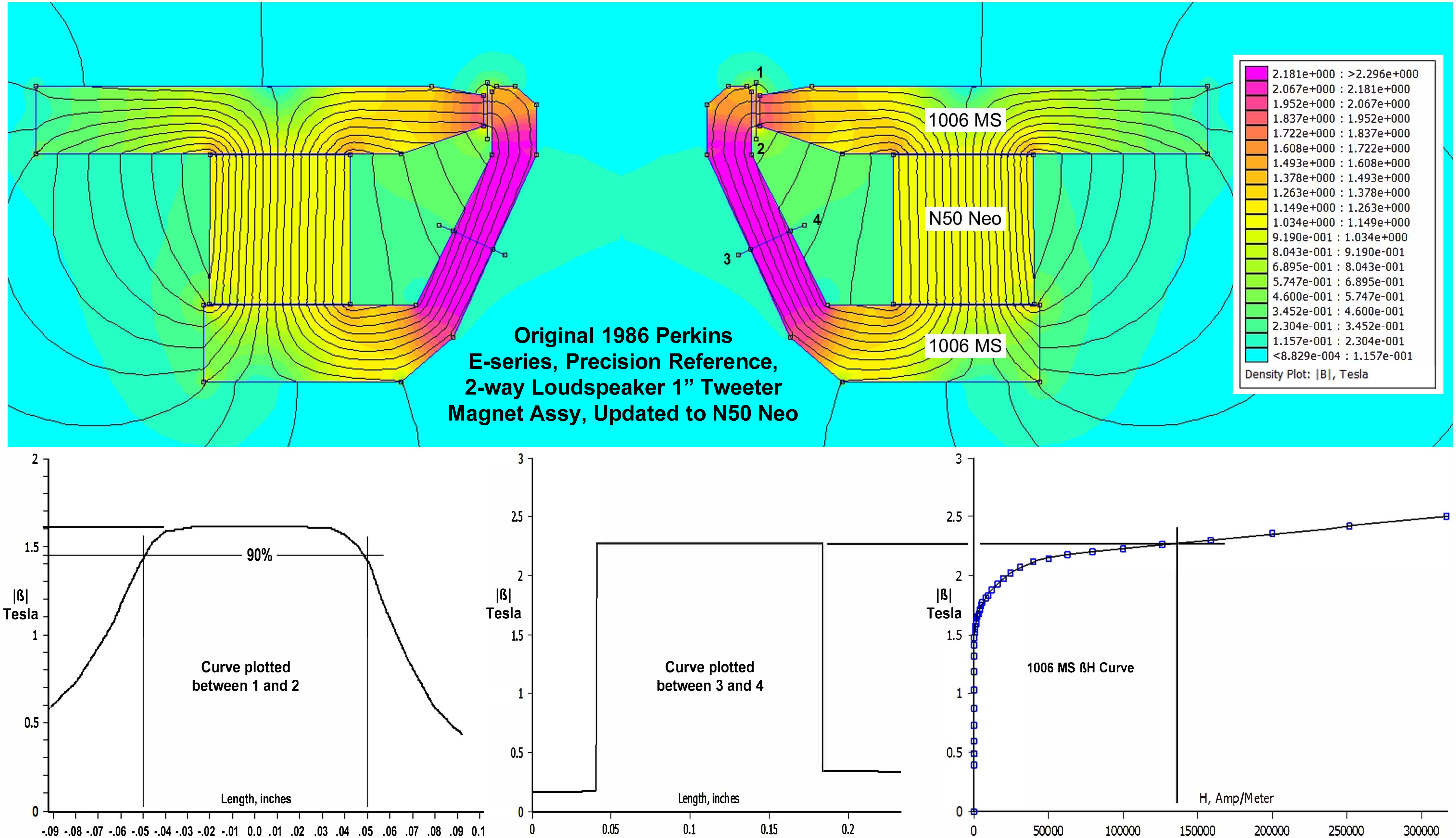
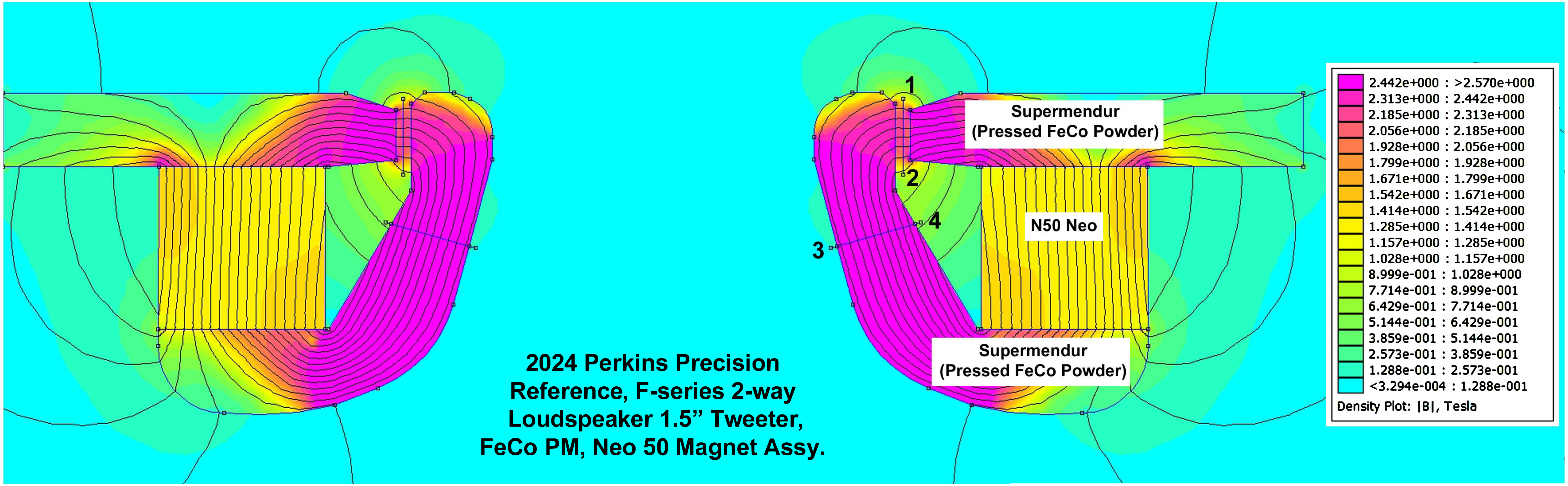
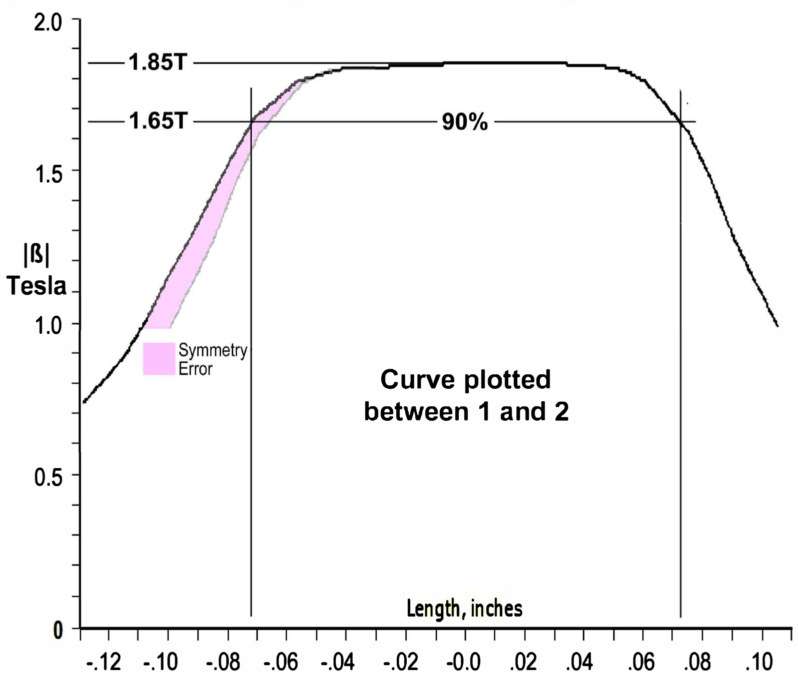
TITLE Blended Radii Tweeter Dome Forming Die			DRAWING # 1068
DRAWN BY Bill Perkins	CHECKED BY D. Connor	APPROVED BY	DATE Jan 15, 1985
SEND COPY TO:			

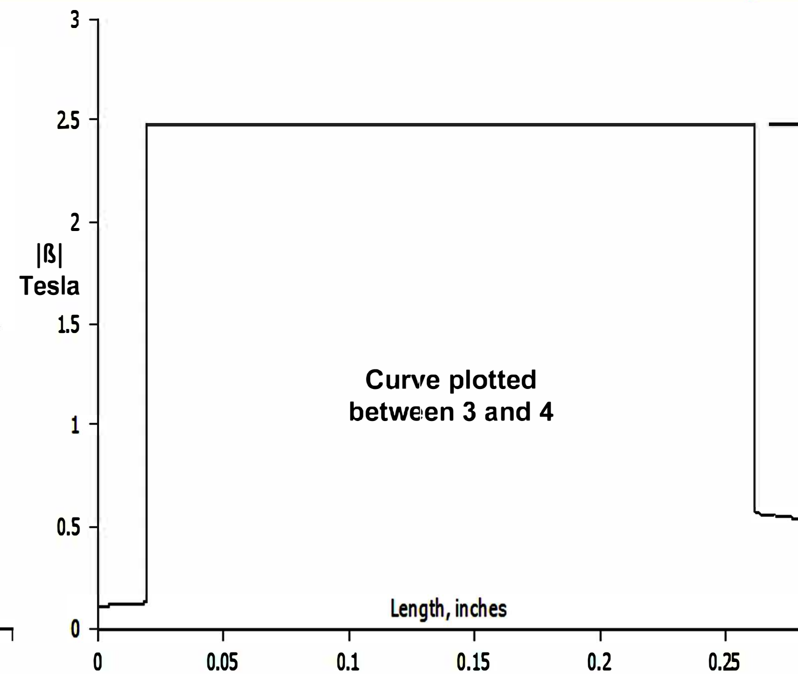

1
2
3
4
1006 MS
Ceramic 8
1006 MS
Original 1986 Perkins
E-series, Precision Reference,
2-way Loudspeaker 1” Tweeter
Magnet Assy with 3” OD Ceramic 8
1.935e+000 : >2.037e+000
1.833e+000 : 1.935e+000
1.731e+000 : 1.833e+000
1.629e+000 : 1.731e+000
1.528e+000 : 1.629e+000
1.426e+000 : 1.528e+000
1.324e+000 : 1.426e+000
1.222e+000 : 1.324e+000
1.120e+000 : 1.222e+000
1.018e+000 : 1.120e+000
9.166e-001 : 1.018e+000
8.148e-001 : 9.166e-001
7.130e-001 : 8.148e-001
6.112e-001 : 7.130e-001
5.094e-001 : 6.112e-001
4.075e-001 : 5.094e-001
3.057e-001 : 4.075e-001
2.039e-001 : 3.057e-001
1.021e-001 : 2.039e-001
<2.359e-004 : 1.021e-001
Density Plot: |B|, Tesla
|ß|
Tesla
90%
Curve plotted
between 1 and 2
Length, inches
|ß|
Tesla
Curve plotted
between 3 and 4
Length, inches
|ß|
Tesla
1006 MS ßH Curve
H, Amp/Meter

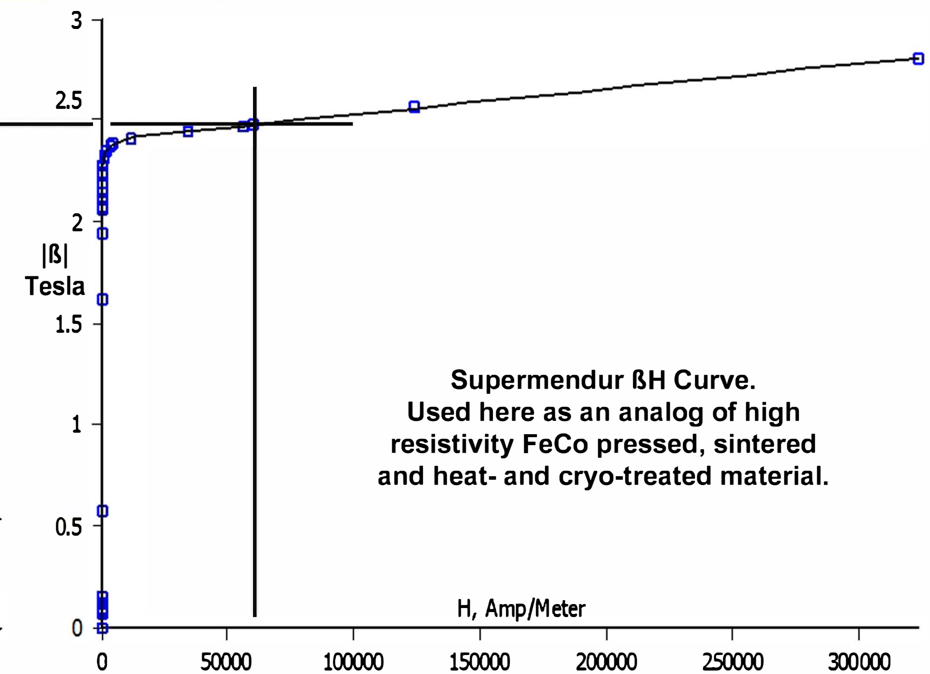


Original 1986 Perkins
E-series, Precision Reference,
2-way Loudspeaker 1" Tweeter
Magnet Assy, Updated to N50 Neo
1006 MS
N50 Neo
1006 MS
1
2
3
4
2.181e+000 : >2.296e+000
2.067e+000 : 2.181e+000
1.952e+000 : 2.067e+000
1.837e+000 : 1.952e+000
1.722e+000 : 1.837e+000
1.608e+000 : 1.722e+000
1.493e+000 : 1.608e+000
1.378e+000 : 1.493e+000
1.263e+000 : 1.378e+000
1.149e+000 : 1.263e+000
1.034e+000 : 1.149e+000
9.190e-001 : 1.034e+000
8.043e-001 : 9.190e-001
6.895e-001 : 8.043e-001
5.747e-001 : 6.895e-001
4.600e-001 : 5.747e-001
3.452e-001 : 4.600e-001
2.304e-001 : 3.452e-001
1.157e-001 : 2.304e-001
<8.829e-004 : 1.157e-001
Density Plot: |B|, Tesla
|ß| Tesla
90%
Curve plotted between 1 and 2
Length, inches
-.09 -.08 -.07 -.06 -.05 -.04 -.03 -.02 -.01 0.0 .01 .02 .03 .04 .05 .06 .07 .08 .09 0.1
Curve plotted between 3 and 4
Length, inches
0 0.05 0.1 0.15 0.2
1006 MS ßH Curve
H, Amp/Meter
0 50000 100000 150000 200000 250000 300000

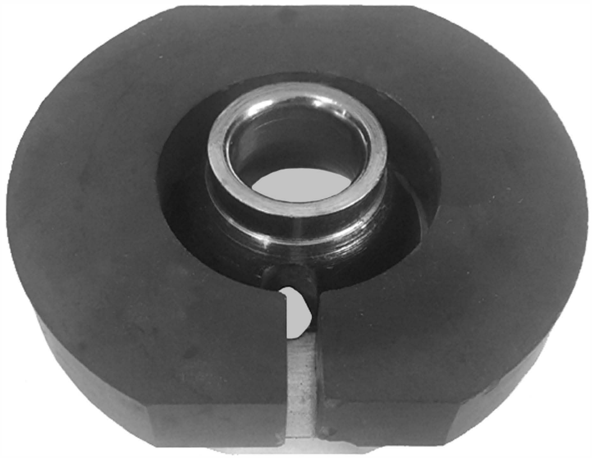

1
2
3
4
Supermendur
(Pressed FeCo Powder)
N50 Neo
Supermendur
(Pressed FeCo Powder)
2024 Perkins Precision Reference, F-series 2-way Loudspeaker 1.5” Tweeter, FeCo PM, Neo 50 Magnet Assy.
2.442e+000 : >2.570e+000
2.313e+000 : 2.442e+000
2.185e+000 : 2.313e+000
2.056e+000 : 2.185e+000
1.928e+000 : 2.056e+000
1.799e+000 : 1.928e+000
1.671e+000 : 1.799e+000
1.542e+000 : 1.671e+000
1.414e+000 : 1.542e+000
1.285e+000 : 1.414e+000
1.157e+000 : 1.285e+000
1.028e+000 : 1.157e+000
8.999e-001 : 1.028e+000
7.714e-001 : 8.999e-001
6.429e-001 : 7.714e-001
5.144e-001 : 6.429e-001
3.859e-001 : 5.144e-001
2.573e-001 : 3.859e-001
1.288e-001 : 2.573e-001
<3.294e-004 : 1.288e-001
Density Plot: |B|, Tesla

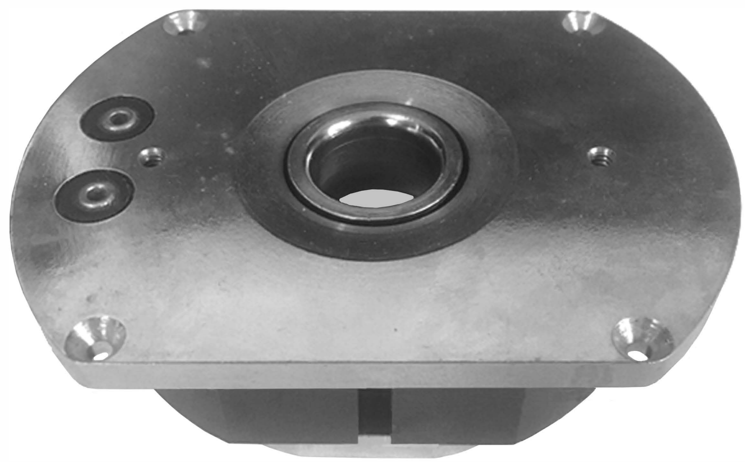

1.85T
1.65T
90%
|ß|
Tesla
Symmetry
Error
Curve plotted
between 1 and 2
Length, inches
2.0
1.5
1.0
0.5
0
-.12 -.10 -.08 -.06 -.04 -.02 -0.0 .02 .04 .06 .08 .10


|ß|
Tesla
Curve plotted
between 3 and 4
Length, inches
3
2.5
2
1.5
1
0.5
0
0 0.05 0.1 0.15 0.2 0.25


|ß|
Tesla
Supermendur ßH Curve.
Used here as an analog of high
resistivity FeCo pressed, sintered
and heat- and cryo-treated material.
H, Amp/Meter
3
2.5
2
1.5
1
0.5
0
0 50000 100000 150000 200000 250000 300000

Seen immediately below is the ceramic magnet assembly for the PR-2 E-series,
Ver. 1 loudspeakers. DC FEA data on this structure's magnetic performance is seen here.





Seen below is the updated NeFeB magnet assembly for the PR-2 E-series,
Ver. 2 loudspeakers. DC FEA data on this structure's magnetic performance is seen here.

Note in both structures the absence of sealed volumes of air.
As seen in almost all other similar structures, sealed
air volumes can and do resonate, with
deleterious effects on sound quality.

Drawing Name: PR-2 E-series Tweeter Magnet Assys		Notes: Based on the voice coil gap's magnetic energy, the output of the Ver. 2 device is expected to be 2-3dB higher than that of the Ver. 1 device.
Drawing Date: Dec. 7, 2023	Version Number: N/A	

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Loudspeaker Damping

ALBERT PREISMAN*

Part 1. A discussion of theoretical considerations of loudspeaker characteristics, together with a practical method of determining the constants of the unit as a preliminary step in obtaining satisfactory performance.

ONE OF THE CONSIDERATIONS in the design and application of loudspeakers is the adequate damping of their motion. Thus, owing to the masses and compliances involved, the sudden application or removal of current in the voice coil tends to produce a transient oscillation of a damped sinusoidal nature.

In particular, the sudden cessation of current in the voice coil may find the loudspeaker continuing to vibrate in the manner described, so that the sound "hangs over". Any one who has experienced this unpleasant effect will seek ways and means to eliminate it.

In the case of a horn type loudspeaker, the horn imposes in general sufficient mechanical loading to damp out such transient response of "hangover", and also serves to limit the excursions of the voice coil so that it does not operate into the nonlinear portion of the air-gap magnetic field. The damping also serves to minimize nonlinear compliance of the suspension system by limiting the amplitude of oscillation.

However, if the horn design is limited by such considerations as maximum permissible mouth area and is operated at a frequency not too low to be transmitted by the horn taper yet low enough so that appreciable reflections occur at the mouth, then the horn may cease to act as a mechanical resistance, but instead become predominantly reactive, and thereupon cease to damp a resonance in the speaker unit occurring in this frequency range. In such an event other means of damping will be of value to the designer or applications engineer.

In the case of the direct-radiator loudspeaker unit, the air load is small, and is mainly reactive at the lower frequencies. Hence mechanical damping of the unit is small in magnitude, and "hangover" effects may be particularly noticeable.

A reflexed cabinet may help to load the loudspeaker, or at any rate to produce a two-mesh mechanical network exhibiting two resonance peaks, neither of which is as high as that of the unit by itself or in a flat baffle. Nevertheless, the damping may still not be sufficient to produce "clean" low-frequency tones.

Hence, in general, it is advisable or at least desirable to provide sufficient damping of the direct-radiator type of unit by means of its electrical characteristics, so that whether it is operated into a horn, reflexed cabinet, or simply a flat baffle, it will be adequately damped.

An important point about electrical damping is that it represents high rather than low efficiency of operation, just as a horn does. On the other hand, were some material such as viscaloid employed to provide the required damping, the electrical input power would in part at least be converted into heat energy in the material instead of into acoustic energy, and thus represent a decrease in efficiency. It will therefore be of interest to examine damping produced by the electrical characteristics of the system.

Motional Impedance

When an alternating current flows in a voice coil, it reacts with the constant magnetic field to produce an alternating force which causes the voice coil to vibrate at the frequency of the current.

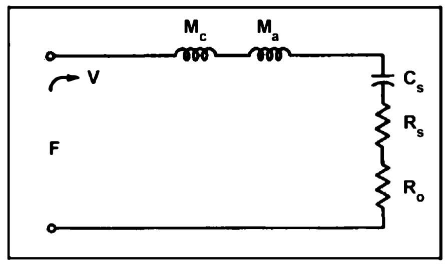

Fig. 1. Equivalent circuit of loudspeaker unit at low frequencies.

In so doing, the voice coil cuts through the magnetic lines, and generates a counter electromotive force, c.e.m.f.

The action is exactly similar to that of the rotating armature of a d.c. motor – the armature generates a c.e.m.f. by its rotation in the magnetic field. Consider the case of the loudspeaker voice coil. The electrical c.e.m.f. which is generated, tends to oppose the flow of current in the coil, just as if its impedance had gone up. After all, one ohm of impedance simply means a one volt drop in the unit for a one-ampere current flowing through it; i.e., volts per ampere. In the case of the loudspeaker the force,

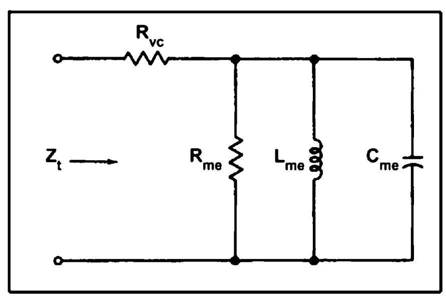

Fig. 2. Mechanical characteristics of speaker seen from voice-coil terminals.

* *Capitol Radio Engineering Institute, Washington, D.C.*

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and hence motion and c.e.m.f., are proportional to the voice coil current, so that a ratio is involved which is an apparent impedance.

Hence, when a loudspeaker voice coil is permitted to vibrate, its impedance apparently goes up. The increase in the impedance owing to its motion is known as the MOTIONAL IMPEDANCE, and it is measured in ohms just as the electrical impedance of the voice coil is measured in ohms.

Several characteristics of the motional impedance can be readily analyzed qualitatively. In the first place, the lower the mechanical impedance, the more readily does the voice coil vibrate, and the higher is the induced c.e.m.f. for a given current flowing through it; i.e., the higher is its motional impedance.

A second point to note is that the greater the magnetic flux density, the greater is the induced c.e.m.f., and the higher is the motional impedance of the voice coil. Finally, we note that if the total length of voice-coil wire is increased, there is more conductor cutting the magnetic field, and hence more c.e.m.f. induced. Therefore the motional impedance increases if the length of voice coil wire is increased.

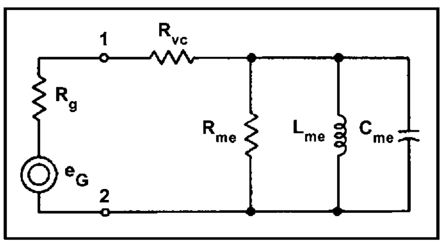

Fig. 3. Circuit of Fig. 2 with addition of generator.

The actual quantitative relations are as follows:

$$Z_{me} = \frac{(Bl)^2 \times 10^{-9}}{Z_m} \qquad (1)$$

where Z_{me} is the motional impedance in electrical ohms; B is the magnetic flux density in gauss; l = length of voice coil conductor in cm., and Z_m is the mechanical impedance in mechanical ohms, i.e. (dynes/cm/sec.).

Loudspeaker Low-Frequency Resonance

The mechanical impedance Z_m of the loud speaker unit varies considerably over the frequency range. However, as a direct radiator its value and effect on the lowest audio frequencies is of greatest importance, particularly with regard to "hangover' effects, and hence will be

At the lowest audio frequencies, the loudspeaker unit acts mechanically as a simple series resonant circuit. This is illustrated in *Fig.* 1. The masses involved are those of the cone, M_c, and the air set in motion by the cone M_a The latter is a function of frequency but can be assumed fairly constant over a narrow frequency range involving the resonant frequency of the unit.

The compliance C_s represents that of the suspension, both of the rim of the cone and of the center spider. It is apt to be nonlinear, particularly for large excursions, but is reasonably constant for moderate and small amplitudes of vibration.

The resistive factors are that of the suspension R_s, and that of the air set in motion by the cone, R_a. The latter is particularly variable with frequency, but is usually very small at the low frequency at which resonance occurs, particularly if the speaker unit is tested by itself, or at most in a flat baffle. Values for several sizes of cones are given by Olson.[1]

From *Fig.* 1, it is apparent that

$$Z_m = (R_s + R_a) + j\omega(M_c + M_a) + 1/j\omega C_s \qquad (2)$$

Substituting this in Eq. (*1*), we obtain

$$Z_{me} = \frac{(Bl)^2 \times 10^{-9}}{(R_s + R_a) + j\omega(M_c + M_a) + 1/j\omega C_s} \qquad (3)$$

If we divide the numerator and denominator of the right side of Eq. (*3*) by $(Bl)^2 \times 10^{-9}$ we obtain

$$Z_{me} = \cfrac{1}{\cfrac{(R_s + R_a)}{(Bl)^2 \times 10^{-9}} + j\omega \cfrac{(M_c + M_a)}{(Bl)^2 \times 10^{-9}} + \cfrac{1}{j\omega C_s (Bl)^2 \times 10^{-9}}} \qquad (4)$$

Let

$$(R_s + R_a)/(Bl)^2 \times 10^{-9} = G_{me} = 1/R_{me}$$

$$(M_c + M_a)/(Bl)^2 \times 10^{-9} = C_{me}$$

$$\text{and} \quad C_s(Bl)^2 \times 10^{-9} = L_{me} \qquad (5)$$

where

R_{me} is the motional resistance corresponding to the mechanical damping R_s and R_a,

C_{me} is the motional capacitance corresponding to M_c and M_a,

and

L_{me} is the motional inductance corresponding to C_s.

In short, we shall assume that the mechanical resistance appears as an electrical conductance $G_{me} = 1/R_{me}$; the mechanical compliance appears as an electrical inductance; and the mechanical mass appears as an electrical capacitance. The latter transformation has been known for a long time in the power field; years ago oscillating synchronous motors were used in Europe as electrical capacitors, since a relatively small armature mass appeared as a surprisingly large electrical capacitance.

If we substitute Eq. (*5*) in Eq. (*4*), we obtain:

$$Z_{me} = \frac{1}{(1/R_{me}) + j\omega C_{me} + (1/j\omega L_{me})} \qquad (6)$$

The quantities on the right side represent a resistance, capacitance, and inductance ***in parallel*** since the parallel impedance is equal to the reciprocal of the sum of the reciprocals of the individual impedances.

Hence we finally arrive at the conclusion that the mechanical characteristics of the loudspeaker at the lower frequencies appear at the electrical terminals of the voice coil as shown in *Fig.* 2. Here R_{vc} represents the electrical resistance of the voice coil; the electrical (clamped) inductance of the voice coil can be disregarded at the lower audio frequencies.

The mechanical characteristics of the speaker appear as a parallel resonant circuit shunted by a certain amount of resistance; these constitute the motional impedance Z_{me} of the speaker, and the total electrical impedance Z_t is Z_{me} plus R_{vc}.

We can now analyze the behavior of the speaker from its electrical motional impedance characteristics. Thus, just as *Fig.* 1 indicated a certain frequency of resonance, so does *Fig.* 2 indicate this fact. Since the two circuits are equivalent, they must have the same resonant frequency. This can be readily shown. Thus, from Eq. (*5*)

$$L_{me} C_{me} = C_s (Bl)^2 \times 10^{-9} \frac{(M_c + M_a)}{(Bl)^2 \times 10^{-9}} = (M_c + M_a) C_s \qquad (7)$$

that is, the electrical LC product equals the mechanical MC product; either

[1] H. F. Olson, "Elements of Acoustical Engineering," p. 126. D. Van Nostrand Co. New York.

therefore represents the same resonant frequency.

It will be of interest to compare the behavior of the electrical circuit of *Fig.* 2. For example, at the resonant frequency of the loudspeaker, namely

$$f_r = \frac{1}{2\pi(M_c + M_a)C_s} = \frac{1}{2\pi L_{me} C_{me}} \quad (8)$$

the mechanical current or velocity v is a maximum, and is in phase with the force F, *Fig.* 1.

This in turn means that the electrical c.e.m.f. will be a maximum and in phase opposition with the force F, which in turn is in phase with the current in the voice coil. Hence this c.e.m.f. will produce an in-phase or *resistive* reaction: the generator will view the voice coil as having increased in impedance, and that this increased impedance is resistive in nature.

Now refer to *Fig.* 2. At the frequency of resonance, L_{me} and C_{me} act as an open circuit shunting R_{me}, so that the electrical impedance is

$$Z_t = R_{vc} + R_{me} \quad (9)$$

and is a maximum. Furthermore, if the mechanical resistance $(R_s + R_a)$ is small, v will be a maximum, as will also be the c.e.m.f., whereupon the electrical source will see a *high* resistive impedance R_{me}. This checks the inverse relation between R_{me} and $(R_s + R_a)$ given in Eq. (*5*); when $(R_s + R_a)$ is small, R_{me} appears large since $(R_s + R_a)$ appears in the denominator of the expression for R_{me} in Eq. (*5*).

Loudspeaker Damping

ALBERT PREISMAN*

Part 2. A discussion of theoretical considerations of loudspeaker characteristics, together with a practical method of determining the constants of the unit as a preliminary step in obtaining satisfactory performance.

We now come to the question of damping of the loudspeaker mechanism by the electrical circuit. In *Fig* 3 is shown the electrical equivalent of a loudspeaker illustrated in *Fig.* 2, with the addition of an electrical source of internal resistance R_G feeding it. This normally represents the R_p of the output tube or tubes as viewed from the secondary terminals of the output transformer.

The apparent generated voltage as viewed from the secondary terminals is e_G. The transient solution, however, is that current which flows in the network when e_G is zero, and subject to whatever initial conditions we seek to impose.

This circuit has been solved innumerable times; the current flow is oscillatory in nature, and of a frequency and decrement determined by the L, C, and R of the circuit. In particular, if

$$R = \sqrt{L_{me}/C_{me}} = \frac{1}{2\pi f_r C_{me}} \qquad (10)$$

where f_r is given by Eq. (*8*), and R is the resistance paralleling L_{me} and C_{me}, then the circuit is critically damped. This means that the natural frequency is zero, or the circuit is no longer oscillatory; physically the loudspeaker has no hangover effect. Of course R can be less than the value given by Eq. (*10*); the latter merely gives the maximum permissible value of R.

An inspection of *Fig.* 3. indicates that R must represent R_{me} paralleled by $(R_{vc}+R_G)$, hence if R_{me} is greater than the value required by Eq. (*10*), $(R_{vc}+R_G)$ must be a low enough shunt to provide in conjunction with R_{me} the critical damping necessary.

It will be recalled from Eq. (*5*) that if the mechanical damping (R_s+R_a) is low, R_{me} will be correspondingly high. An example which is to follow will show that usually the mechanical damping (R_s+R_a) is very low, so that it can be

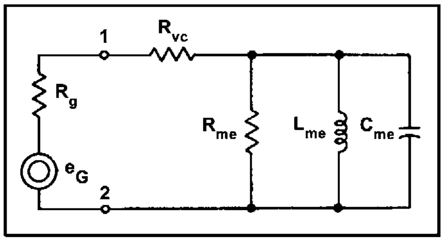

Fig. 3. Circuit of Fig. 2 with addition of generator.

expected that R_{me} will be relatively very high; much higher than will provide critical damping.

From this it follows that $(R_{vc}+R_G)$ must be a sufficiently low shunt to satisfy the critical damping condition given by Eq. (*10*). However, it is possible that the voice coil resistance R_{vc} is itself so high that Eq. (*10*) cannot be satisfied. In the usual case R_{vc} is not too high, but the maximum value left for R_G to assume can be quite low. In such a case a large amount of inverse voltage feedback may be necessary to reduce the source impedance to the requisite low value.

Numerical Example

The following numerical example will serve to illustrate the above analysis. Suppose we take a 16-inch cone type loudspeaker, whose mass is 40 grams, plus 4 grams for the voice coil. Assume further that the compliance of the suspension is $C_s = 3.2\times 10^{-7}$ cm/dyne, and that the mechanical resistance is 2400 mechanical ohms.

To the mass of the cone and voice coil must be added that of the mass of the air. In the neighborhood of 25 cps or so, Olson[2] gives the reactance of the air load as 7500 mechanical ohms. The corresponding mass is

$$M_a = \frac{7500}{2\pi\times 25} = 48 \text{ grams}$$

Hence the total mass is

$$M_t = 40+4+48 = 92 \text{ grams}$$

The resonant frequency is, by Eq. (*8*)

$$f_r = \frac{1}{2\pi\sqrt{92\times 3.2\times 10^{-7}}} = 29.3 \text{ cps}$$

which is close to the value of 25 cps initially used to calculate the air mass.

The air also imposes a certain amount of damping in the form of radiation resistance. This is a rapidly varying function of frequency; from Olson's book we find it to be 600 mechanical ohms at 29 cps. Hence the total mechanical damping is

$$R_s + R_a = 2400 + 600 = 3000 \text{ mech. ohms.}$$

Now suppose the flux density B is 10,000 gauss, and the length l of voice coil conductor is 1500 cm. Assume further that the voice coil resistance R_{vc} is 10 ohms.

Then, from Eq. (*5*), we have

$$R_{me} = \frac{(1500\times 10^4)^2\times 10^{-9}}{3000} = 75 \text{ ohms}$$

$$C_{me} = \frac{92}{(1500\times 10^4)^2\times 10^{-9}} = 409\ \mu f$$

$$L_{me} = (3.2\times 10^{-7})\ (1500\times 10^4)^2\times 10^{-9} = 0.072 \text{ henry}$$

Observe how large C_{me} is even though the mass responsible for this capacitive effect is only 92 grams.

For critical damping, the total resistance shunting L_{me} and C_{me} must be, by Eq. (*10*):

$$R = \sqrt{\frac{.072}{409\times 10^{-6}}} = 13.3 \text{ ohms}$$

Since R_{me} is one branch in parallel with R_{vc} plus the generator resistance, and this all totals 13.3 ohms, the voice coil branch must be

$$R_e = \frac{R_{me}\times R}{R_{me}-R} = \frac{75\times 13.3}{75-13.3} = 16.18 \text{ ohms}$$

Since the voice coil resistance R_{vc} is 10 ohms, the generator or source resistance, as viewed from the secondary terminals of the output transformer, must be

$$R_G = 16.18 - 10 = 6.18 \text{ ohms.}$$

Although this is a low value, it is by no means prohibitively low. For example,

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[2] H. F. Olson, "Elements of Acoustical Engineering," p. 126. D. Van Nostrand Co. New York.

if in the case of a single-ended triode output stage, $R_L = 2R_p$, then at the secondary terminals R_p should reflect as half of the voice coil load, if R_{vc} is 10 ohms, the reflected tube resistance R_G would be $10/2 = 5$ ohms. In short, a triode tube may be expected to act as critical damping in conjunction with the voice coil resistance.

In the case of a pentode tube, R_p is so high that no damping can be expected from it unless inverse voltage feedback is employed to an extent sufficient to lower the apparent source resistance to the required degree.

However, note that all this depends upon how low R_{vc} is compared to the length of wire used, and also how high the flux density B is. If the product (Bl) is low, both R_{me} and R may come out so low that R_{vc} alone may be in excess of that which paralleling R_{me}, will give the required value of R for critical damping. This means that even if the source resistance is zero, R_{vc} is too large and will not permit critical damping to be obtained.

Experimental Determination of Circuit Constants

It is possible to measure the motional impedance by simple electrical means, and from these measurements to determine the critical damping required. Since the measurements are to be made at the very low audio frequencies, ordinary iron vane meters can be used if so desired, and even a d.c. measurement of the voice coil resistance should be sufficient to furnish the value of R_{vc}.

If, however, it is desired to determine this quantity at the resonant frequency of the cone, or at any rate at some a.c. frequency, then the cone should be clamped so that it does not vibrate and generate a c.e.m.f., thereby furnishing a motional impedance value.

To measure the motional impedance, a set-up such as that indicated in *Fig.* 4 can be used. The audio oscillator wave shape should be reasonably free of harmonics, and the audio amplifier should be capable of furnishing several watts of power without distorting. The ammeter can be of the iron-vane type, and should read one ampere or less at full scale. The voltmeter is preferably of a high-impedance type. A preliminary run should be made to determine the resonant frequency of the cone and its suspension. This is done by varying the frequency upward in steps starting from say, 20 cps, and noting E and I at each step. Their quotient is the impedance seen looking into the voice coil. This should be done with the field fully energized if it is of the electrodynamic type.

At the mechanical series resonant frequency of the cone, I will drop to a very low value, and E will tend to rise. In short, the quotient will be relatively large, and will represent $(R_{vc} + R_{me})$. If the value found previously for R_{vc} is subtracted from this reading, R_{me} is obtained. The resonant peak is normally quite sharp for reasons that will be explained further on.

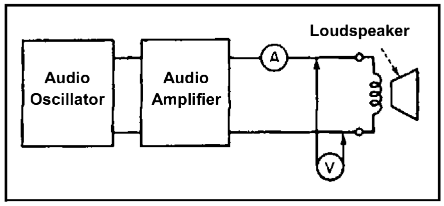

Fig. 4. Circuit arrangement used for making measurement of motional impedance.

In order to determine the value of critical damping R, it would appear necessary to measure L_{me} and C_{me}. However, R can also be determined by measuring the Q of the circuit; critical damping is obtained if $Q = 1$. To measure Q, ordinarily one merely has to plot the selectivity curve for the device, whether this curve represents transmission, impedance, admittance, or whatever other quantity gives this characteristic.

In the case of the loudspeaker, the resonant Q of the circuit is determined by the impedance as measured across R_{me}, L_{me}, and C_{me} in *Fig.* 3, with the electrical resistance $(R_{vc} + R_G)$ in parallel with R_{me}. In other words, the condition given by Eq. (*10*) for critical damping is also the condition for the resonant Q to be unity, where Q_r is in general determined by $\omega_r C_{me}$, and R_{me} and $(R_{vc} + R_G)$ in parallel.

Unfortunately, measurements must be made at terminals 1–2 in *Fig.* 3, since there are no accessible terminals across Z_{me}. The resulting impedance, Z_t, represents R_{vc} in series with Z_{me}, that is—with R_{me}, C_{me}, and L_{me} all in parallel. To find the above-defined resonant Q therefore requires some preliminary analysis, which will be given below.

Experimentally, however, all one has to do is to measure the impedance Z_t at and around resonance over a range including frequencies at which Z_t drops to $1/\sqrt{2}$ of its value at resonance (where it has the maximum value $R_{vc} + R_{me}$). Then, knowing the two frequencies at which this occurs, as well as the resonant frequency f_r, Q can be calculated. Once Q is known, the necessary value of R can be found, and then the maximum permissable generator resistance R_G.

Let us therefore proceed to evaluate this impedance. The impedance looking to the right into terminals 1–2 of *Fig.* 3 can be calculated from the circuit elements shown. It is:

$$|Z_t| = \sqrt{\frac{1 + Q_r^2 (1-\rho^2)^2}{\left(\frac{1}{R_{me}+R_{vc}}\right)^2 + \frac{Q_r^2}{R_{vc}^2}(1-\rho^2)^2}} \qquad (11)$$

where Q_r is the resonant Q of the circuit *if terminals 1–2 of Fig. 3 were short-circuited;* i.e.,

$$Q_r = \omega_r C_{me} R = R/\omega_r L_{me} \qquad (12)$$

in which R represents R_{me} and R_{vc} in parallel, and ω_r is the resonant angular velocity of L_{me} and C_{me}. Furthermore,

$$\rho = f/f_r \qquad (13)$$

where f is the frequency at which Z_t is being measured, and f_r is the resonant frequency; in short, ρ represents the fractional deviation from the resonant frequency.

In particular, if $\rho = 1$, $(f = f_r)$, Eq. (*11*) reduces to

$$Z_t = R_{me} + R_{vc} \qquad (14)$$

which is correct from an inspection of *Fig.* 3, since at the resonant frequency L_{me} and C_{me} form a negligibly high shunt impedance across R_{me}, so that Z_t becomes $R_{me} + R_{vc}$, as stated above.

Furthermore, if $\rho = 0$, $(f = 0)$, or $\rho = \infty$, $(f = \infty)$, Z_t becomes equal to R_{vc} alone, as is also clear from *Fig.* 3, since L_{me} is a short circuit across R_{me} at $f = 0$, and C_{me} is the short circuit at $f = \infty$.

If Eq. (*11*) is solved for Q_r in terms of the other variables, there is obtained:

$$Q = \frac{1}{(1-\rho^2)}\sqrt{\frac{\left(1 - \frac{Z_t}{R_{me}+R_{vc}}\right)^2}{(Z_t/R_{vc})^2 - 1}} \qquad (15)$$

Now suppose the frequency is varied, which is the same as saying ρ is varied until Z_t drops to $1/\sqrt{2}$ of its maximum value; i.e.,

$$Z_t = \frac{R_{me} + R_{vc}}{\sqrt{2}} \qquad (16)$$

If this value is substituted in Eq. (*15*), together with the corresponding specific value of ρ, call it ρ_1, there is obtained:

$$Q_r = \left(\frac{1}{1-\rho_1^2}\right)\sqrt{\frac{1}{\left(\frac{R_{me}+R_{vc}}{R_{vc}}\right)^2 - 2}} \qquad (17)$$

If $\left(\frac{R_{me}+R_{vc}}{R_{vc}}\right)^2 \gg 2$, say twenty times two, then Eq. (*17*) simplifies to

$$Q_r = \left(\frac{1}{1-\rho_1^2}\right)\left(\frac{R_{vc}}{R_{me}+R_{vc}}\right) \quad (18)$$

If ρ_1 is nearly unity, the difference between the actual frequency f_1 and the resonant frequency f_r is small; that is,

$$\Delta f_1 = f_r - f_1$$

or

$$\Delta f_1 = f_1 - f_r$$

(depending upon whether the excursion is below or above the resonant frequency) is small. This is usually the case, and under such conditions Eq. (*18*) can be rewritten as

$$Q_r = \left(\frac{f_r}{2\Delta f_1}\right)\left(\frac{R_{vc}}{R_{me}+R_{vc}}\right) \quad (19)$$

which can form the basis of our experimental procedure as well as Eq. (*18*) can. If we re-write Eq. (*19*) as follows:

$$\frac{2\Delta f_1}{f_r} = \left(\frac{1}{Q_r}\right)\left(\frac{R_{vc}}{R_{me}+R_{vc}}\right) \quad (20)$$

we recognize the form to be similar to that of the well-known resonance formula, in which the fractional bandwidth $(2\Delta f/f_r)$ for the half-power points is the reciprocal of the resonant Q of the circuit. Eq. (*20*) shows that owing to the point in the circuit at which the measuring instruments are introduced, the fractional bandwidth is reduced bv a factor $R_{vc}/(R_{me}+R_{vc})$, which would not occur if the measurements could be made across the motional impedance component itself.

The significance of Eq. (*20*) is that even though Q_r for a loudspeaker system may be less than unity, the fractional bandwidth will nevertheless be quite small because of the reducing factor $R_{vc}/(R_{me}+R_{vc})$. This makes the measurements somewhat critical and requires a well-calibrated frequency scale on the audio oscillator.

To see how this all fits together, let us proceed with an experimental run. The first measurement is R_{vc}; this is found to be 10 ohms. Then the test setup of *Fig.* 4 is connected to the loudspeaker and the frequency varied from say 20 to 50 cps.

At 29.3 cps the current is found to dip to a minimum value of 83.2 ma, and the voltmeter reads 7.07 volts. The impedance is resistive, and of a value

$$R_{vc} + R_{me} = 7.07/.0832 = 85 \text{ ohms.}$$

Hence $Z_t = R_{me} = 85 - 10 = 75$ *ohms.*

Now the frequency is varied above and below 29.3 cps to the point where Z_t drops to $85\sqrt{2}/2 = 60.1$ ohms, as found by taking the ratio of the voltmeter to ammeter readings in exactly the same way as $(R_{vc}+R_{me})$ was calculated.

Suppose the frequency drops from 29.3 to 26.7 cps before $Z_t = 60.1$, and rises to 31.9 cps before this value is reached once more. Then $\Delta f_1 = 29.3 - 26.7 = 2.6$ cps, or $\Delta f_1 = 31.9 - 29.3 = 2.6$ cps, and

$$2\Delta f_1/f_1 = 2 \times 2.6/29.3 = 0.1776.$$

We can now use Eq. (*19*) to calculate Q_r. Thus

$$Q_r = \left(\frac{1}{0.1776}\right)\left(\frac{10}{85}\right) = 0.663.$$

This is the Q of the loudspeaker circuit if the source impedance R_G were zero. Since Q_r is less than unity, it can be raised to that figure by allowing R_G to be greater than zero. It remains to calculate this value.

We have, for a parallel resonant circuit such as in *Fig.* 3, that

$$Q = \omega_r C_{me} R \quad (21)$$

where R is the resistance shunting C_{me} and L_{me} (*Fig.* 3), and is therefore R_{me} in parallel with $(R_{vc}+R_G)$. However, in the measurement and calculation yielding Q_r, R_G is essentially zero, and R represents simply R_{me} and R_{vc} in parallel.

We seek a value R', such that the Q is equal to unity; i.e.,

$$1 = \omega_r C_{me} R'$$

or

$$R' = 1/\omega_r C_{me} \quad (22)$$

Substituting from Eqs. (*21*) and (*20*) in Eq. (*22*), we obtain

$$R' = \frac{R}{Q_r} = \frac{R_{me}R_{vc}}{R_{me}-R_{vc}} \times \frac{2\Delta f_1}{f_r} \times \frac{R_{me}+R_{vc}}{R_{vc}} = \frac{2\Delta f_1}{f_r} R_{me} \quad (23)$$

This represents R_{me} paralleled by $(R_{vc} + R_G)$, hence

$$R_{vc} + R_G = \frac{R'R_{me}}{R_{me}-R'} \quad (24)$$

and

$$R_G = \frac{R'(R_{me}+R_{vc}) - R_{vc}R_{me}}{R_{me}-R'} \quad (25)$$

Hence let us finish our experimental determination of R_G. From Eq. (*23*) we can find R'. If we use the last form, we have

$$R' = \frac{2\Delta f_1}{f_r} R_{me} = (0.1776)\ (75) = 13.31$$

ohms and from Eq. (*25*) we obtain

$$R_G = \frac{(13.31)\ (85) - (10)\ (75)}{(75 - 13.31)} = 6.19 \text{ ohms}$$

which of course checks the previous computation from the values for the mechanical constants, since it is the same loudspeaker that we have under consideration.

An Alternative Viewpoint

It is possible to reflect the electrical constants into the mechanical side of the circuit, and obtain an alternative viewpoint of the behavior of the system as a whole. The results, so far as the low-frequency resonance is concerned, are the same, as will be shown. There is, however, another advantage of this alternative point of view with regard to the acoustical design; it permits the designer to incorporate the electrical constants into the acoustical design with a corresponding improvement in the performance of the loudspeaker.

First, the design formulas will have to be presented. The electrical impedance of the source and the voice coil appears in the mechanical side of the system as follows:

$$Z_{em} = \frac{(Bl)^2 \times 10^{-9}}{Z_e} \quad (26)$$

where Z_{em} is the mechanical impedance equivalent to the actual electrical impedance Z_e, and B and l have the same significance as before.

The output stage and voice coil in series with it exhibit essentially an inductive and resistive impedance at the higher audio frequencies. The inductance is the leakage inductance of the output transformer, plus that of the voice coil, and the resistance is the apparent source resistance R_G as viewed from the secondary terminals of the output transformer, plus that of the voice coil.

Hence, set

$$Z_e = R_e + j\omega L_e \qquad (27)$$

where $R_e = R_{vc} + R_G$ (see *Fig.* 3), and L_e is the inductance defined above, and which we have not heretofore taken into account. At the lower audio frequencies $j\omega L_e$ can be ignored, whereupon Z_e reduces to R_e.

However, if Eq. (*27*) be substituted in Eq. (*26*), and then numerator and denominator divided by $(Bl)^2 \times 10^{-9}$, as before, there is obtained:

$$Z_{em} = \cfrac{1}{\cfrac{R_e}{(Bl)^2\,10^{-9}} + \cfrac{j\omega L_e}{(Bl)^2\,10^{-9}}} \qquad (28)$$

If we consider $R_e/(Bl)^2\,10^{-9}$ as a mechanical conductance G_{em}, so that its reciprocal R_{em}, a mechanical resistance, is given

$$R_{em} = 1/G_{em} = 1/[R_e/(Bl)^2\,10^{-9}] \qquad (29)$$

and if we further consider $L_e/(Bl)^2\,10^{-9}$ as a mechanical compliance C_{em}, then we can write Eq. (*28*) as

$$Z_{em} = \frac{1}{(1/R_{em}) + j\omega C_{em}} \qquad (30)$$

or the electrical resistance and inductance in series appear in the mechanical system as a mechanical resistance and compliance in parallel. Hence, the counterpart of *Fig.* 3 is that shown in *Fig.* 5: a constant-velocity mechanical generator (counterpart of a constant-voltage electrical generator) feeds the mechanical resistance R_{em} equivalent to the electrical resistance R_e, in parallel with the mechanical compliance C_{em} equivalent to the electrical inductance L_e, and the actual mechanical impedance

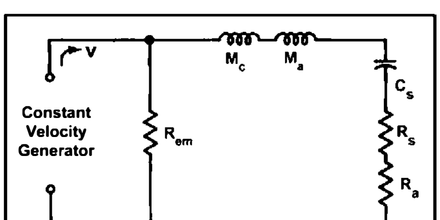

Fig. 5. Counterpart of Fig. 3, in mechanical terminology.

Z_m of the loudspeaker. This circuit has interesting implications both at the low- and at the high-frequency ends of the audio spectrum.

Consider the low-frequency end first. In this range C_{em} can be ignored, and

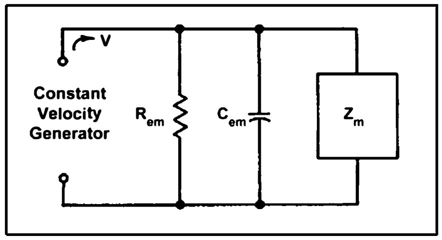

Fig. 6. Equivalent circuit corresponding to Fig. 5, showing damping due to electrical resistance.

Z_m consists, in the case of a direct-radiator cone loudspeaker, of the elements shown in *Fig.* 1. Hence *Fig.* 5 becomes circuit shown in *Fig.* 6. Here it is apparent how the electrical resistance R_e introduces in effect damping into the mechanical circuit by its transformed element R_{em}.

From *Fig.* 6 it is apparent that for critical damping,

$$(R_{em} + R_s + R_a) = \sqrt{(M_c + M_a)/C_s} \qquad (31)$$

or alternatively, that the mechanical Q at resonance is unity:

$$Q_m = \frac{\omega_r(M_c + M_a)}{R_{em} + R_s + R_a} = 1 \qquad (32)$$

from which the required electrical resistance must be

$$R_{em} = \omega_r(M_c + M_a) - (R_s + R_a) \qquad (33)$$

Once R_{em} is evaluated from Eq. (*33*), the equivalent electrical resistance R_e can be found from Eq. (*29*). Then the voice coil resistance R_{vc} is subtracted from R_e to yield the maximum permissible value of apparent generator resistance R_G.

Let us try out these formulas on the loudspeaker constants given previously. It will be recalled that the total mass (including that of the voice coil) was 92 mechanical ohms. This will be the value used for $(M_c + M_a)$. The resonant frequency was 29.3 cps, so that $\omega_r = 2\pi\ 29.3$ *rad./sec.* Also $(R_s + R_a)$ came out to be 3000 mechanical ohms.

Hence, if the appropriate values be substituted in Eq. (*33*), there is obtained:

$$R_{em} = (2\pi\,29.3)(92) - (3000)$$
$$= 16{,}980 - 3{,}000 = 13{,}980 \text{ mech. ohms}$$

Now, from Eq. (*29*), the equivalent electrical resistance R_e that is required to obtain critical damping is

$$R_e = \frac{(Bl)^2\,10^{-9}}{R_{em}} = \frac{(10000 \times 1500)^2}{13980}$$
$$= 16.13 \text{ ohms (electrical)}.$$

Since the voice coil resistance R_{vc} is 10 ohms, the apparent source resistance can be

$$R_G = 16.13 - 10 = 6.13 \text{ ohms}$$

which checks our previous calculations, as it should.

High-Frequency Response

The same equivalence between circuits can be utilized in the analysis of a high-frequency tweeter unit of the horn type. This employs a small diaphragm and voice coil, which feeds the cavity in front of it that leads to an exponential horn. The physical arrangement is shown in cross-section in *Fig. 7.* Here m_d represents the mass of the diaphragm and associated voice coil; C_a, the compliance of the air chamber in front of the diaphragm, necessary to furnish clearance for the motion of the diaphragm and useful in building out the mechanical circuit; and finally r_h represents the acoustical resistance of the horn throat in the frequency range above its low-frequency cutoff point.

The mechanical circuit has been analyzed many times in the past; it is given in *Fig.* 8. The resistance r_h is that of the throat of the horn, and is equal to the area of the throat in sq. cm. multiplied by 41.4 mech. ohms, which is the radiation resistance of air per sq. cm. A_d is the area of the diaphragm; in conjunction with A_h it forms a kind of hydraulic press which is the mechanical counterpart of an electrical transformer. The step-down ratio is A_d to A_h; conversely r_h is reflected to the diaphragm as an equivalent resistance r'_h such that

$$r'_h = (A_d/A_h)^2 \qquad (34)$$

The reflected resistance r'_h shunts the air chamber compliance C_a. This is because the lower r'_h is, the more readily can it relieve the pressure built up in the air chamber by the motion of the diaphragm. This is exactly analogous to the reduction in the charge and voltage across a capacitor when it is shunted by a low resistance.

From *Fig.* 8 the loudspeaker unit is recognized as forming an L-section low-pass filter. For proper transmission up to the cut-off frequency, it is necessary that

$$r'_h = \sqrt{M_d/C_a} \qquad (35)$$

The cutoff frequency is given by

$$f_c = \frac{1}{\pi\sqrt{M_d C_a}} \qquad (36)$$

If twice the mass ($2M_d$) were employed and another compliance C_a placed at the left end, a π-section filter would be obtained, to which Eqs. (*35*) and (*36*) would apply equally well. In short, the same cutoff frequency can be obtained

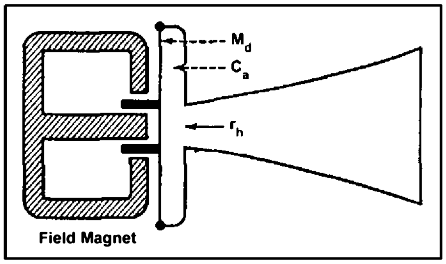

Fig. 7. Physical arrangement of mechanical elements of a high-frequency horn and unit.

for double the mass, if a compliance is placed at the other end of it.

If only the mass is doubled, then the cutoff frequency is reduced to 70.7 per cent of its original value, as is evident by substituting $2M_d$ for M_d in Eq. (*36*). (The corresponding changes in r'_h are not of importance as they involve merely a change in the ratio of A_d to A_h.

For a given high-frequency cutoff and power-handling ability of the speaker, the diaphragm mass M_d comes out to be a certain amount. If M_d can be kept the same, and yet a compliance placed at the front end, the high-frequency cutoff can be extended to $\sqrt{2}$ or 1.414 times its original value without altering the speaker's power handling ability. Hence it is of interest to see how this can be done.

At the higher audio frequencies, the output transformer appears at its secondary terminals essentially as a series inductance L_L (its leakage inductance). The power amplifier tubes, as reflected to the secondary of the transformer appear as a resistance R_G in series with L_L. To this must be added the voice coil resistance R_{vc} and its inductance L_{vc} in series with R_G and L_L. Hence finally the electrical current appears as

$$Z_e = R_e + j\omega L_e$$

where

$$R_e = R_G + R_{vc} \qquad (37)$$

and

$$L_e = L_L + L_{vc}$$

Figure 5 and Eq. (*30*) show how these appear in the mechanical circuit. The mechanical impedance Z_m is in this case illustrated by *Fig.* 8, so that finally in *Fig.* 9 is given the complete mechanical circuit including the equivalent electrical circuit parameters.

Here, in accordance with Eq. (*29*)

$$R_{em} = \frac{1}{R_e/(Bl)^2\,10^{-9}} \qquad (29)$$

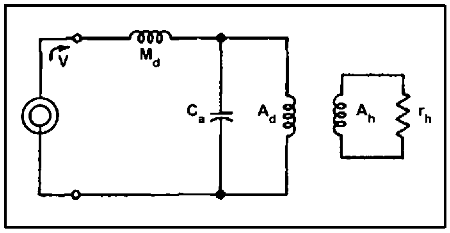

Fig. 8. Mechanical circuit of high frequency speaker.

and this should match r'_h for maximum power transfer, or

$$R_e = (Bl)^2 \times 10^{-9}/r'_h \qquad (38)$$

from which the apparent source impedance should equal

$$R_G = R_e - R_{vc} = \frac{(Bl)^2 \times 10^{-9}}{r'_h} - R_{vc} \qquad (39)$$

The apparent mechanical compliance equivalent to the electrical inductance is indicated by Eqs. (*28*) and (*30*), namely:

$$C_{em} = L_e/(Bl)^2 \times 10^{-9} \qquad (40)$$

However, in order to convert the L-section mechanical low-pass filter of *Fig.* 8 into the π-section low-pass filter of *Fig.* 9, it is necessary that

$$C_{em} = C_a = \frac{L_e}{(Bl)^2 \times 10^{-9}} \qquad (41)$$

If such coordination in electrical and mechanical design be accomplished, a 41 per cent increase in frequency response may be expected over the case of no electrical inductance at all. Of course, in actual practice the electrical system inherently has inductance and resistance so that the "building-out" of the L-section into a π-section tends to take place; all that it is desired to point out here is that the electrical and mechanical circuit elements can be coordinated so as to improve the performance rather than to have a haphazard relationship to one another, and that furthermore, electrical inductance is not necessarily an undesirable characteristic in the output stage, but can serve a useful purpose.

Undoubtedly, in most systems the inductance – particularly that of the voice coil itself – is too high and produces a C_{em} in excess of C_a. Also, R_{em} may be too low compared to r'_h because of excessive electrical resistance $R_G + R_{vc}$. However, this serves to counterbalance an excessive value for C_{em} and therefore tends to smooth out the response.

The interested experimenter can calculate the actual response of the network shown in *Fig.* 9 on the basis that it is not a truly terminated low-pass filter section, since a resistance such as r'_h is but a nominal match over the pass

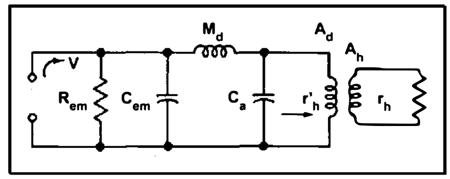

Fig. 9. Conversion of Fig. 8 circuit to pi-section equivalent.

band, and is a considerable mismatch near the cutoff frequency, where the termination should approach zero. He can also calculate the response for his actual speaker and amplifier output stage, in order to see directly the effect of varying, for example, the electrical circuit constants.

Conclusion

A method of coordinating the motional impedance of a loudspeaker with the electrical impedance has been presented here with the object of reducing "hangover" effects and objectionable transients in general at the low-frequency resonance of the speaker.

An experimental method has also been presented to enable the necessary measurements to be made in order that the correct source impedance be obtained for critical damping of the system. The method requires merely an audio oscillator, an a-c voltmeter and an a-c ammeter in order to determine the impedance over a range of frequencies. From the shape of the impedance curve the Q of the system can be determined, and from the value of voice coil and motional impedance at resonance, the requisite source resistance for critical damping can be calculated.

An alternative method based on viewing the electrical constants from the mechanical side was then presented, and it was shown that this method led to the same answers as above. Finally, it was shown by this method how inductance and even resistance in the electrical system could be put to use to obtain a coordinated system in the case of a high-frequency loudspeaker.

ACOUSTIC DAMPING FOR LOUDSPEAKERS*

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The fundamental resonance of loudspeakers is recognized by many as a source of annoyance. Usually this resonance can be damped electrically by suitable selection of the amplifier impedance.[1] What is less well-known is that damping can also be achieved by acoustical means incorporated into the loudspeaker or the enclosure. This paper deals with the theory and methods for providing acoustic damping.

Electrical damping requires the use of a low impedance amplifier. When this type of source is not available, and in the absence of other damping means, performance of the system may be seriously impaired. For example, a typical home radio or record player has a pentode output stage without inverse feedback. This type of output stage is known to provide a source of high impedance. A loudspeaker driven from this source will exhibit a resonant condition which may cause poor transient response or "hang-over" and it may be responsible for the acoustic feedback in record players. Acoustic damping may be found helpful in this instance.

Another use for acoustic damping will be found in the design of high-fidelity systems, where frequent attempts are made to improve damping by lowering the amplifier impedance to a value approaching zero. There is a limit to the amount of damping which may be obtained in this manner.[2, 3] Furthermore, electrical damping per se is not very effective in eliminating the resonance in cabinets with reflex ports. Acoustic damping can readily provide such additional damping as may be required.

Much effort and circuitry has been devoted to attempts to obtain damping from the electrical side. By contrast, the use of acoustic damping has been given little attention. In this paper we outline a simplified theory of acoustic damping for loudspeakers and enclosures. To provide a rational basis for the design of acoustic damping the acoustic constants of the loudspeaker and the enclosure must be known. Thence, an equivalent electrical circuit can be set up and the damping resistance may be determined experimentally by adjusting the electrical circuit constants. Keeping this approach in mind, first we derive the resonant frequency equations of a loudspeaker in a flat baffle and in an enclosure. Next, from these two equations we determine the acoustic mass and compliance of the loudspeaker cone. Thirdly, we set up an equivalent electrical circuit and determine the required damping resistance. And finally we build the acoustic damping into the enclosure and test its acoustic performance.

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II. EQUIVALENT CIRCUITS OF LOUDSPEAKERS AND ENCLOSURES

The equivalent circuit of a loudspeaker installed in an "infinite" flat baffle is shown in Fig. 1. In actual practice, of course, the baffle need not be infinite, but only considerably larger than the loudspeaker. The acoustic mass of the moving system (consisting of the voice coil and the piston diaphragm) is represented as an inductance L_{AM}; the acoustic compliance of the elastic suspension is shown as a capacitor C_{AM}. These terms can be calculated from measurements of the resonant frequency, as shown later in this paper. The driving pressure due to the voice-coil force is represented by a constant voltage e_o. We assume that the loudspeaker is driven from a high impedance source. Therefore, the damping factor reflected from the electrical side Z_{AP} equals zero. Either side of the piston is confronted by the acoustic impedance of the medium, Z_{AP}. This impedance is shown grounded because, from the acoustical point of view, it has only one available driving point terminal.[4]

It will suffice our purpose to represent Z_{AP} by a parallel combination of an inductance L_{AP} and a resistance R_{AP}.[5] These elements can be calculated in terms of the properties of the medium and the effective radius of the piston <u>r</u>:

$$L_{AP} = \rho/\sqrt{2}\pi r = 0.00027/r \quad \text{grams-cm}^{-4} \text{ "Acoustic henrys"} \qquad (1)$$

$$R_{AP} = \rho c_v/\pi r^2 = 42/\pi r^2 \quad \text{dynes-sec-cm}^{-5} \text{ "Acoustic ohms"} \qquad (2)$$

where, r is measured by the perpendicular distance from the axis to the mid-point of the flexible annulus of the cone, cm.

ρ is the density of air, grams per cc = 0.0012 for normal atmosphere.

c_v is the velocity of sound in air, = 34,400 cm per sec. in normal atmosphere.

These equations are given for circular pistons; however, they may be used for pistons of other shapes, in terms of circular pistons of equivalent area.

R_{AP} predominates at high frequency; L_{AP} predominates at low frequency. Loudspeaker resonances occur at low frequency, and hence, R_{AP} can be neglected in the resonant frequency equation:

$$(2\pi f_1)^2(L_{AM} + 2L_{AP})C_{AM} = 1 \qquad (3)$$

Resonant frequency can be measured by connecting the voice-coil to a variable frequency oscillator through a high series resistance (to approach a constant current condition) and measuring the voice-coil voltage with a high-impedance voltmeter. At the resonant frequency the voice-coil voltage is a maximum.

When a loudspeaker is installed in a shallow cabinet with open back, its resonant frequency is essentially the same as with a flat baffle of similar dimension.

Next, consider the loudspeaker in an enclosure as shown in Fig. 2. The simplest enclosure is a closed box with a hole of an appropriate size to accommodate the loudspeaker. The design of such enclosure has been treated in detail by Beranek[6] and others. Our purpose is limited to providing its equivalent circuit elements.

To minimize reflections at high frequency, the enclosure may be lined with sound absorbing material such as Fiberglas or Ozite. These materials, when mounted against the walls of the enclosure, offer a negligible amount of sound absorption at low frequency and contribute little to the damping. Therefore, the low frequency impedance confronting the piston is predominantly reactive, and it can be represented by an inductance L_{AB} in series with a capacitance (or compliance) C_{AB}. For rectangular boxes, L_{AB} is given by the semi-empirical equation:

$$L_{AB} = \frac{4}{\pi^2} \frac{\rho \ell}{A_B} \left(\frac{A_B}{A_P} \right)^{2/3} \text{ grams-cm}^{-4} \text{ ("acoustic henrys")} \qquad (4)$$

The equation for acoustic compliance is well-known.[7]

$$C_{AB} = \frac{V}{\rho c_v^2} = \frac{V}{1.41} \times 10^{-6} \text{ cm}^5\text{-dyne}^{-1} \text{ ("acoustic farads")} \qquad (5)$$

where ρ is the density of air - 0.0012 grams per cc for normal atmosphere

A_B is the cross-sectional area of the box, sq. cm.

ℓ is the depth of the box, cm.

A_P is the effective area of the piston, sq. cm.

V is the volume of the box, cu. cm.

Equation (4) holds quite well under these conditions:

1. The loudspeaker is about equally spaced from all the points at the inside periphery of the box.

2. The ratio of the box area A_B to the effective piston area A_P is not in excess of about 10:1.

3. The depth ℓ is about $0.7 \sqrt{A_B}$.

The use of the word "about" is intended to signify that the conditions indicated can be violated considerably without undue error.

The air-load impedance at the front of the cone can be assumed to be given approximately by equations (1) and (2). The resistive component may be neglected in the equation for resonant frequency, as before. The resonant frequency f_2 of this system is contained in the equation:

$$(2\pi f_2)^2(L_{AB} + L_{AM} + L_{AP}) \frac{C_{AM}C_{AB}}{C_{AM} + C_{AB}} = 1 \tag{6}$$

The resonant frequencies measured in the baffle (f_1) and in the box (f_2) enable us to calculate the acoustical constants of the loudspeaker. From equations (6) and (3) the following expressions are derived:

$$L_{AM} = \frac{1 - (2\pi f_2)^2 C_{AB}(L_{AB} + L_{AP}) + 2(2\pi f_1)^2 L_{AP} C_{AB}}{4\pi^2(f_2^{\,2} - f_1^{\,2})C_{AB}} \quad \text{gram-cm}^{-4} \text{ ("acoustic henrys")} \tag{7}$$

$$C_{AM} = \frac{1}{(2\pi f_1)^2(L_{AM} + 2L_{AP})} \quad \text{cm}^5\text{-dyne}^{-1} \text{ ("acoustic farads")}, \tag{8}$$

Having thus determined all the important circuit elements of the loudspeaker and the simple enclosure, we are ready to set up the equivalent electrical circuit and determine the damping resistance. Before proceeding, however, we must digress briefly to introduce the subject of acoustic resistance.

III. ACOUSTIC RESISTANCE

Acoustic resistance is encountered when sound is made to flow through thin crevices or slits. Consequently, any broad surface member may be converted into an acoustic resistance by perforating it with small holes, slits, etc., which are calculable by well-known equations.[8] Cloth, felt, Ozite, etc. mounted on a suitable supporting member in the path of sound flow constitutes a simple and inexpensive method for providing good acoustic resistance. The acoustic resistance of these materials cannot be calculated, but it can be measured easily by causing air to flow at a known rate through a given area of material and measuring the pressure drop.

An instrument designed to perform this measurement is shown in Fig. 3. The material is held between two disks which provide an aperture having a known area. The rate of air flow is adjusted by means of the lower gauge which indicates the pressure drop across a standard slit. The pressure drop across the material is shown on the upper gauge which is calibrated to read the specific acoustic resistance of the material in acoustic ohms per sq. cm.

Cloth is an especially satisfactory material and it can be selected to provide any desired specific resistance. For example, the open weave fabrics designated "Ninon" and "Georgette" have resistances from 1/4 to 5.0 ohms per sq. cm. Most sateens, broadcloths, etc. have resistances between 5 and 75 ohms per sq. cm.; sailcloths and similar tightly woven fabrics have resistances upward of 75 ohms per sq. cm.

Usually the acoustic resistance element should have an area about equal to the area of the piston, or greater; this is to avoid constriction of the air flow which would have the effect of added acoustic mass. The specific resistance of the cloth to be employed is easily determined by multiplying the required acoustic damping resistance R_{AC} by the chosen area of the element in sq. cm. The cloth is supported by cementing to a heavy wire mesh or perforated grille to avoid motion in a diaphragm-like fashion. When a perforated grille is used, the effective area is, of course, the net area of the openings. The resistance may be adjusted experimentally, as by blocking some of the holes of the supporting member by means of Scotch tape or cement.

IV. ACOUSTIC DAMPING

Acoustic damping may be applied to loudspeakers and enclosures in many ways. Some of these are shown in Fig. 4. In general, the damping resistance must be coupled to the cone by a sufficiently small enclosure. In this manner, a good deal of the air displaced by the diaphragm is forced through the acoustic resistance resulting in a damping action. In the simplest case, A, the small enclosure, is the sole enclosure and the damping is placed at the back or one of the sides of the box. This small enclosure may be attached to a large baffle, as shown in B, to improve low frequency response. At C, a large enclosure is used, the small enclosure being provided within the larger enclosure. D is similar to C, except that a damped port is provided for reflex action. At E, a port has been added to the enclosure of A; and finally, F shows an arrangement similar to D in which both the speaker and the port derive a measure of damping from a single acoustic resistance element. Many other combinations are possible.

When an acoustic resistance element confronts the external medium, as in A or B for example, the air particles moving through the element are confronted with an acoustic impedance. The approximate value of this impedance may be estimated through the use of equations (1) and (2).

Since our purpose is to describe acoustic damping techniques rather than to evaluate the merit of various designs, only the simplest arrangement, namely that shown in 4-A, will be treated in detail. An experimental speaker system utilizing this arrangement is shown in Fig. 5. It consists of a medium-priced 8" loudspeaker installed in an enclosure with inside dimensions 14 x 14 x 9 inches deep. This enclosure may well be typical of those used in a medium-sized radio or P.A. system. The enclosure has interchangeable backs; one of them is solid, and the other is provided with an 11" diameter hole for installation of the acoustical damping resistance. The choice of this resistance by electrical circuit analysis is to be described.

First, the acoustic constants of this system were determined by the steps previously mentioned. The constants are shown in Table III.

Table III

Measured resonant frequency in box, with back removed	f_1 = 62 cps
Measured resonant frequency in box with hard back in place	f_2 = 98 cps
Measured effective radius of the piston . .	r = 8.5 cm
Acoustic mass due to air-load, from equation (1)	$L_{AP} = 0.032 \times 10^{-3}$
Acoustic resistance due to air-load, from equation (2)	$R_{AP} = 0.19$
Acoustic mass of the closed box, from equation (4)	$L_{AB} = 0.028 \times 10^{-3}$
Acoustic compliance of the closed box, from equation (5)	$C_{AB} = 20.4 \times 10^{3}\,\mu f$
Acoustic mass of the piston, from equation (7)	$L_{AM} = 0.147 \times 10^{-3}$
Acoustic compliance of the piston, from equation (8)	$C_{AM} = 31 \times 10^{3}\,\mu f$
Acoustic mass due to air-load upon damping screen, from equation (1)	$L_A = 0.019 \times 10^{-3}$ henrys
Acoustic resistance due to air-load upon damping screen, from equation (2)	$R_A = 0.07$ ohms

The equivalent circuit using these constants is shown in Fig. 6. The acoustic values are shown in parenthesis. Some of the acoustical elements have a much lower impedance than the electrical components available in the laboratory; in the equivalent circuit, therefore, the impedance of all the constants was multiplied by the factor 2000. The switch S_1 at position A portrays the action with the back removed, because the fluid from the back of the cone is able to enter the external medium. The switch in position B (open circuit) portrays the action with the solid back, since the fluid displaced by the cone is unable to enter the external medium. The switch at C causes the insertion of an adjustable resistance R_{AC} which portrays the acoustical damping resistance due to cloth when the damping back of the box is used. A simplifying assumption is made that the interaction between the back and the front radiation has no effect upon damping and that the air-load impedance at the back of the box remains the same with and without the damping resistance.

The optimum damping resistance was chosen by observing the transient response of the system. A steep pulse from a General Radio Strobotac was applied to a small (10 ohm) resistor "e" inserted in the circuit.

The Strobotac was adjusted to fire about 10 pulses per second. The transient potential developed across Z_{AP} was observed on the screen of an oscilloscope. (The circuit was rearranged to permit the single-ended Strobotac and Oscilloscope to be properly grounded.) Varying degrees of damping were obtained by adjusting R_{AC}. Some of the resulting transients are shown in Fig. 7. "A" is obtained with the switch at A to represent the open-box condition; "B" is obtained with the switch at B to represent the solid back condition; "C" is taken with the switch at C to represent the condition resulting from adjusting R_{AC} to 100 ohms. The corresponding acoustic resistance is 100/2000 = 0.05 ohms. If a greater degree of damping were desired, some of the remaining circuit elements would require alteration.

To provide 0.05 ohms acoustic damping in the loudspeaker enclosure, the damping cloth was mounted on a perforated metal screen with a total net area of 200 sq. cm. Cloth having a specific resistance of 10 acoustic ohms was used.

The actual acoustic performance of the system was tested with the aid of the Strobotac generator connected to the loudspeaker (Fig. 8). A high resistance was inserted in the circuit to simulate the action of a pentode output stage (without inverse feedback). The sound pressure generated immediately in front of the loudspeaker cone was detected by means of a Sound Level Meter Microphone. A flat amplifier was used to connect the microphone to the oscilloscope. Transients were observed with the open box (A), with solid back (B), and with damping back (C). The resulting transients are shown in Fig. 9. "A" is the open-box response; "B" is the response with the solid back attached; under these two conditions a hang-over tone is clearly heard together with the pulses. "C" is the response with the damped back attached. The pulses are heard clearly without the hang-over tone. These oscillograms and listening tests confirm the results predicted by the equivalent circuit.

V. ELECTRICAL VS. ACOUSTIC DAMPING

To compare the electrical damping with the acoustic damping, the following experiment was performed. The hard back was installed and the loudspeaker was connected to the pulse generator as before -- except that a small resistor was connected across the loudspeaker terminals to simulate the effect of low amplifier output impedance. The input was readjusted to compensate for the power loss in the resistor. The resulting transient is shown in Fig. 10A. This confirmed the fact that the action of the electrical damping is similar to that of the acoustic damping, although, in this instance, not quite as effective. Next, the solid back was again replaced with the damped back and the acoustic impedance was slightly adjusted. The resultant transient is shown in Fig. 10B. In this instance, combining acoustic damping with electrical damping resulted in a near aperiodic response which is considered by many as ideal for the reproduction of percussion sounds.

With regard to frequency response, acoustic damping generally should have an effect similar to that of electrical damping. Response may be affected, however, by the specific arrangement of acoustic damping. It is interesting, therefore, to compare response curves for the system which we have described under similar conditions of damping -- electrical vs. acoustic. The measuring set-up for this purpose is shown in Fig. 11. The response measurements were made on the axis at three feet from the cone, with a constant generator voltage and suitable series resistance to simulate the source impedance. The resulting curves are shown in Fig. 12. Only the low frequency is affected by damping, and, hence, the curves stop at 1000 cps. The solid curve is for the loudspeaker in the acoustically damped enclosure and high source impedance. The dash-curve is for the loudspeaker in the open back enclosure and a low source impedance. The dotted curve is for the loudspeaker in the enclosure with the solid back and also a low source impedance. It would appear that the damped-back curve might be chosen as the best compromise because it is the smoothest and the easiest to equalize. It must be reiterated that the system under study is not intended to qualify as a "high-fidelity" system, but rather to represent the conditions which might exist in a home radio or a public address system.

Of further interest is the performance of the acoustically damped loudspeaker as a function of the source impedance. The response curves for high and low impedance source are shown in Fig. 13, and they are quite alike. This may be accounted for by the fact that the voice-coil impedance remains fairly constant owing to the acoustic damping. This comparative immunity from source-impedance effects is not to be minimized, since it provides the audio designer with a degree of freedom in the choice of equipment.

VI. CONCLUSION

Transient response of loudspeakers and enclosures can be effectively controlled by acoustic damping. Furthermore, the response-frequency characteristic of the loudspeaker system need not be adversely affected, and it actually may be improved. Loudspeakers with acoustic damping may operate from high-impedance amplifiers without "hang-over". Performance characteristics become largely independent of the amplifier impedance. Acoustic damping may be designed in a straightforward manner by ascertaining the acoustical constants and using standard experimental techniques of equivalent circuit analysis. We conclude, therefore, that acoustic damping for loudspeakers merits far more serious consideration than it has had heretofore.

ACKNOWLEDGMENT

In the preparation of this paper, the author wishes to acknowledge with thanks the able assistance of Mr. Leo Rosenman of Shure Brothers, Inc., who has been of immeasurable help with the planning and execution of these experiments.

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(7) For example, in text of reference (1), p. 89.

(8) For example, in text of reference (1), p. 87.

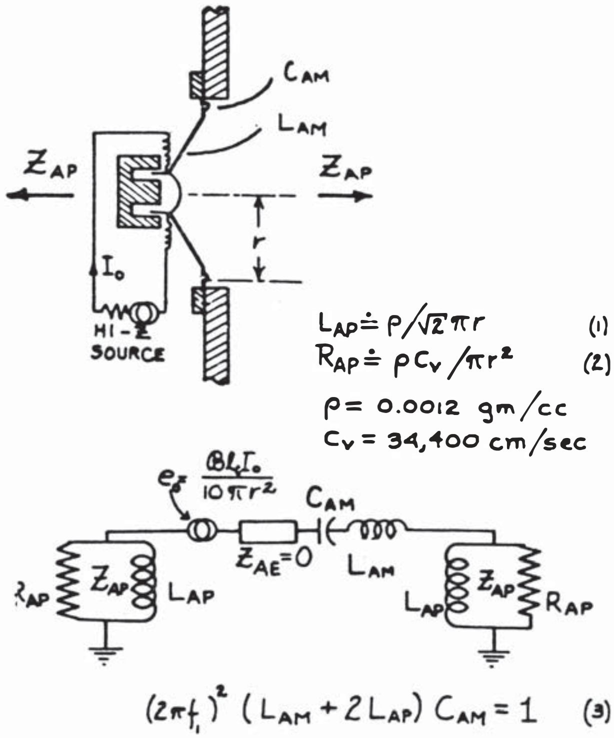


Fig. 1. Equivalent circuit of a loudspeaker mounted in an infinite baffle.

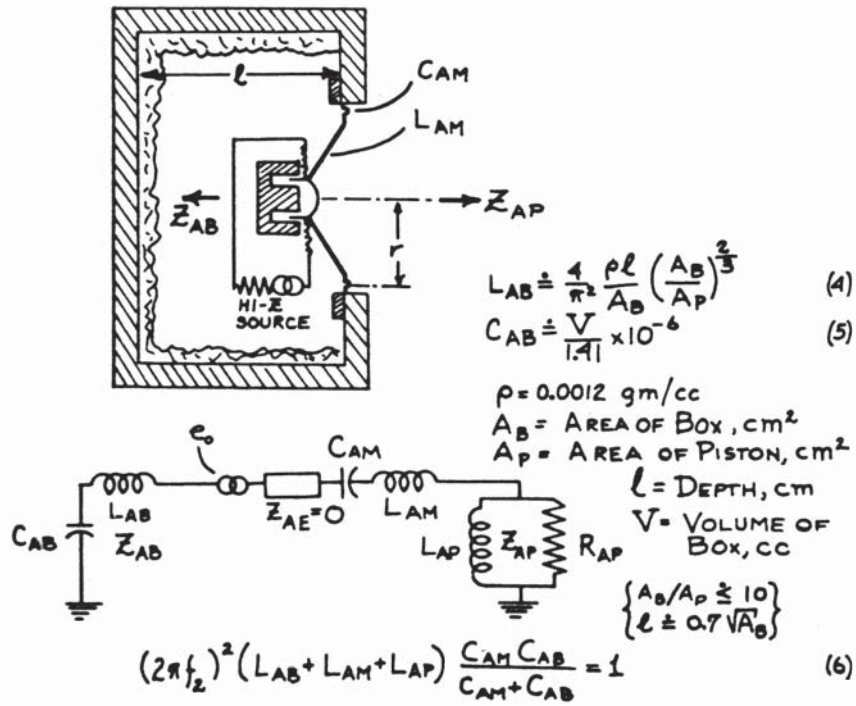


Fig. 2. Equivalent circuit of a loudspeaker mounted in a sealed enclosure.

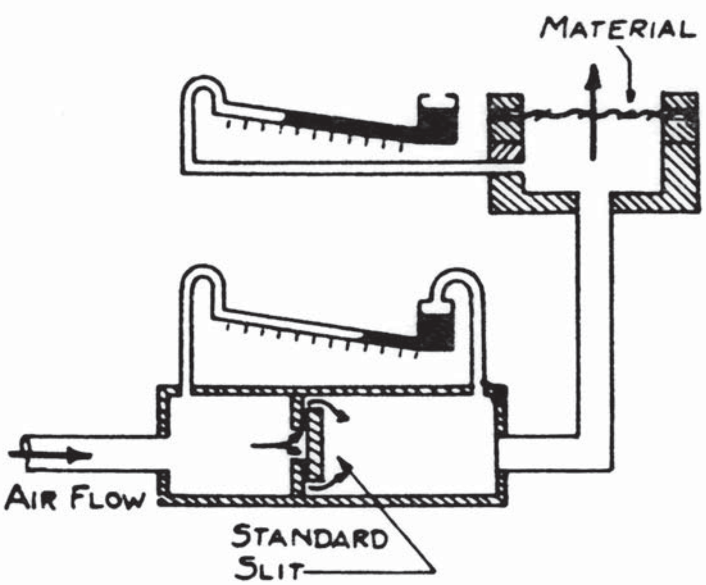


Fig. 3. Schematic arrangement of an acoustic resistance meter.

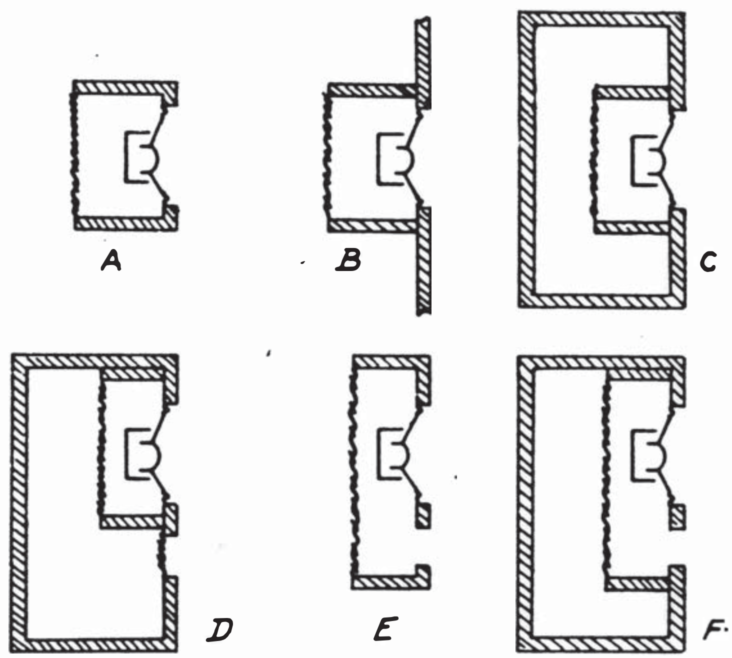


Fig. 4. Some of the means for applying acoustic damping to loudspeakers.

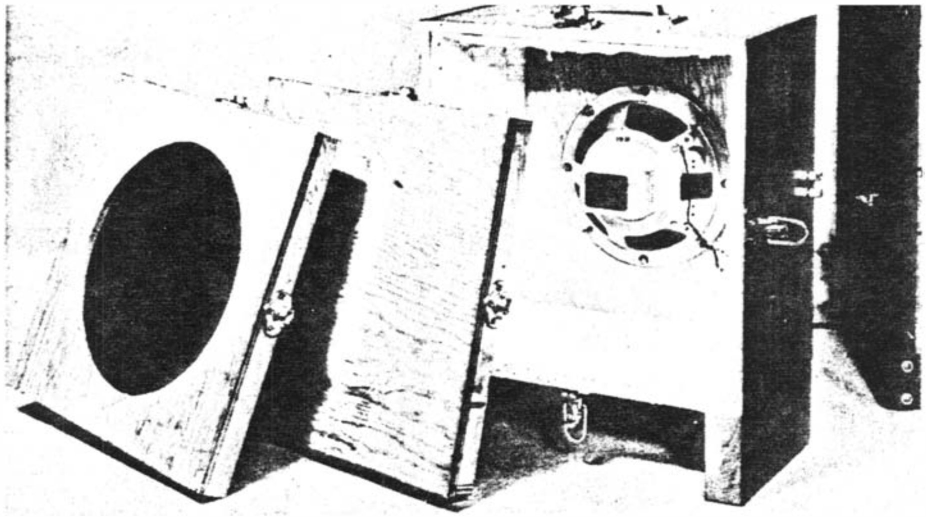


Fig. 5. Experimental loudspeaker system demonstrating acoustic damping.

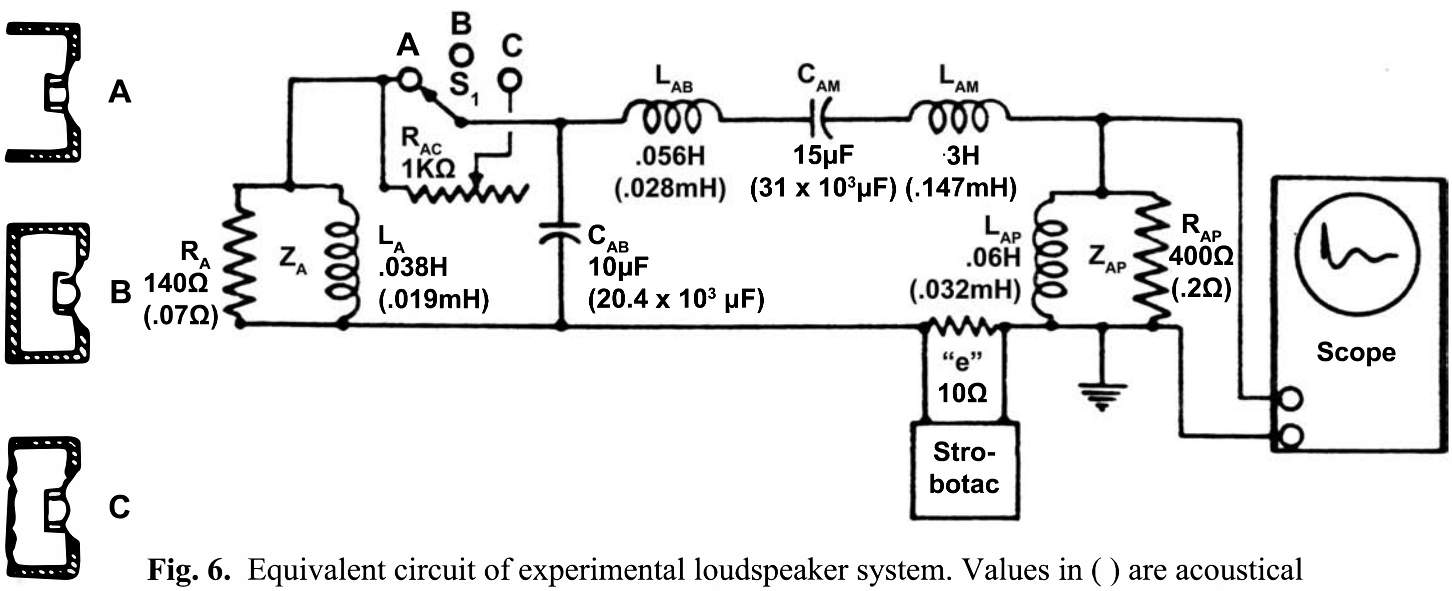


Fig. 6. Equivalent circuit of experimental loudspeaker system. Values in ( ) are acoustical values. Electrical values for resistance and inductance are acousticals times 2,000, capacitance values are acousticals times 1/2,000. The GenRad 1531 Strobotac is used as a pulse generator.

(a) (b) (c)

Fig. 7. Transients obtained with equivalent circuit of Fig. 6.
(a) simulating open box
(b) simulating closed box
(c) simulating damped box

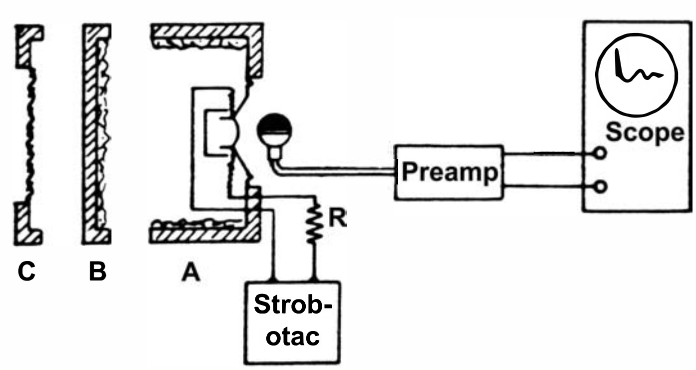
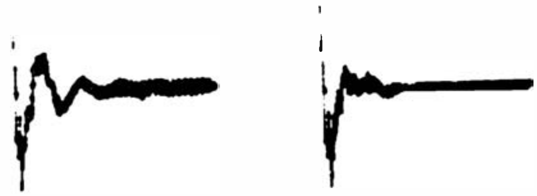


Fig. 8. Schematic arrangement for studying acoustical transients.

(a) (b) (c)

Fig. 9. Transients obtained by acoustic measurements:
(a) open box
(b) closed box
(c) damped box



(a) (b)

Fig. 10. Transients obtained by acoustic measurements:
(a) closed box w/electric damping
(b) acoustically damped box w/electric damping

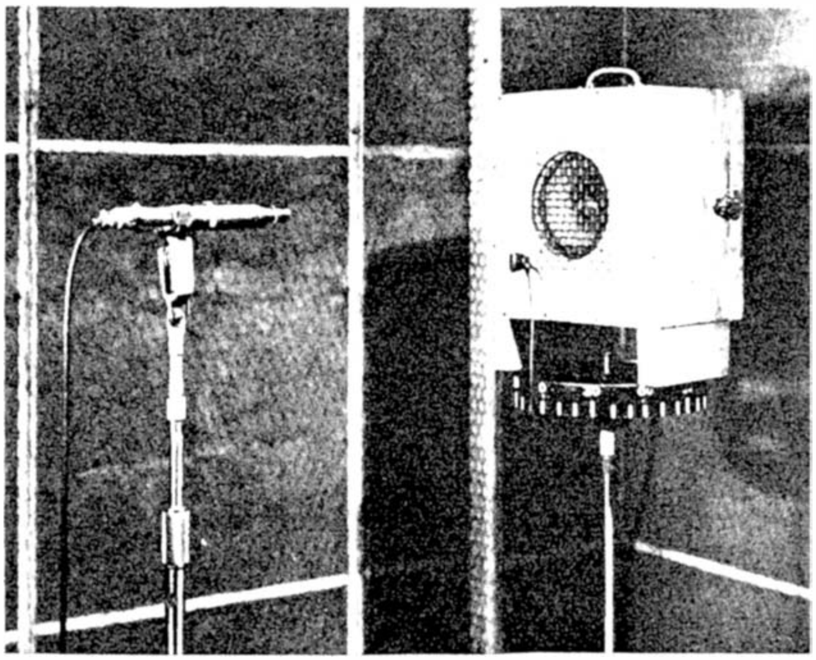


Fig. 11. Setup for frequency response measurements

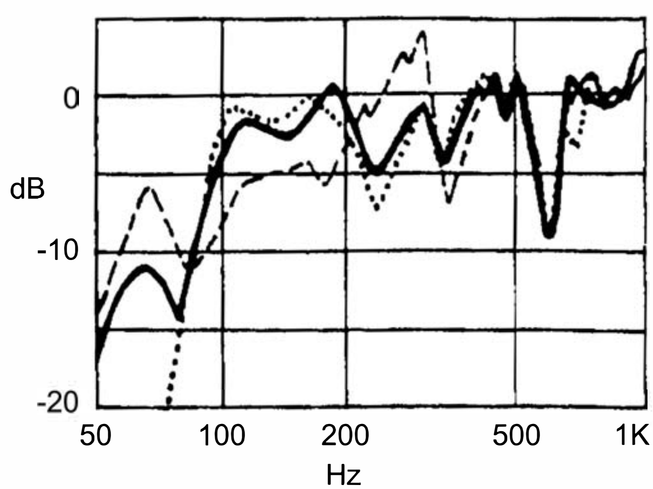


Fig. 12. Frequency responses:
(a) dashed line - open
(b) dotted line - closed
(c) solid line - damped

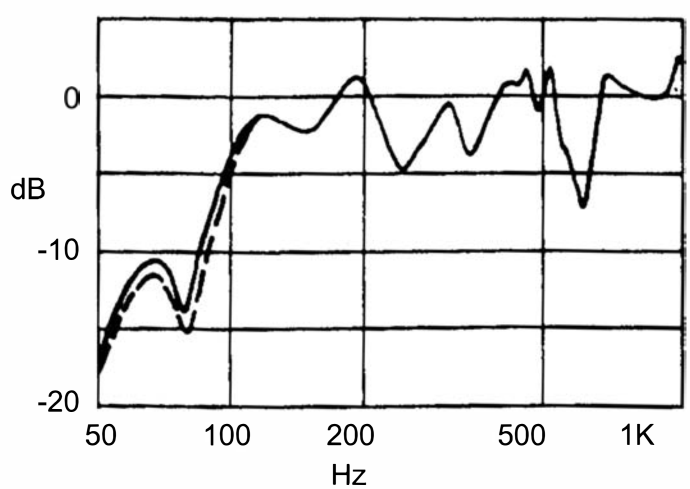


Fig. 13. Response of acoustically damped loud-speaker as a function of source impedance:
(a) solid line - high source Z
(b) dashed line - low source Z

Missing Link in Speaker Operation

Parts 1 & 2

by D. J. Tomcik — Chief Electronics Engineer, Electro-Voice, Inc.

PART 1

THE AMPLIFIER DAMPING FACTOR AND ITS APPLICATION TO SPEAKER PERFORMANCE

IN AUDIO REPRODUCTION, a subject of considerable importance to the high-fidelity enthusiast is amplifier damping factor and its effects on speaker operation. Misconceptions have arisen concerning this subject, and vague and incomplete answers have too often been given to the many questions involved.

Are the high damping factors found in present high-fidelity amplifiers by-products of high-feedback circuits and, as such, unimportant in the operation of the system? Or is the ultimate, as some loudly proclaim, to have the highest possible damping factor built into the amplifier? Why does a particular speaker sound better with amplifier A than with amplifier B, although both show identical frequency response and power capabilities under bench checks? Why does that $2.00 speaker with the 6-ounce magnet (inefficiency and distortion included) seem in some cases to have more bass than the high-fidelity unit with the 5-pound magnet? Why is it that one enthusiast found reproduction more pleasing when he used a little current feedback from the output circuit yet another didn't when using the same circuitry?

Some simple laboratory experiments and straightforward analysis can answer these questions and clear the air.

SPEAKER MECHANICS

The speaker can be considered, for the moment, a purely mechanical device. As such, the cone with its inherent mass and the cone suspension with its compliance or stiffness make up a resonant system. Some mechanical damping is present but such a slight amount that the system can be considered highly underdamped — if the cone is displaced and then released, it oscillates about its normal resting position for several cycles at its natural resonant frequency. This oscillation decreases in amplitude and finally reaches a state of rest due to the small amount of damping.

If this underdamped speaker is driven by a voltage source having a very high internal impedance so as to maintain the underdamped condition, the cone will vibrate at a greater amplitude at frequencies close to its natural resonance. (This action is similar to pushing a swing or pendulum "in time" with its natural period so as to obtain large amplitudes.) The frequency-response curve of the speaker under these conditions will show a peaked output near cone resonance, usually between 30 and 80Hz. Operation in this manner produces high transient distortion and is undesirable in high-fidelity systems. This can be shown by pressing and then releasing the cone. The oscillation which results is all distortion, since the cone does not follow the applied square waveform of depressing and releasing it.

THE SPEAKER AS A TRANSDUCER

To reduce the transient distortion as well as the peaked bass response, it is necessary only to damp the cone. If the speaker were purely a mechanical device this would be difficult. But since it is an electromechanical transducer, damping is obtained easily. In analyzing the electrical portion of the speaker we find a coil of wire wound around a form and attached to the cone. The coil is placed in a magnetic field and in this way constitutes a simple motor or generator.

If a voltage is applied to the coil, it moves in the magnetic field which in turn moves the cone. If the cone is mechanically moved, the motion of the coil in the magnetic field generates a voltage in the coil. From this it can be seen that cone damping can be obtained by using the magnetic braking action present when the coil terminals are externally closed through a resistance. The motion of the cone in trying to oscillate generates a voltage in the coil. This voltage produces a current flow through the coil and external resistance. The current flow tries to move the cone by motor action, but opposite in direction to the motion producing the current. Therefore the cone is damped in its free motion.

For a given speaker, the amount of damping can be varied by changing the value of the external resistance and consequently the value of the braking current. There is one value of damping at which the cone returns to rest in the quickest possible time without going past the rest position. This condition is called the critically damped state. Transient distortion is greatly reduced and the low-frequency response is more nearly uniform.

Excessive damping returns the cone slowly to its rest position. If the speaker is driven by a voltage source with very low internal resistance, the low-frequency response lacks intensity. (The action now is similar to pushing a pendulum while it is submerged in grease or heavy oil.)

Fig. 1 shows the extent to which a speaker's output is affected by various damping values. Actual speaker performance is shown for overdamped, critically damped and underdamped conditions. To obtain these curves, the amplifier driving the speaker had means for varying its internal resistance. The method for accomplishing this will be discussed later.

Speaker-Damping Variables

The easiest method for determining these factors as well as their effect on the damping action is to resort again to the magnetic brake. First, we know that the induced voltage in the coil is directly related to the amount of flux cut by the coil. Therefore a larger magnet or smaller gap volume will induce a higher voltage. As a result, a larger external resistance is needed to limit the current so that the desired braking action is retained.

Second, the amount of induced voltage is directly related to the length of conductor cutting the magnetic field. Since both the coil diameter and the number of turns are directly proportional to the length of wire, we may conclude that these factors also enter into the determination of the critical damping resistance (R_{CD}).

Third, the R_{CD} is made up of two components. The d.c. coil resistance as well as the external resistance limits the amount of braking current. So the R_{CD} is the sum of the coil resistance and the external, or amplifier, resistance. Conditions might exist, and they surely do, where the d.c. coil resistance of a given speaker itself is greater than the resistance necessary to critically damp the cone. Nothing can be done externally to remedy this situation short of using a negative resistance amplifier output impedance.

The fourth factor, which is a little more difficult to explain without mathematical illustration, is the effect of the cone mass and suspension stiffness on the value of the R_{CD}. It is logical that to stop a heavier cone from moving in a given time, a greater force in opposition to the motion is required. To increase the opposing force, it is necessary to have a larger braking current, obtainable by lowering the circuit resistance. Also, the stiffer suspension, when displaced, possesses a greater restoring force. The tendency for the cone to overshoot its rest position is increased. Therefore, the damping force necessary to overcome this restoring force must be increased proportionately and can be obtained by again decreasing circuit resistance. It must be remembered that the effective mass and stiffness of a speaker are dependent to some extent on the type of

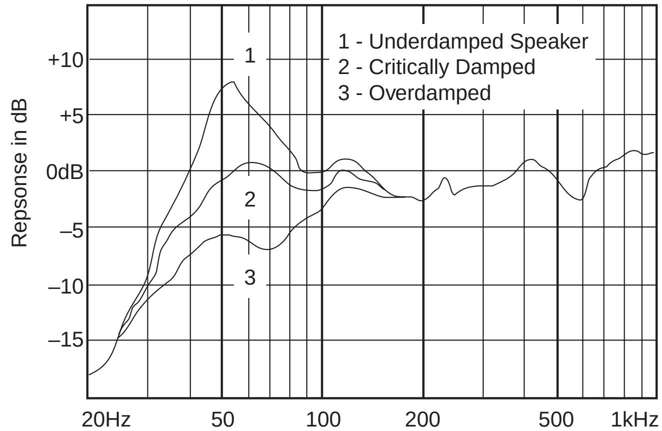


Fig. 1. The effect on frequency response of various values of damping

enclosure in which the speaker is housed.

To summarize, the factors that determine the R_{CD} can be mathematically expressed as follows:

$$R_{CD} = C\frac{(Bl)^2}{\sqrt{kM}}$$

where R_{CD} is the critical damping resistance; B, flux density; l, length of conductor in the magnetic field; k, effective stiffness; M, effective mass, and C is a constant.

This formula is not given to encourage experimental calculations—the test equipment necessary is far beyond the means of the average audio enthusiast. Rather, it is used to indicate the relationship of the various factors in determining the critical damping resistance.

Experimental Results

Figs. 2, 3 and 4 show actual response curves of three speakers for various values of damping resistance. Infinite baffles were used in obtaining all curves. The speaker whose curves are shown in Fig. 2 is an inexpensive unit with a 6.8-ounce magnet and 1-inch coil diameter. By referring to the equation we can expect the R_{CD} to be low because of the low values of B and l. The curves bear this out since both show the speaker in an underdamped condition. The d.c. coil resistance of this speaker is greater than the R_{CD} and, even though the amplifier resistance is 0.5 ohm for the lower curve, the speaker is still underdamped. Fig. 3 shows curves for a high-fidelity 12-inch speaker with a 3-pound magnet and a 2½-inch voice-coil diameter. As expected, the R_{CD} is much higher than in the first case. The overdamped, critically damped and underdamped curves show plainly that the smoothest response occurs when the speaker is critically damped. Fig. 4 indicates that this 15-inch speaker with a 5-pound magnet and large

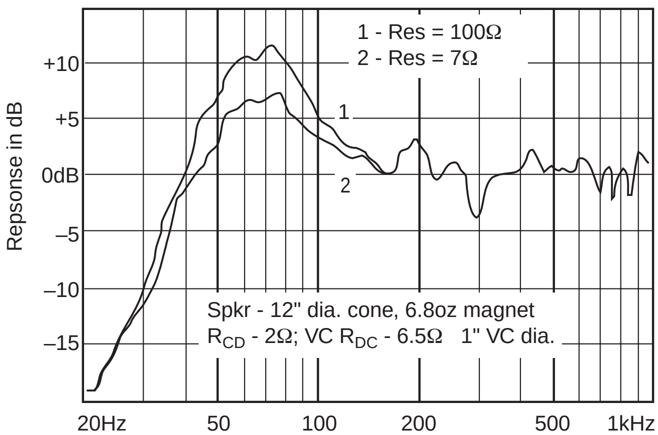


Fig. 2. Frequency response curves for an inexpensive speaker with different values of "damping resistance."

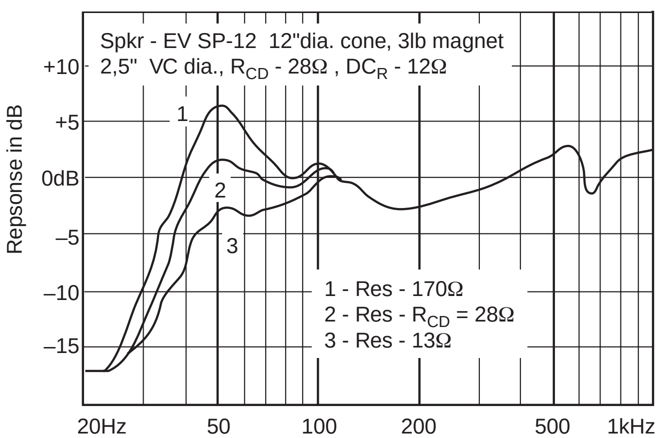


Fig. 3. Frequency response curves for a hi-fi speaker speaker with different values of "damping resistance."

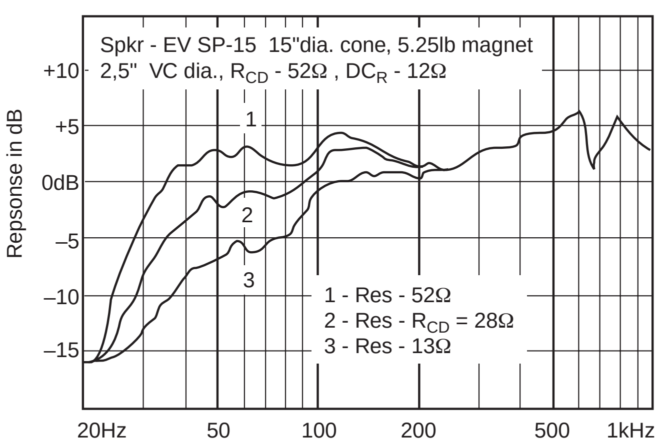


Fig. 4. Frequency response curves for a 15" speaker with different values of "damping resistance."

voice coil has still a higher R_{CD}. The overdamped curve, No. 3, shows what can be expected from the speaker when the amplifier damping factor is 10 or greater, commonly considered to be the criterion of a good high-fidelity amplifier. However, to obtain critical damping with this particular speaker, the amplifier damping factor had to be adjusted to a value of 0.4—an internal resistance of 40 ohms on the 16-ohm tap! The increase in efficiency by operating this speaker critically damped instead of overdamped is 9dB or 8 times the acoustic power output.

Conclusion

To return to our questions in the early part of this article, let us now see how many can be answered. High damping factors should not be considered by-products of inverse feedback, but should be controlled. They play a very important part in the reproducing chain. Neither should an amplifier be designed with very high damping factors only. A good high-fidelity amplifier demands that the damping factor be variable within wide limits. It is important not only to present the correct load impedance to the amplifier, but also to present the correct load impedance to the speaker. These two load values are seldom the same. The means of true amplifier-to-speaker matching is obtained with the aid of correct amplifier damping factor selection. The answer to the question of why amplifier A works better than amplifier B with a given speaker is that the damping factor of amplifier A more nearly critically damped the speaker than did amplifier B. During the bench test with resistive loads, the two performed in an identical manner. With the variable load of the speaker, the operation was entirely different. The inexpensive speaker seemed to have more bass than the high-fidelity unit because it was working in an underdamped condition, even with a high amplifier damping factor, whereas the hi-fi speaker was heavily overdamped. And finally, the man who improved his speaker performance with current feedback was merely altering his amplifier damping factor to suit his speaker combination. Your speaker, being entirely different, did not perform with that particular value of damping as it would with critical damping. It's all as simple as that.

From the foregoing we see that speakers vary greatly in their requirements of source impedances to critically damp the cone and achieve optimum speaker performance. It also has been conclusively shown with laboratory curves that best speaker performance occurs with critical damping. No one value of amplifier internal impedance can satisfactorily match all speakers and enclosures. The missing link has been found in critically damping the speaker.

Part 2

Obtaining Variable Damping Factors in Amplifiers; Determining Critical Damping Factors

Part 1 of this article (December, 1954) discussed the effects of various damping factors on the operation of cone type speakers, particularly in the region of cone resonance. It was determined that a given speaker performs best only when critically damped. The matched reproducing system therefore requires the speaker as well as the amplifier be terminated in their proper loads. The amplifier is matched over the greater portion of the frequency spectrum by proper design, so the speaker should be matched to its desired load by proper amplifier design.

As was shown, the proper speaker load for critical damping is the numerical difference between the critical damping resistance (R_{CD}) and the d.c. resistance of the voice coil. This value should be equal to the amplifier internal impedance. Of course, the speaker can be critically damped by using an amplifier of very low internal impedance and putting a fixed resistor in series with the speaker. However, this method results in a power loss in the resistor which may be much greater than that supplied to the speaker. The correct and efficient method of matching is by controlling the amplifier internal impedance, which does not absorb power. (The amplifier nominal impedance should not be confused with the amplifier internal impedance. The nominal impedance, 4, 8 or 16 ohms, is what the amplifier should work into whereas the amplifier internal impedance refers to regulation, as explained later in this article. The two values are seldom the same.)

Damping Factor

To simplify matters and eliminate the variable of nominal impedance, the term amplifier damping factor is often used. The damping factor is equal to the nominal impedance divided by the internal resistance of the amplifier. For example, an amplifier whose internal resistance on the 16-ohm tap is 8 ohms has a damping factor of 2.

For a given speaker there is one value of internal resistance and consequently one value of damping factor which results in critical damping. This value can be called the critical damping factor (CDF).

To visualize the damping factor concept better, consider the amplifier output terminals a voltage source with zero impedance in series with a resistor equal in value to the internal impedance. Fig. 1 shows this arrangement with the proper amplifier load. The amplifier may be push-pull, cathode follower, or any other type, since the equivalent output circuit for all can be considered to be the same as that in Fig. 1. To measure the damping factor of any amplifier it is necessary only to measure the output voltage under no-load and rated-load conditions. The formula for damping factor then becomes:

Fig. 1. Equivalent output circuit.

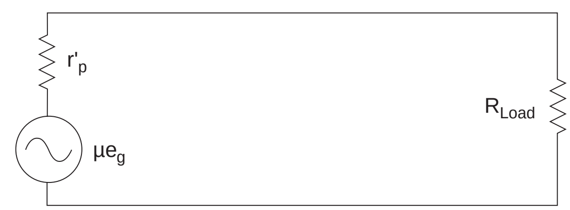


Fig. 2. Schematic of the equivalent plate circuit of the output stage.

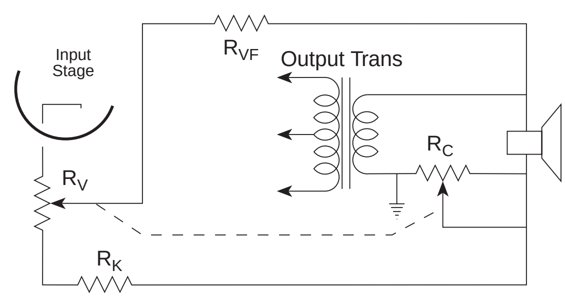


Fig. 3. Circuit using variable feedback.

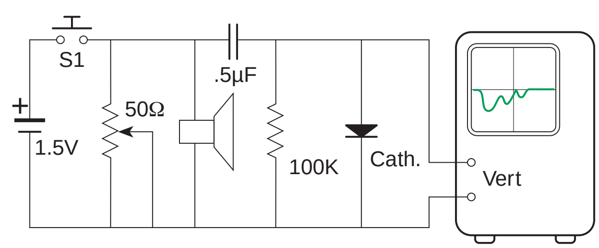


Fig. 4. Circuit for determining CDF.

$$DF = \frac{E_{RL}}{E_{NL} - E_{RL}}$$

where E_{RL} = rated-load voltage and E_{NL} = no-load voltage.

From this formula we see that the damping factor is also a measure of the output regulation—how far the output varies from a constant-voltage source with changes in load. Amplifiers with high damping factors

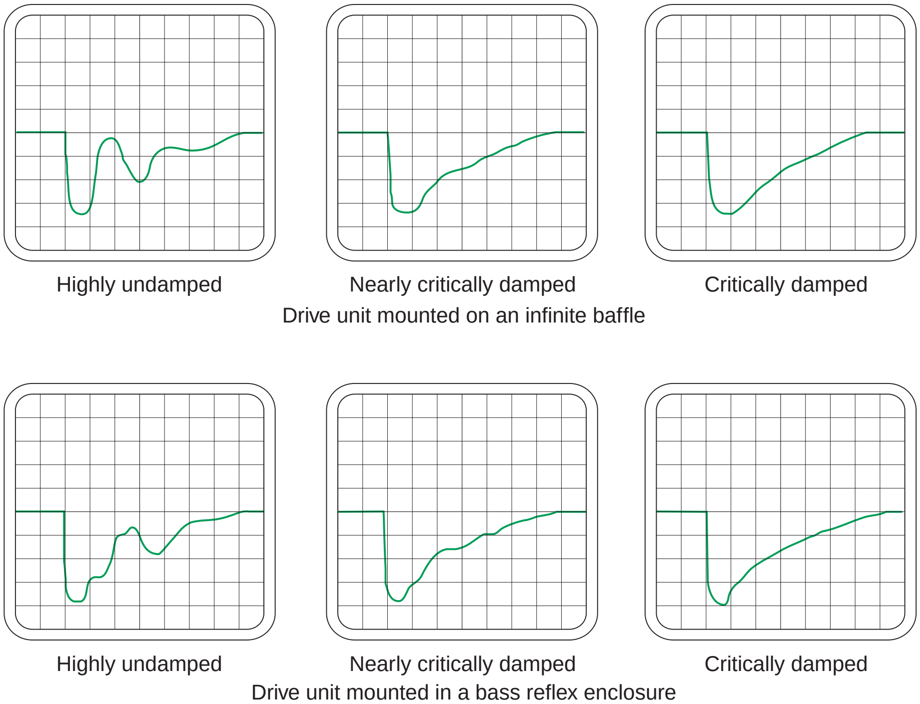


Fig. 5. Three time responses for differing values of damping. The upper traces were taken with the drive unit mounted on an "infinite baffle" while the lower ones were taken with the unit mounted in a bass reflex enclosure.

act more like constant-voltage sources than those with lower damping factors. Conversely, amplifiers with low damping factors act more like constant-current sources than those with higher damping factors. Hence a means is provided to vary the damping factor simply by controlling the regulation. This is easily done by using feedback in the circuit to obtain more or less constant-voltage or constant-current amplifiers. But, first, what damping factor requirements are necessary in a really good high-fidelity amplifier?

We have seen in Part I that all speakers classed as high-fidelity units require positive values of amplifier resistance for critical damping. Only inefficient speakers in the very low price range require negative resistances. It is possible to obtain negative-resistance characteristics from an amplifier by using positive feedback. But since the speaker that requires this type of damping is mediocre at best, and since positive feedback may result instability and increased distortion, negative-resistance amplifiers are unnecessary in the present state of the reproducing art.

The highest practical damping factor is approximately 10 to 15. Since the d.c. coil resistance of a speaker generally makes up more than 75% of the total nominal impedance, amplifier damping factors greater than 10 or so have no appreciable effect on speaker damping. For example, a 16-ohm speaker with a 12-ohm d.c. coil will have a total series resistance of 13.6 ohms with a damping factor of 10, and a total series resistance of 12.4 ohms with a damping factor of 40. The difference in frequency response is not noticeable to the ear and hardly measurable.

The lowest damping factors found in today's high-fidelity speakers range from around 0.3 or 0.4. A good low limit for amplifier design would be 0.1 or 160 ohms on the 16-ohm tap.

We have now established the limits to be between 0.1 and 15 to cover amply the range in modern-day speakers. The damping control should be calibrated directly in damping factor or internal resistance to simplify adjustment to a given speaker.

Other Considerations...

A good high-fidelity amplifier should also have a constant sensitivity with rated load applied, as the damping factor is varied. This results in a constant negative feedback which maintains the distortion and hum figures at constant low levels. In addition, the damping-factor control system should not be frequency-discriminating. Frequency discrimination will affect sensitivity more at some frequencies than at others and produce an undesirable tone-control action in the system.

An effect which might be noticed in a non-frequency-discriminating system is that of high frequency accentuation at low damping-factor values where a single wide-range speaker is used. This high-frequency boost is caused by the speaker inductive reactance becoming appreciable and affecting the gain of the amplifier at high frequencies. The effect is not present in multiple-speaker systems since the reproducing components are designed to present the nominal impedance over their working range of frequencies and are then cut off by the crossover network above their range. However, in a single-speaker system, the several decibels of treble boost are not serious and probably complement the high-end speaker roll-off due to the single-cone operation. In any event, if desired, the treble control can nullify this effect.

Amplifier Design

With the requirements mentioned, we can proceed to design the output circuit of the amplifier, remembering the criterion that negative voltage feedback lowers the internal resistance while negative current feedback raises the resistance. We will probably want to settle on some minimum value of negative feedback to take care of the frequency response, distortion and hum. This usually falls between 15 and 20dB of loop feedback. It may turn out that we will need more than this minimum value, depending on the circuit and particular constants used.

The circuit is first considered without loop feedback so as to determine the starting-point damping factor. The equivalent circuit shown in Fig. 2 applies to all types of circuits as long as rp is considered the effective plate resistance and R_L the load impedance referred to the primary of the output transformer. For example, the damping factor for push-pull 6V6's pentode-connected, with rp = 64,000 ohms and R_L = 8,000 ohms, is:

$$DF = \frac{8,000}{64,000} = .125$$

without loop feedback. To obtain the high damping factor of 10, negative voltage feedback is used around the loop until that value is obtained, say 18dB. Then if 9dB of negative current feedback is added while 9dB of negative voltage feedback is removed, the total negative feedback remains at 18dB; but the damping factor is now 0.125 as with no feedback. However, the sensitivity, frequency response, distortion and hum remain constant. This amplifier covers approximately the desired range of damping factors while the set requirements are maintained. The design procedure is applicable to all types of amplifiers, the only difference being the initial value of damping factor with no loop feedback. The value of r_p will vary greatly with the type of circuit. A cathode-follower output stage results in a very low value of r_p, approximately equal to the reciprocal of the transconductance. Generally, pentodes in push-pull have high values. Triodes fall in between pentodes and cathode-followers. The ultra-linear circuit is a combination midway between pentode and triode operation. The Electro-Voice Wiggin's Circlotron circuit using two 6V6s in a bridge arrangement results in an r_p equal to 2kΩ and an RL equal also to 2kΩ. Under these conditions, no loop feedback or equal voltage and current feedback results in a damping factor of 1. Over-all negative voltage feedback increases the damping factor to values greater than 1, and over-all negative current feedback to values less than 1.

Fig. 3 shows an arrangement in which the current and voltage feedback can be varied in any amount while the total feedback remains a constant. Resistor R_c permits maximum desired current feedback when it is fully in the circuit. The value of R_{vf} is such that the total required voltage feedback is obtained when the slider of R_v is at the cathode position. The ratio of R_v to R_k is chosen so that the voltage feedback is equal in value to the maximum current feedback when the movable arm of R_v is at the R_k end. The two potentiometers are ganged in such a way that an increase in voltage feedback causes a decrease in current feedback.

DETERMINATION OF CDF

With the Fig. 3 circuit arrangement it is possible to obtain various damping factors by the turn of a knob. It is now necessary only to determine the CDF of the speaker to be used and then adjust the amplifier damping factor to that value.

The Electro-Voice line of high-fidelity speakers includes the CDF value in their specifications. For those who would like to determine the CDF of their present speaker system, the equipment needed includes an oscilloscope, a calibrated variable resistance of about 50 ohms maximum, a flashlight battery, a momentary pushbutton switch, a 0.5-µf capacitor, a type 1N34 germanium diode and a 100kΩ resistor. The components are arranged as in Fig. 4. Carefully observe the polarity of the flashlight cell and the 1N34. Mount the speaker under test in its permanent enclosure since the baffle has some influence on the CDF value. The amount of coupling between the speaker cone and enclosure is not very great. The cabinet resonances will not be appreciably affected by changing damping on the speaker, but the enclosure does contribute to the effective mass and stiffness of the speaker, which directly affects the value of CDF.

The circuit of Fig. 4 operates as follows. When the switch is closed, the speaker cone is displaced due to the current flowing through the voice coil. The switch is then opened and the cone returns to the rest position. The voltage it generates in so doing is observed on the oscilloscope. The external resistance shunting the speaker is adjusted so that the scope trace indicates critical damping. This value of resistance is then equal to the necessary amplifier resistance to critically damp the cone. Thus the CDF is determined. The capacitor and 100kΩ resistor act as a filter to stabilize the scope trace when the switch is repeatedly opened and closed. If they weren't included, the trace would bob up and down on the screen, making observation almost impossible.

This effect is due to the slow charge and discharge of the input capacitor of the oscilloscope. The 1N34 diode shorts out the positive voltage surge when the switch is closed so that only the motion of the cone on opening the switch is analyzed on the scope.

In performing the test, start with the variable resistance high so the speaker is underdamped. Decrease resistance slowly until critical damping is obtained. In so doing, the point where the second and succeeding cycles just barely disappear is more easily seen. Approaching critical damping from the overdamped state is more difficult to observe. The switch should be continuously operated several times a second and the horizontal sweep adjusted to a low sweep rate.

The upper curves in Fig. 5 show the traces obtained for a speaker in an infinite baffle for the underdamped and critically damped states. Notice the absence of any second cycle when critically damped. In the lower curves of Fig. 5, the speaker was mounted in a very large bass-reflex box. Notice how the enclosure resonance remains even after the speaker is critically damped. In small bass-reflex enclosures, the resonant frequency of the port is close to the cone resonance and does not show distinctly on the curve. In this case, the patterns appear as in Fig. 5a. The point of critical damping is reached when the second full cycle is just barely eliminated in all cases. The wave form for a slightly overdamped condition looks the same as that for the critical damping, the only difference being an amplitude decrease in the overdamped wave.

Conclusion

The CDF of any speaker can now be obtained and the amplifier matched to give optimum performance. Errors of ± 50 % in CDF do not appreciably change the performance of the speaker, so it is unnecessary to determine the needed amplifier resistance down to the last ohm. In fact, the point at which critical damping occurs will be rather broad when the above procedure is followed.

However, great mismatch can result in a very appreciable loss of bass power—as much as eight or nine times. It is therefore worth the time and effort to determine the CDF and match the speaker-amplifier combination. You will have the satisfaction of knowing that the components are performing at their best and in this way providing more listening pleasure.

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Damping for Loudspeakers: Acoustical and Electromagnetic

By F. Langford-Smith and A. R. Chesterman.

The most prominent resonance displayed by a loudspeaker is the bass resonance which, if lightly damped, is most objectionable, giving rise to "one note bass", "hang-over", and poor transient response. For any set of conditions and for any one listener, there is an optimum amount of damping, to give the most pleasing results. It is important to remember that the amount of damping has only a small effect outside the region of the bass resonant frequency. Taking a typical case with a bass resonant frequency as 85 c/s, the effects of damping would be small above 150 c/s.

Introduction to damping

Loudspeakers have some damping inherent in the speaker, but all additional damping to meet the requirements of good musical reproduction must come from external electro-magnetic or acoustical damping, or both. Most modern amplifiers and radio receivers use negative voltage feedback to give a low output resistance, that is, a high "damping factor". Damping factor is defined as R_L/R_o, where R_L is the nominal load impedance and R_o is the amplifier output resistance. High fidelity amplifiers are sometimes advertised as having very high damping factors. E.g. 45, or in another case infinity. This term "damping factor" is quite misleading, since the damping is in no way proportional to the damping factor. The writers prefer to express this in the alternative inverse form where the output resistance is given as a fraction or percentage of the load resistance (Ref. 1). Thus an output resistance of zero (corresponding to a "damping factor" of infinity) gives a more accurate impression, particularly the non-technical person. However, the use of the term "Damping Factor" is so strongly entrenched that it cannot be displaced, and it will therefore be used in this article.

The effects of damping are shown by the equivalent circuit of Fig. 1 (Ref. 2). This may be applied to an infinite flat baffle merely by short-circuiting C_c. It will be seen that this is a series resonant circuit, with an applied voltage E_s across R, L and C in series. The Q of the circuit is given by

$$Q = \frac{2\pi f\ L_u}{R_s + R_u} \qquad (1)$$

and the acoustical output of the loudspeaker is proportional to E_{LU}.

$$\text{Now } R_S \propto \frac{B^2}{R_o + R_{vc}} \text{ at low frequency} \qquad (2)$$

where B = flux density in gap
R_o = output resistance of amplifier referred to the voice coil circuit
and R_{vc} = resistance of voice coil.

Two important facts are shown by eqn. (2). The first is that, so long as R_o is positive, the damping is limited by R_{vc}, and changing R_o from 10 to one-tenth of R_{vc} to zero (i.e. changing damping factor from 10 to infinity) only effects the damping resistance by 10%. The damping can only be truly infinite if R_o is made negative—ways of accomplishing this result will be described later in this article.

The second important fact shown by eqn. (2) is that R_S is directly proportional to the square of the flux density and inversely proportional to $R_o + R_{vc}$. Loudspeakers with low flux density may have insufficient damping even when R_o is made zero, whereas those with high flux density may be too heavily damped when R_o is zero. It is thus quite obvious that it is impracticable to select any value of "damping factor" which will give optimum results with any loudspeaker.

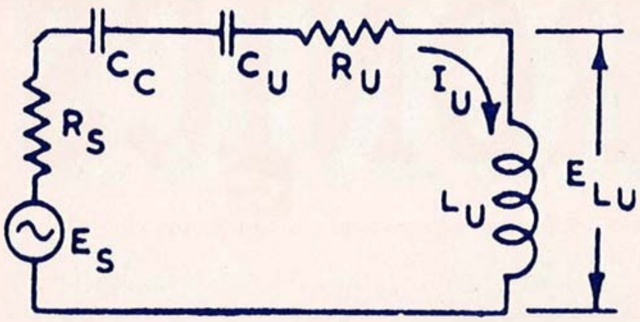


Fig. 1. Equivalent electrical circuit of loudspeaker in enclosed cabinet at low frequencies. RDH Fig. 20.11.

Effect on frequency response

Varying the damping on a loudspeaker has a second, and very important, effect which is not always fully realized. Its effect on frequency response is shown in Fig. 2 for a particular case with a totally enclosed cabinet. A similar effect occurs with a flat baffle, but the curves show that values of Q below about 1 result in bass attenuation. There are two common practices to meet this difficulty. The first is to maintain the loudspeaker Q about 1, or slightly over, to avoid the attenuation. The second is to accept the bass attenuation as a price to pay for high damping, and to make it good by adding bass boosting — about 7 db at the bass resonant frequency. There is absolutely no point in reducing Q below 0.5, the value to give critical damping.

Fig. 2. Theoretical response of enclosed cabinet loudspeaker for various values of damping resistance R_s. Resonance frequency 45 c/s. RDH Fig. 20.12.

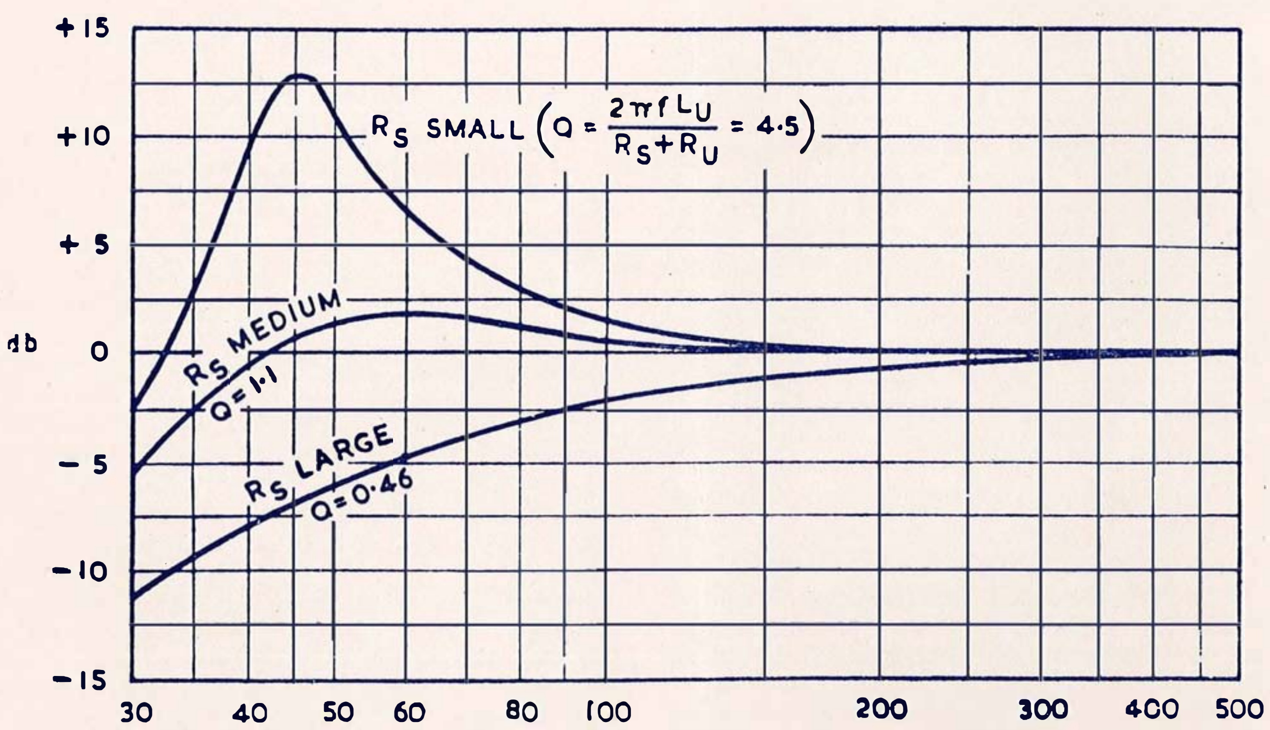
Damping controls

This introduces the desirability, in high fidelity amplifiers, of adding a control to alter the "damping factor". Two methods of achieving this result are shown in Figs. 3 and 4. Both use a combination of negative voltage feedback and positive current feedback. Although these add further control to the already formidable array on many amplifiers, it does serve a useful purpose. Normally it would be pre-set to suit a particular loudspeaker and listener, and not used as a regular control.

It has been stated above that loudspeakers with low flux density may have insufficient damping when R_o is made zero. What can be done about it? Short of changing the loudspeaker for one with higher flux density, there are two possible courses — to provide a negative output resistance, or to add acoustical damping. Both of these methods are described in this article.

Negative output resistance

A negative output resistance is produced by sufficient positive current feedback. This must be accompanied by an increased negative voltage feedback to reduce the harmonic distortion to a desirable level. Only one such commercial amplifier is



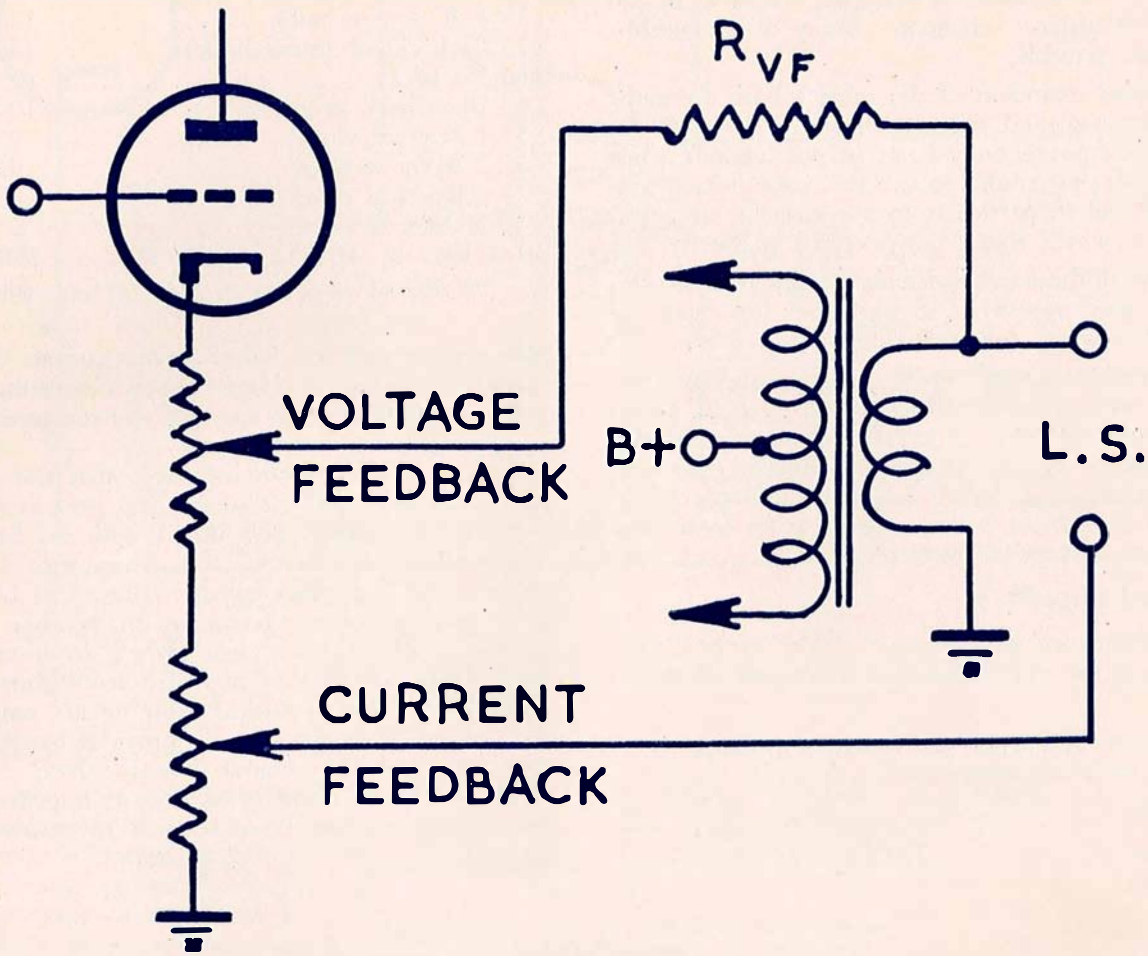


Fig. 3. Damping factor control used in Electro-Voice Amplifiers A-20C and A-30. .Range 0.1 to 15.

known at the present time—the Bogen Model D030A. The circuit is shown in Fig. 5 (Ref. 3). It uses 25 db negative voltage feedback, with current feedback controllable from negative to positive. When the switch is closed there is no positive feedback and the amplifier operates normally. When the switch is opened, with the damping control at the extreme negative current feedback position, the damping factor is positive and low (+2). As the damping control is moved towards the positive end the damping factor passes through the normal value (i.e. with switch closed) and then increases to infinity and beyond to high negative values, and finally low negative values (−2). The 4 μF condenser is used to limit the current feedback to low frequencies.

With such a control the amplifier will oscillate if the control is turned too far in the negative direction. Care is necessary to avoid possible damage to the loudspeaker. It is doubtful whether such an "unlimited" control will be widely used, but it would be of great interest in a laboratory or when carefully handled by an expert.

Acoustical damping

Acoustical damping has received scant attention in the literature, but it is a very helpful tool, and, as will be shown below, it can be used to reduce the peak of loudspeaker impedance at the bass resonant frequency—being the only known method of producing this result without horn loading or the vented baffle. The latter does not come within the scope of the present article, but it may be covered at some future date.

The present subject is limited to open back cabinets, the back of which may be closed by a sheet of expanded metal on which is glued a piece of suitable cloth (Fig. 6A). However, the same principles may be applied to various forms of enclosures. Fig. 6B shows the small enclosure attached to a large baffle to increase its low frequency response. In C, a totally enclosed cabinet is damped by the small enclosure shown. D is similar to C, except that a damped port is provided for a vented baffle. In E, a port has been added to the enclosure A, while F shows an arrangement similar to D, in which both the speaker and the port derive a measure of damping from a single acoustical resistance element. Many other combinations are possible.

A detailed treatment of the subject, both theoretical and experimental, has been made by Bauer (Ref. 4), and tests were carried out in our laboratory to see whether we could obtain the same results on damping, and in particular to measure the electrical impedance which Bauer only derived indirectly.

In general, the acoustical resistance element should have an area somewhat greater than the area of the cone, to avoid constriction of the air flow.

Measurements were made on an enclosure as Fig. 6A, with a medium-priced 9 in. × 6 in. elliptical speaker (M.S.P.) in an enclosure with internal dimensions 14 × 14 × 9 inches deep. The enclosure has interchangeable backs, one solid, and the other provided with an 11 in. diameter hole for mounting the acoustical damping resistance.

voice coil impedance

The impedance was measured under various conditions, and the most significant results are tabulated below.

f_1 = frequency at which impedance is a maximum,

ratio = ratio between impedance at f_1 and that at 400 c/s (= 3.5 ohms).

Condition	f_1	ratio
(1) 3 ft. square baffle	120	4.1
(2) Back closed (solid back)	143	4.7
(3) No back	105	3.5
(4) Open back in position, no cloth	100	3.1
(5) 1 layer of cloth	95	2.5
(6) 2 layers of cloth	95	2.1
(7) 3 layers of cloth	95	1.8
(8) 5 layers of cloth	95	1.5
(9) 1 layer of $\frac{1}{16}$ in. felt	150*	1.4

*a second peak occurred at 95 c/s, with ratio also 1.4.

The cloth used was loosely-woven cotton, 12 thou. thick, about the thickness of thin sheeting. The thickness of both cloth and felt was measured when compressed.

It will be seen from the table that the impedance ratio from 400 c/s to the bass peak is reduced from 4.7 with solid back, or 4.1 with flat baffle, to 1.4 in the most extreme case. Even with 3 layers of cloth the impedance ratio is reduced to 1.8. This is a very important feature with pentode power valves, which are very sensitive to an increase in load resistance and give high distortion under these conditions. Thus acoustical damping not only gives the desired damping, but also provides better working conditions for pentode power valves.

Fig. 7 shows curves of voice coil impedance for the various conditions set out in the table. The impedance at 400 c/s is 3.5 ohms.

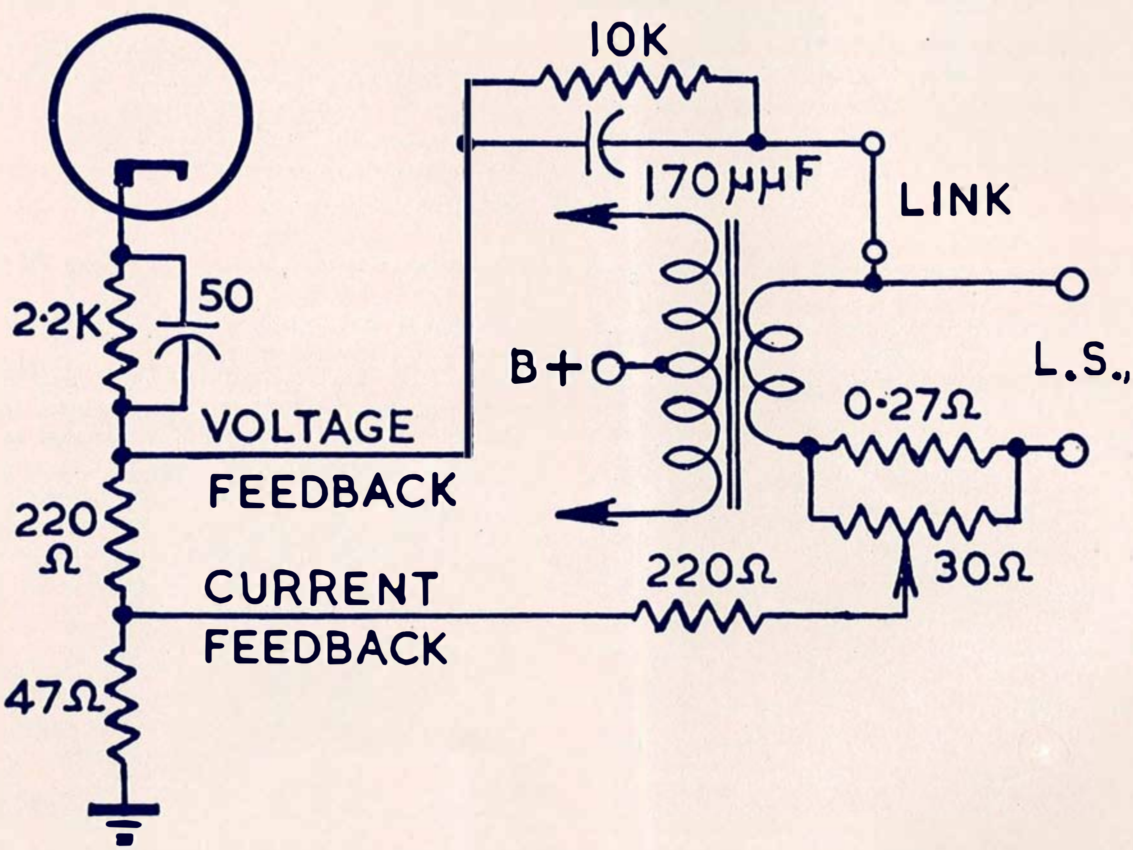


Fig. 4. Damping factor control used in Pye Model PF91 and Pamphonic Model 1002. Range up to infinity.

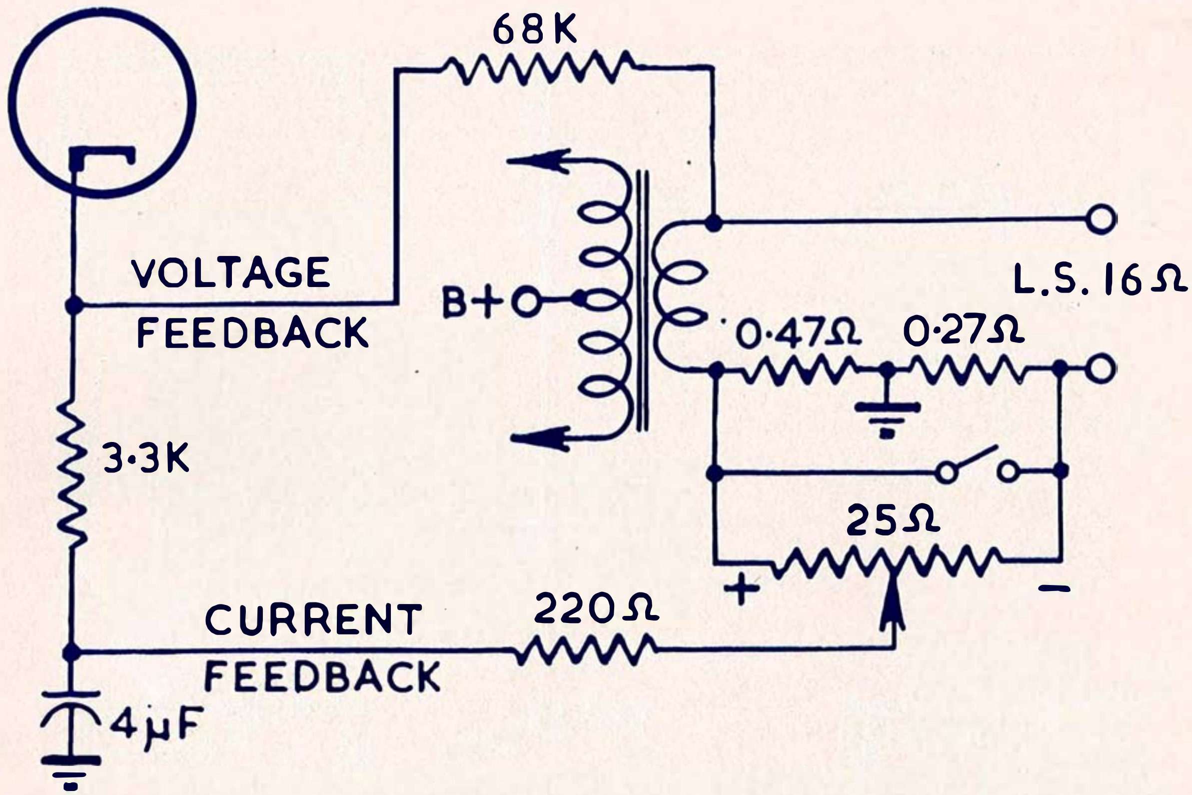


Fig. 5. Damping factor control used in Bogen D030A Amplifier. Range from low positive through infinity to low negative.

Frequency response

The low frequency response curves for three conditions are shown in Fig. 8. On a listening test curve B seems to be the best compromise. Curve A was preferred by some untrained listeners, apparently owing to the heavier apparent bass, particularly when listening to a certain male singer's voice. Curve C suffered from some attenuation in the bass, but was preferred to Curve A by two trained listeners.

Transient response

The transient response was tested using a Cintel pulse generator operating with a pulse 0.3 μsec. into a large amplifier driving the loudspeaker. A microphone was placed 1 foot in front of the loudspeaker and its output applied to an oscilloscope. Fig. 9 shows tracings made on the screen under various conditions. The increased damping is indicated by the decreasing height of the upwards peak immediately following the pulse. These results substantially agree with those published by Bauer.

General remarks

There is no doubt that acoustical damping is an effective and cheap way of getting damping, or increased damping. It has the valuable additional feature of reducing the loudspeaker impedance peak at low frequencies. It is therefore particularly suitable for use in expensive radio receivers and radio gramophones using a pentode output stage. This may be used with an unbypassed cathode resistor to give negative current feedback with reduction in distortion, and the whole of the damping will be provided acoustically. This may necessitate somewhat heavier damping cloth than that used in the tests above, which were all carried out with a low output resistance.

In most practical cases it will be found most convenient to adjust the thickness of the cloth to give the desired low frequency response, leaving the damping to look after itself.

Normally, a material will be found which will give the desired damping in one thickness. In our tests multiple thickness of thin cloth were only used as a convenient way of increasing the damping by known amounts.

Measuring voice coil impedance

When carrying out tests with acoustical damping, it is also advisable to measure the impedance of the voice coil at the low frequency peak to provide a measure of the damping. A very simple approximate test is to use a B.F.O. or other form of low frequency oscillator, and to connect it to the voice coil through a series resistance of at least 40 times the nominal voice coil impedance, and to measure the voltage across the voice coil by a rectifier type voltmeter.

All that is necessary for the purpose of this article is to measure the maximum value of the impedance below 400 c/s.

The series resistance provides nearly constant current, and the voltmeter reading is approximately proportional to the voice coil impedance. It may be calibrated by replacing the voice coil by a known resistance. Alternatively the impedance of the voice coil at 400 c/s may be taken as unity, and the voltmeter will then read the impedance ratio from the low frequency peak to 400 c/s.

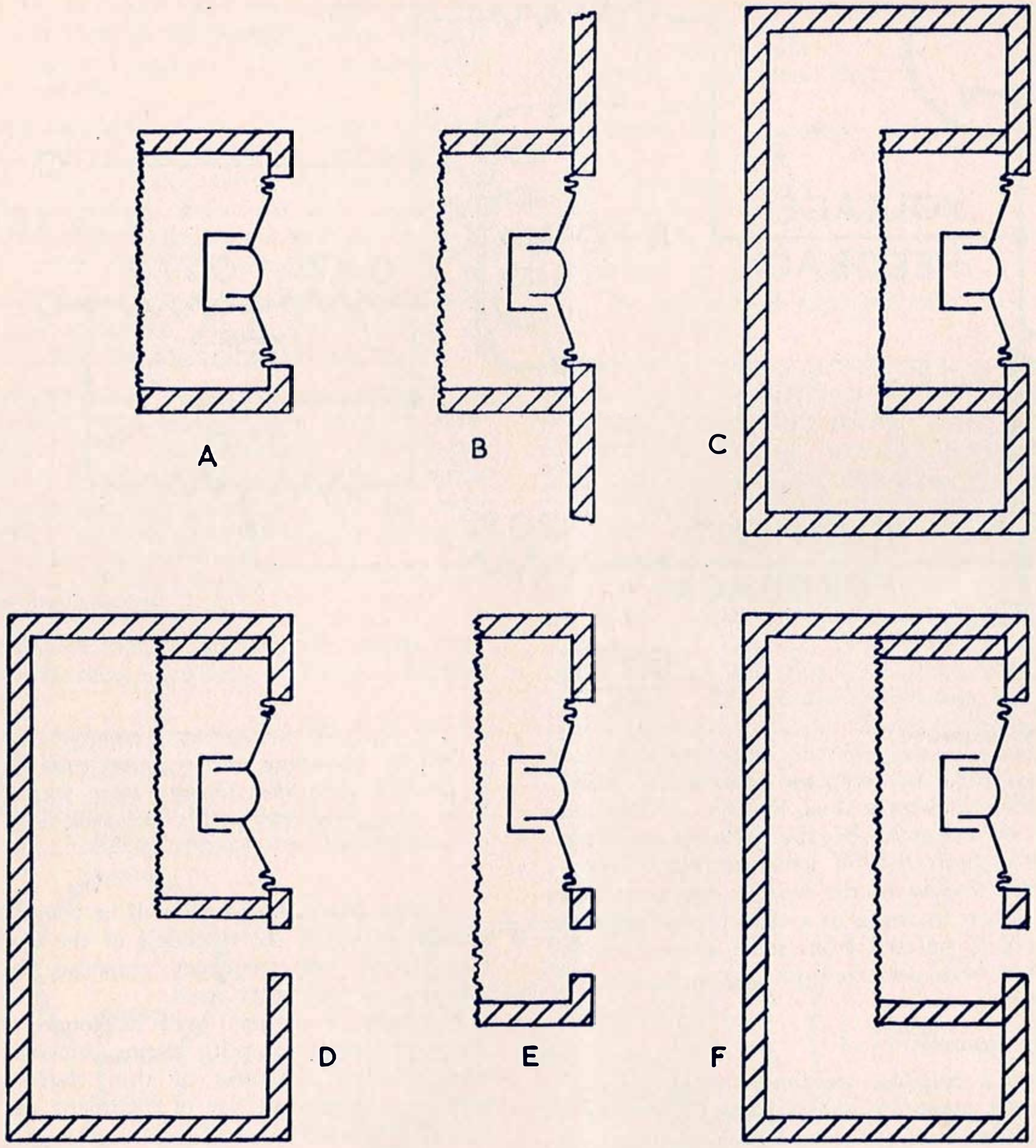


Fig. 6. Methods of applying acoustical damping to loudspeakers. After Bauer, Ref. 4.

L_u represents cone mass + effect of radiation reactance.

R_u represents radiation resistance (which varies with frequency).
Note that R_u is small compared with other impedances in the circuit.

C_c represents acoustical capacitance of cabinet volume.

C_u represents equivalent capacitance of cone suspension.

R_s represents effect of electrical circuit of loudspeaker and driving amplifier reflected into acoustical circuit. The mechanical resistance of the cone suspension may be taken as being included with R_s.

E_s = constant voltage generator.

I_u = alternating air current produced by cone, which is proportional to cone velocity.

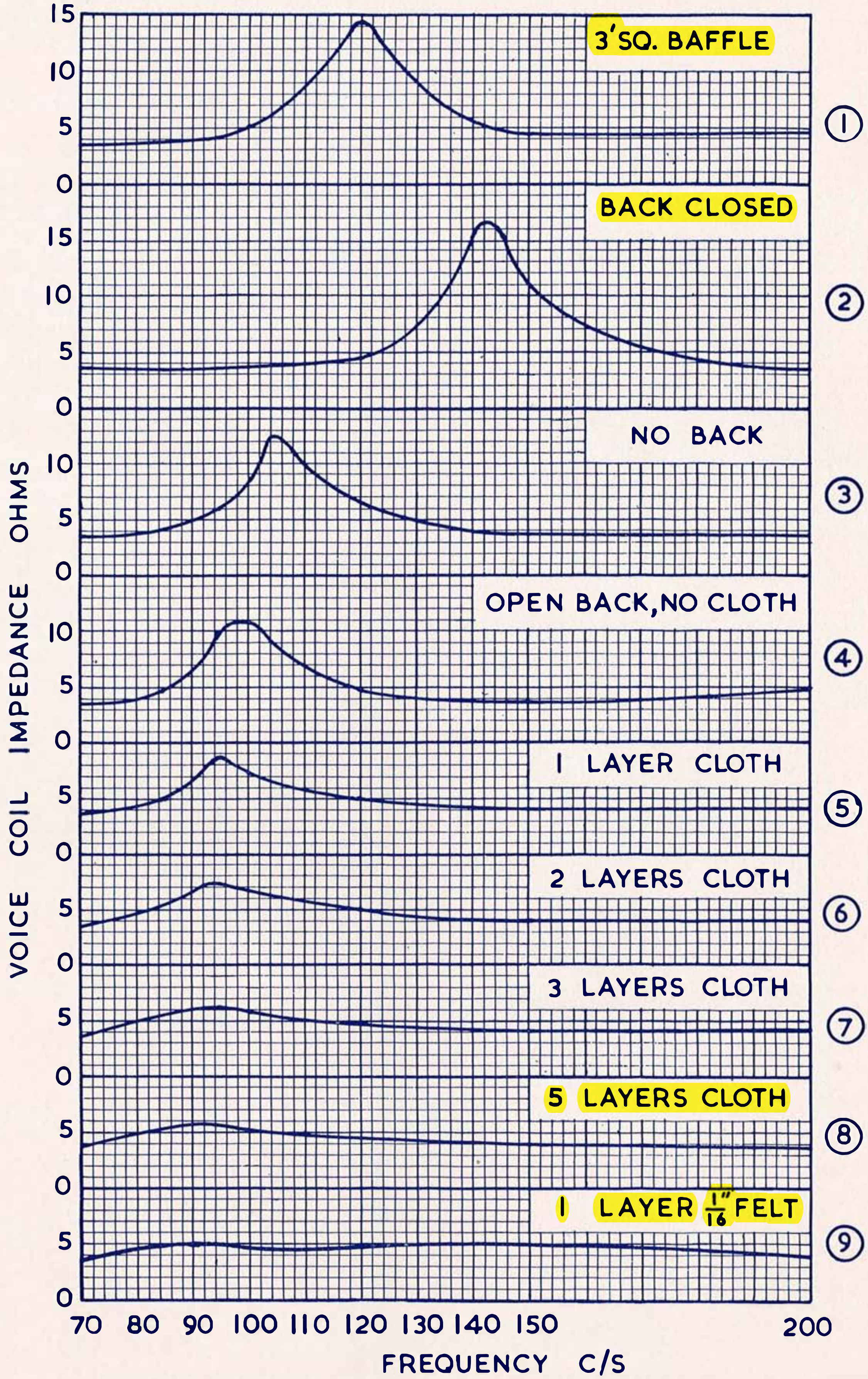


Fig. 7. Curves of voice control impedance voice control under conditions specified in table.

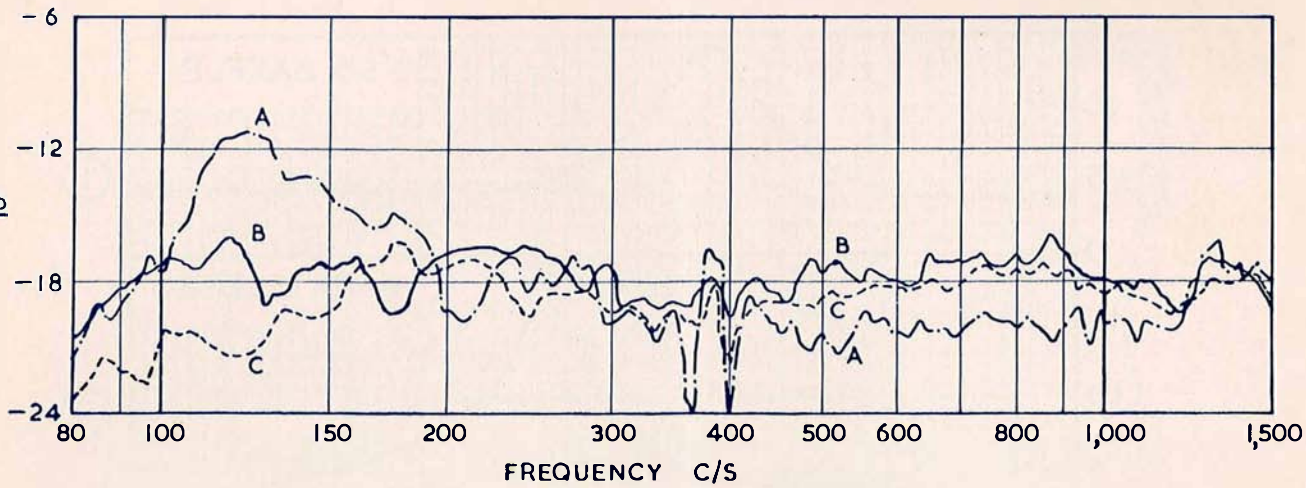


Fig. 8. Frequency response of loudspeaker— microphone 1 ft. in front of speaker. (A) on 3 ft. baffle; (B) in cabinet with 3 layers of cloth; (C) in cabinet with 1 layer of $\frac{1}{16}$ in. felt.

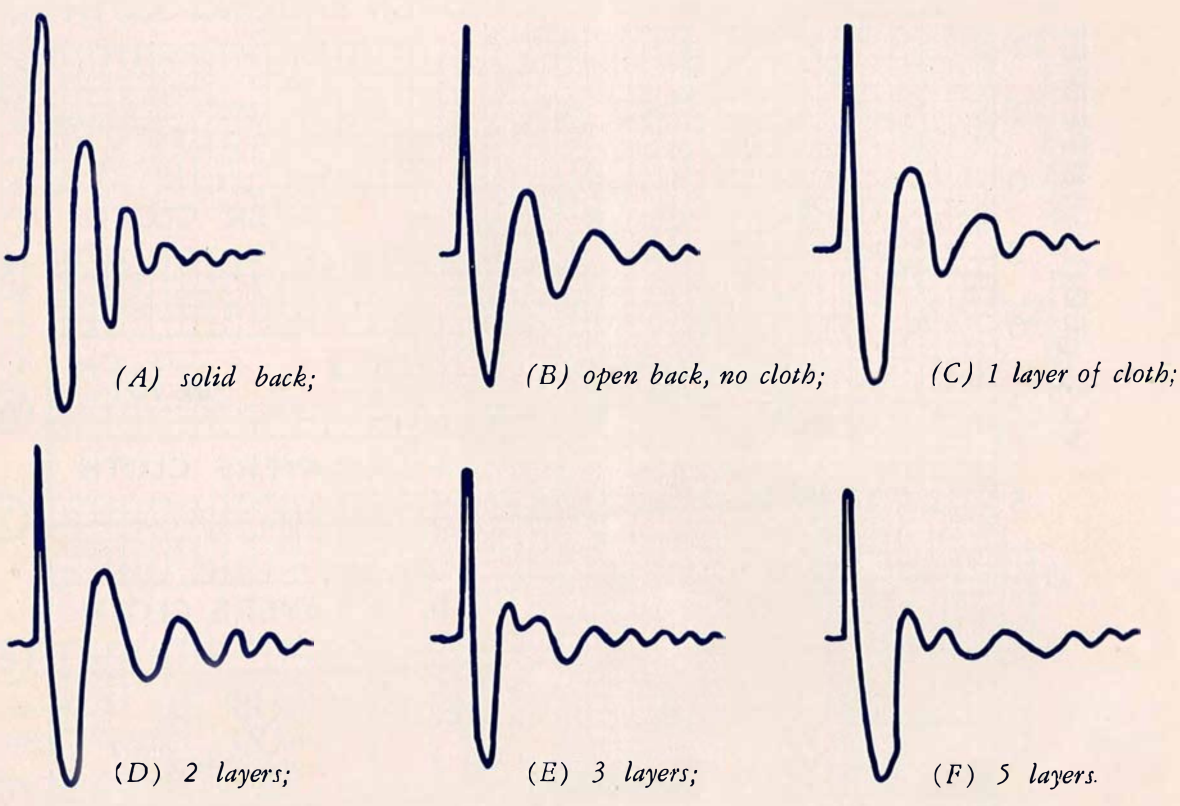


Fig. 9. Transient responses

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2. Taken from Radiotron Designer's Handbook, 4th ed., p.844.
3. Wilkins, C. A., "Variable damping factor control", Audio 38.9 (Sept., 1954), 31. See also High Fidelity (Nov., 1954), 89.
4. Bauer, B. B., "Acoustic damping for loudspeakers", Trans. I.R.E. — PG A AU1.3 (May/June, 1953), 23.

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LOUDSPEAKER ENCLOSURE DESIGN

by E. J. Jordan

Goodmans

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LOUDSPEAKER ENCLOSURE DESIGN

By E. J. JORDAN*

I.—Alternative Methods : Their Advantages and Disadvantages

IN the first part of this article the theory underlying the principal types of loudspeaker enclosure is reviewed, and formulæ associated with the major design factors are given.

This will be followed later by a discussion of some recent developments in which an improved low-frequency performance has been achieved in cabinets of relatively small volume.

THE loudspeaker enclosure has the task of doing something (useful or otherwise) with the low-frequency radiation from the rear of the loudspeaker cone which would otherwise cancel the radiation from the front of the cone.

Before examining various methods of overcoming this, let us establish the principles on which our future arguments will be based.

We shall regard the moving parts of a loudspeaker as a mechanical system which at low frequencies is analogous to an electrical circuit, as shown in its simplest form in Fig. 1.

The complete analogy is revealed by an examination of the electrical and mechanical equations viz.

$$\text{Force} = M\frac{d^2S}{dt^2} + R\frac{dS}{dt} + SK$$

$$\text{E.m.f.} = L\frac{d^2Q}{dt^2} + R\frac{dQ}{dt} + \frac{Q}{C}$$

where M = mass, L = inductance, S = displacement, Q = charge, C = capacitance, K = stiffness and R = resistance.

There are, of course, other analogies, but the above lends itself more readily to discussions of the proposed nature.

Assume for a moment that the loudspeaker is mounted on an infinite baffle. It will be seen, that the power developed in R_a (Fig. 1) is a function of the current through it. Comparing the above equations it will be seen that $i\left(=\frac{dQ}{dt}\right)$ is analogous to the cone velocity $v\left(=\frac{dS}{dt}\right)$. Hence it is the cone velocity and not the displacement, that is responsible directly for the radiated output power, v^2R_a.

From this it would seem that, if the radiated power is to be independent of frequency, the resistive components of the circuit should be high relative to the reactive components. This is not so in practice, since at frequencies where the wavelength is longer than twice the cone diameter the value of R_a falls as the frequency is lowered. The reactance of M_c also falls however, and the increasing velocity resulting from this may largely compensate for the fall in R_a to the extent that the radiation remains substantially constant, down to a frequency where $\omega M_c - \frac{1}{\omega C_c} \rightarrow 0$. Here the velocity of the cone rises sharply and is limited only by R_d, R_c and R_a. This produces an increase in the radiated power and is the resonant frequency of the loudspeaker.

Below this frequency the impedance of the circuit rises as the frequency falls due to the reactance of C_c, consequently the radiation falls very sharply. The resonant frequency may thus set the limit to the low-frequency response of the loudspeaker.

The above may be shown by considering the expression for the radiated power at the frequencies being discussed:

$$P = v^2 R_a = \frac{\text{Force}^2}{Z^2_M} \cdot \frac{2\pi\rho f^2}{c} \cdot (\pi r^2)^2, \text{ where } r \text{ is}$$

the radius of the cone.

Above resonance if $R_M << X_M$ (mass)

$$P \propto \frac{\text{Force}^2}{X^2_M} \cdot f^2$$

This is the condition of mass control, and since $X_M{}^2 \propto f^2$, P is independent of f.

Above, at, or below resonance, if $R_M >> X_M$

$$P \propto \frac{\text{Force}^2}{R_M{}^2} \cdot f^2 \propto f^2$$

This is the condition of constant velocity, and P falls with f at the rate of 6dB/octave.

Below resonance if $R_M << X_M$ (stiffness),

$$P \propto \frac{\text{Force}^2}{X_M{}^2} \cdot f^2 \propto f^4$$

This is the condition of constant amplitude and P falls with f at the rate of 12dB/octave.

Above resonance if R_M is comparable to X_M

$$P \propto \frac{\text{Force}^2}{Z_M{}^2} \cdot f^2$$

and P falls with frequency at a rate determined by the ratio $\frac{f^2}{R_M{}^2 + X_M{}^2}$.

In all cases the radiation resistance is small

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relative to the total mechanical impedance of the system; its effect on the velocity has therefore been neglected.

So far it has been assumed that the loudspeaker is mounted in an infinite baffle. The analogous circuit is similar to that of a loudspeaker mounted in free air, except that the baffle produces a large increase in R_a and a small increase in L_a.

It is very important to realize that any baffle or enclosure may be represented in the analogy by a series impedance Z_b which will tend to reduce the cone velocity, but, depending upon the nature of this additional impedance, partial compensation may be effected by resonant phenomena over at least part of the low-frequency range.

The effective mechanical impedance presented to the cone due to any acoustical impedance Z_A is given by: $Z_M = Z_A(\pi r^2)^2$
where Z_A is the vector sum of Z_r and the acoustic impedance due to the mounting. At low frequencies:

$$Z_r = R_r + j\omega L_r \approx \frac{2\pi\rho f^2}{c} + j\frac{0.85\rho\omega}{\pi r}$$

Impedance Curves — A very convenient way of measuring the effects of the enclosure on the output of the loudspeaker is to plot the impedance/frequency curve of the loudspeaker when housed in the enclosure. If a base line is drawn at a value equal to the clamped impedance of the voice coil then the impedance curve relative to this line is directly proportional to the velocity of the cone.

The relationships between the electrical impedance (Z_E) the mechanical impedance (Z_M) and the velocity (v) of a loudspeaker system are as follows: where B = flux density in the magnet system, l = length of voice coil conductor enveloped by flux and i = current flowing in coil.

Back e.m.f. due to the motion of the coil:

$$E \propto Blv = \frac{B^2l^2i}{Z_M}$$

Motional impedance of the coil:

$$Z_m = \frac{E}{i} \propto \frac{B^2l^2}{Z_M}$$

Total electrical impedance:

$$Z_E = Z_{vc} + Z_m$$

where Z_{vc} is the clamped impedance of the voice coil.

From above $v \propto \dfrac{1}{Z_M} \propto Z_m$

If the component parts of Z_M are expressed in c.g.s. terms then Z_m will be in electromagnetic units. Impedance curves often give a far more accurate assessment of the performance of an enclosure than pressure response curves since the latter depend not only on the cone velocity but, in the case of vented enclosures, upon the port radiation as well. Pressure curves are also greatly affected by diffraction and while they are invaluable in demonstrating the overall radiation from a loudspeaker system, they do not show clearly the action of the various acoustic components due to the enclosure on the loudspeaker cone.

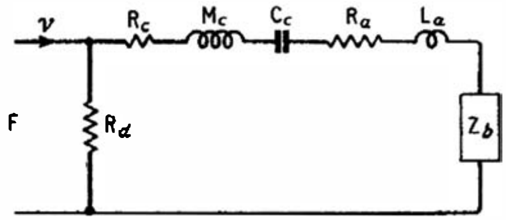


Fig. 1. Simplified electrical analogue of the mechanical properties of a moving coil loudspeaker.

Wall Mounting — The nearest practicable approach to the infinite baffle condition is by mounting the loudspeaker in a wall e.g. a partition wall between two rooms.

This method of baffling a loudspeaker ensures complete separation between the front and rear radiation of the cone and imposes a relatively low mechanical impedance to the cone velocity. The extent of the low-frequency response is limited by the resonant frequency of the cone.

For wall mounting it is therefore desirable to use a loudspeaker having a low frequency, highly damped cone resonance. The damping in this case will be mainly electromagnetic, i.e. a high value of R_d in the analogy, tending to produce constant velocity conditions and resulting in a falling low frequency response, as we have shown. Since under these conditions the cone displacement at resonance does not exceed the level required to maintain the velocity constant, a considerable amount of bass lift may be applied from the amplifier to compensate for this loss at low frequencies. The bass lift required commences at the frequency at which the wavelength is equal to twice the cone diameter and has a slope which may be determined either aurally or from the expressions previously given, the latter being possible only when the necessary loudspeaker parameters are known.

SYMBOLS

c = velocity of sound in air
C_b = compliance of air in closed cabinet
C_c = compliance of cone suspension
F = force applied to cone
k = ω/c = wave constant
L_a = $L_r(\pi r^2)^2$
L_r = acoustic radiation mass
M_c = mass of cone system
M_v = mass of air in vent
P = radiated acoustic power

R_a = $R_r(\pi r^2)^2$
R_c = resistance due to friction in cone
R_d = mechanical resistance due to voice coil damping
R_M = total mechanical resistance
R_r = radiation resistance
R_s = viscous resistance of vent
R_v = total resistance component of vent = $R_r + R_s$
v = velocity of cone

X_{cb} = reactance of air in closed cabinet
X_M = total mechanical reactance
Z_A = total acoustic impedance
Z_r = acoustic radiation impedance
Z_b = impedance due to loudspeaker mounting
Z_M = total mechanical impedance
Z_m = motional impedance of coil
μ = coefficient of shear viscosity
ρ = density of air
ω = $2\pi f$

C.g.s. units for mechanical and acoustical quantities, and e.m. units for electrical, have been assumed throughout.

A consideration which should be borne in mind, particularly in the case of wall mounting, is that the aperture in which the loudspeaker is mounted will behave as a tube of length equal to the thickness of the wall or baffle, and in so doing will exhibit a number of harmonically related resonances and anti-resonances causing irregularities in the treble response. There are of course, a number of obvious remedies for this, e.g. bevelling the edges of the aperture or mounting the loudspeaker on a sub-baffle.

Finite Baffles—If the baffle is finite, at some low frequencies, depending on its size, back-to-front cancellation will occur and the limiting baffle size for a given low-frequency extension is:

$$l = \frac{c}{2f}$$

if the baffle is rectangular and l is the length of the smallest side.

If the bass response is to extend down to a reasonably low frequency the necessary baffle size will be relatively large, e.g. a square baffle suitable for reproduction down to 60 c/s will have a side of 9.42ft. A loudspeaker acting as a treble unit in a crossover system should be mounted on a baffle large enough to work down to half the crossover frequency.

For the sake of convenience baffles often take the form of open-backed cabinets. In such cases, in addition to the normal baffle action, the cabinet will behave more or less, according to its depth, as a tuned pipe and will exhibit a number of harmonically related resonances, the lowest of which will approximate to:

$$f = \frac{c}{2\,(l + 0.85r)}$$

where l is the depth of the cabinet, $r = \sqrt{A/\pi}$ if A is the area of the open back.

It is these resonances that contribute to the unnatural "boomy" quality evident in many commercial reproducers.

Closed Cabinets—Alternatively a method of preventing back-to-front cancellation is to completely enclose the rear of the loudspeaker cone. Under these conditions however, the enclosed air will apply a stiffness force to the rear face of the cone.

This may be represented by a mechanical reactance X_{cb} the value of which is given by:

$$\frac{\rho c^2 (\pi r^2)^2}{\omega V}$$

where πr^2 = piston area of cone and V = volume of enclosure.

In the analogy this reactance appears as a series capacitance as shown in Fig. 2.

In order not to raise the cone resonance unduly, the value of C_b must be large relative to C_c. Since, for a given loudspeaker system, C_b is the only variable, it must be large.

It has been found that, for a 12-in loudspeaker having a fundamental cone resonance at 35 c/s, the volume of an enclosing box would need to be of the order of 12 cu ft for its reactance to be sufficiently low not to impair the low-frequency performance of the speaker.

There are a number of factors in the design of loudspeaker enclosures which should be considered.

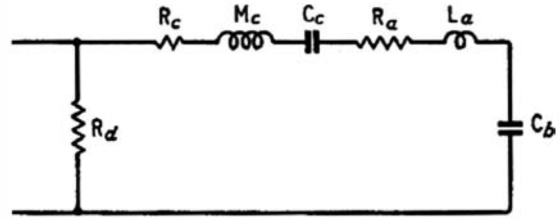


Fig. 2. Analogue circuit of a moving coil loudspeaker in a closed cabinet.

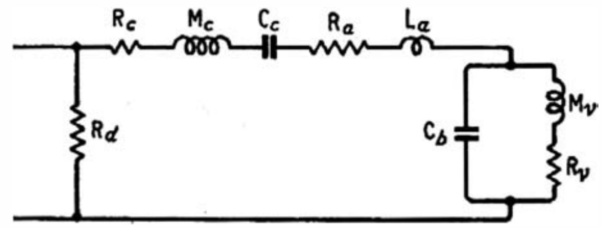


Fig. 3. Analogue circuit of a moving coil loudspeaker in a vented cabinet.

These are common to most types of enclosure and are:

Shape of the Enclosure—As the frequency is lowered the radiated wavefront from the loudspeaker cone tends to become spherical, consequently the boundary edges of the loudspeaker enclosure constitute obstacles in the path of the wavefront. This results in (a) bending of the wavefront (diffraction) and (b) secondary radiation from these edges. This secondary radiation will produce interference patterns causing irregularities in the frequency response of the system.

These effects are largely dependent on the shape of the enclosure and will be smallest for a spherical enclosure and greatest for a cube. Since the cabinet has to be a presentable piece of furniture, there are certain limitations on its shape. Fortunately however, the effects of diffraction are not very serious and it is not difficult to reach a compromise.

Corner Position—Consider a source of sound that is small compared to a wavelength and situated in free space. The radiation from this source will be of equal intensity at a given distance in all directions, i.e. spherical.

If a large flat wall is placed near the sound source then the total radiation will be concentrated into a hemisphere and its intensity will then be doubled. Similarly, if a second wall is placed near the sound source at right angles to the first the total radiation will be concentrated into one-quarter of a sphere and its intensity will be four times greater. A third wall at right angles to the other two will increase the intensity eight times.

A loudspeaker standing in the corner of the room may at medium-low frequencies, be regarded as similar to the second case and approaching the third case as the frequency falls to a point where the wavelength is much greater than the height of the speaker above the floor.

Construction—At frequencies where the wavelength is comparable to the internal dimensions of the enclosure reflections between inside wall faces will occur resulting in standing-wave patterns which in turn will produce irregularities in the frequency response of the system.

These standing waves may be considerably reduced (a) by lining the enclosure walls with soft felt or wool thus providing absorption at points of maximum pressure, (b) by hanging curtains of the same material near the centre of the enclosure, thereby introducing resistance at points of maximum velocity.

A further point to be considered is that the material (usually wood) from which the enclosure is made, possessing both mass and compliance, will be capable of movement and will resonate at one or more frequencies and in so doing will (a) behave as a radiating diaphragm and (b) modify the air loading on the cone, both of which will produce unwanted coloration in the reproduction. Therefore the enclosure should be made of as thick and dense a material as possible.

Vented Enclosures, Reflex Cabinets — One method of overcoming the disadvantage of the closed cabinet is to include in the cabinet wall an orifice or vent.

A suitably vented enclosure will apply to the rear of the loudspeaker cone an impedance which offers the cone a maximum degree of damping at or near its resonant frequency and the radiation from the vent around this frequency will be more or less in phase with the frontal radiation from the cone, i.e., the back radiation is inverted. Before we describe the nature of this impedance we will describe the Helmholtz resonator, the principle on which the design of vented and reflex cabinets is based.

For the benefit of readers not familiar with this resonator, it consists simply of a partially enclosed air cavity having a communicating duct to the outside air.

An enclosed volume of air will have a stiffness reactance equal to $\rho c^2/\omega V$.

The air in the duct will move as a homogeneous mass, the reactance of which is given by:

$$\frac{\rho l' \omega}{\pi r_v^{\,2}}$$

where $\pi r_v^{\,2}$ is the cross-sectional area, and l' is the effective length of the duct.

This system will have a resonant frequency at which the mass of air in the duct will move most readily, bouncing as it were, on the elasticity of the air in the enclosure. This occurs when the sum of the reactances, which are opposite in sign, is zero.

Equating the two expressions and transposing for f, we have

$$f = \frac{c}{2\pi}\sqrt{\frac{\pi r_v^{\,2}}{Vl'}}$$

which is the usual expression for the natural frequency of a Helmholtz resonator.

In actual fact this is only an approximation since the full expression for the mass reactance should contain a Bessel term for the load on the vent, due to the air outside the cabinet, but in practice this is small enough to be neglected.

Some of the air adjacent to the end of the duct moves with the air in the duct and thus becomes added to it. The effective length of the duct therefore, is greater than its actual length. Rayleigh shows that the increase at each end is:

$$\delta l = \frac{8}{3\pi} r_v$$

where r is the radius of the duct.

The total effective length is, therefore:

$$l' = l + \frac{16}{3\pi} r_v = l + 1.7 r_v$$

If the duct is not circular, $r_v = \sqrt{A/\pi}$, where A is the cross-sectional area of the duct.

Returning now to the subject of loudspeaker enclosures, a vented cabinet containing a loudspeaker will exhibit a resonance in accordance with the above description which will be reasonably independent of the loudspeaker cone resonance.

When the cabinet resonance is excited by the loudspeaker the motion of the air in the vent will reach its maximum velocity and will be in phase with the motion of the cone. At this frequency therefore, the air in the cabinet will come under the double compressive and rarefactive forces of both the cone and air in the vent; consequently its effective stiffness rises and the resulting impedance applied to the rear of the cone becomes much higher at this frequency than at any other.

If the resonant frequency of the enclosure is made to coincide with that of the cone, the latter receives maximum damping at its resonance and any peak in the radiated power at this frequency is removed.

In addition to this, the reduction in cone displacement results in a considerable increase in the power-handling capacity of the loudspeaker and

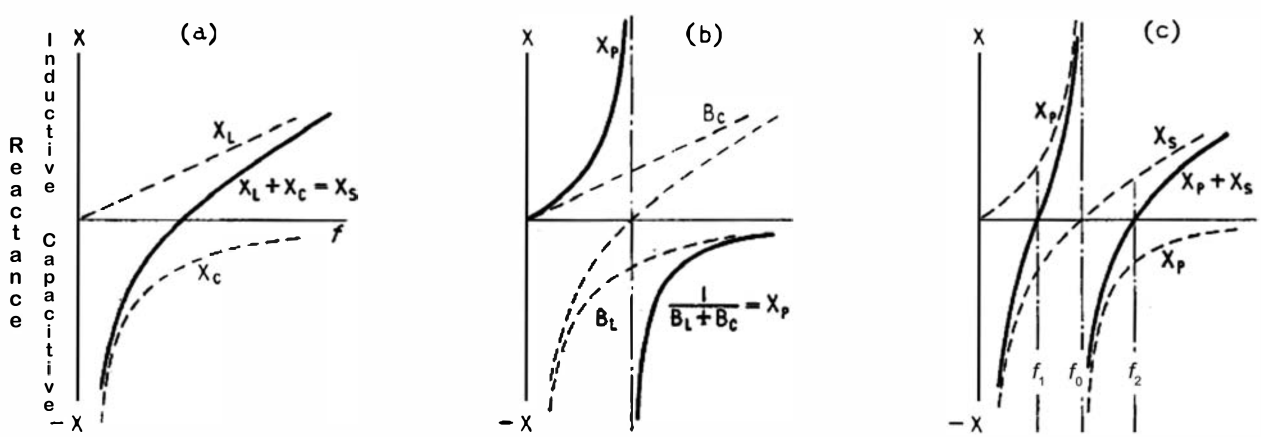


Fig. 4. Variation of reactance with frequency of the circuit elements of Fig. 3:

a) X_L = total mass reactance of the series section, X_C = total stiffness reactance of the series section,

b) B_L and B_C are the mass and compliance suseptances of the parallel section and,

c) X_S and X_P are the total series and parallel reactances, respectively.

in a reduction of harmonic and intermodulation distortion. Although the velocity and therefore the power radiated from the cone is reduced around this frequency, the overall radiated power from the system is increased considerably due to the very high air velocity at the vent. Unlike the cone there is no physical limitation to the displacement of the air in the vent.

Below the resonant frequency of the enclosure the stiffness reactance becomes high and the system behaves as though the air mass in the vent were coupled directly to the mass of the cone. At some frequency the reactance of this combined mass will become equal to the stiffness reactance of the suspension system of the cone. A resonance will occur at this frequency, the amplitude of which will be considerably lower than that of the initial cone resonance and the radiation from the vent will be in anti-phase with that from the cone.

Above the resonant frequency of the enclosure the mass reactance of the vent becomes high, and the cabinet behaves as though it were completely closed, presenting a purely stiffness reactance to the rear of the cone. At some frequency the combined stiffness reactance of the cone suspension system and the enclosure will become equal to the mass reactance of the cone. At this frequency a further resonance will occur, and again the amplitude will be considerably less than the cone resonance.

Now let us consider the nature of the impedance presented to the rear of the cone by a vented enclosure. Since this impedance rises to a maximum value, a parallel tuned circuit is indicated in the analogy Fig. 3, where R_v and M_v are the vent components.

By drawing the reactance sketches for the complete system, we are able to see clearly the derivation of the resonant frequencies described above Fig. 4.

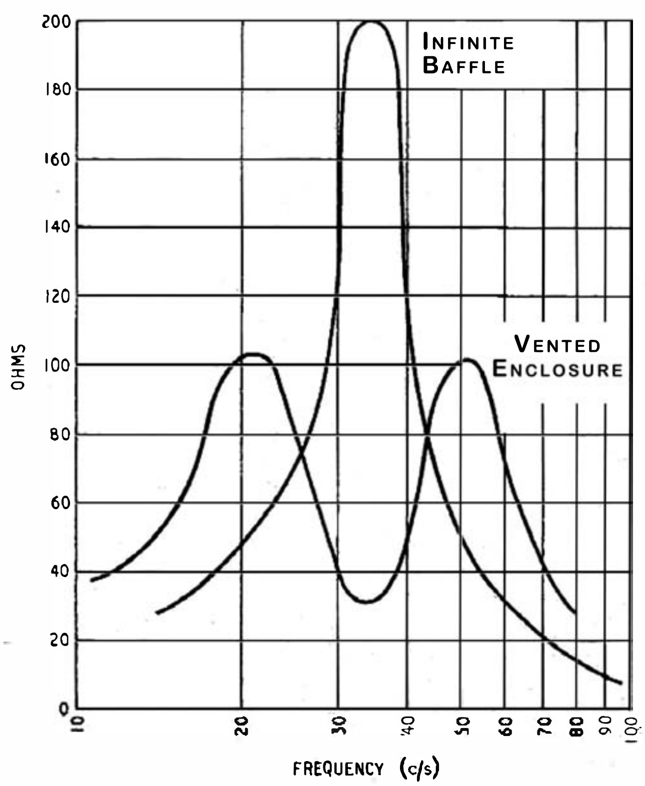


Fig. 5. Impedance/frequency response of a loudspeaker on an infitite baffle and in a vented enclosure.

Figs. 4(a) and 4(b) show the well-known reactance sketches for the series and parallel sections of the circuit respectively. When these are added, we have Fig. 4(c) which exhibits three critical frequencies f_1, f_o and f_2. It will be noticed that at f_1 and f_2 the reactance falls to zero, and at f_o rises to infinity. The corresponding impedance curve, together with the impedance curve, taken with a loudspeaker mounted on an infinite baffle is shown in Fig. 5.

Whilst a reflex (vented) enclosure is much smaller than a completely closed cabinet for a given bass extension, the reduction in size is limited by the mechanical impedance it imposes on the cone, at frequencies away from its resonance (f_o). In the design of these enclosures it is important therefore to calculate the impedance over a wide range of frequencies, to ensure that this does not become excessive.

To accomplish this, the various components of the enclosure are expressed as follows:
Referring to Fig. 3.

$$C_b = \frac{V}{\rho c^2}$$

$$R_v = R_s + R_r$$

$$M_v = \frac{\rho l}{\pi r_v^2}$$

R_s is resistance due to air viscosity in vent

$$= \frac{\sqrt{2\mu\rho\omega}}{\pi r_v^3}\, l$$

R_r is radiation resistance of port $= \dfrac{\rho c k^2}{2\pi}$

Having already met the first two expressions, the new symbols appearing in the second two expressions are: μ, the coefficient of shear viscosity and $k = \omega/c$, the so-called wave number or wave constant.

It is convenient to express all dimensions in c.g.s. terms.

The acoustical impedance of the enclosure Z_{ab} may be obtained from the usual expression for an L C R circuit of this type, i.e.

$$Z_{ab} = \frac{R_v - j\omega[C_b R_v^2 + M_v(\omega^2 M_v C_b - 1)]}{\omega^2 C_b^2 R_v^2 + (\omega M_v C_b - 1)^2}$$

where all terms are in acoustical units.

Expressing this as the modulus of the mechanical impedance, we have:

$$|Z_b| = \left[\frac{R_v^2 + \omega^2 M_v^2}{\omega^2 C_b^2 R_v^2 + (\omega M_v C_b - 1)^2}\right](\pi r^2)^2$$

At the resonance of the enclosure, the right-hand expression in the denominator becomes zero, the

Z approximates to $\dfrac{M_v}{C_b R_v}(\pi r^2)^2$.

This is the dynamic impedance of the circuit and is the value of a purely resistive component which may replace the parallel circuit at a resonance in the analogy.

The "Q" of the enclosure is given by $\dfrac{\omega M_v}{R_v}$ and is normally much higher than that of the cone system and is therefore not critical. It has been found that an optimum performance is given by the reflex enclosure if the cross-sectional area of the vent

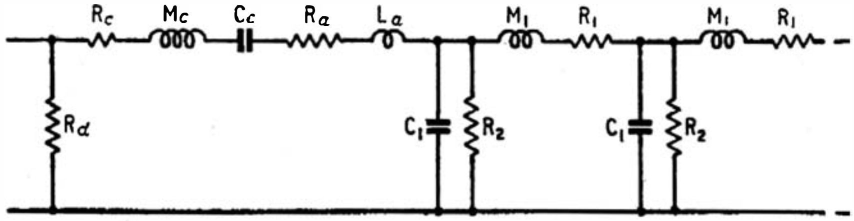


Fig. 6. Equivalent circuit of a moving coil loudspeaker mounted in a labyrinth. M_1 and C_1 are the labyrinth distributed mass and compliance respectively while R_1 and R_2 are viscous and absorbtion resistances.

is made approximately equal to the piston area of the cone.

The required enclosure volume for coincident resonance is then obtained from a derivation of the formula for a Helmholtz resonator and is given by:

$$V=\pi r^2\left[\frac{c^2}{\omega^2}\cdot\frac{1}{l-1.7r}+l\right]$$

In this equation l is the length of the duct or tunnel which usually extends into the enclosure, and the volume of the duct has therefore been added to the expression. Broadly speaking, increasing the tunnel length decreases the overall volume until a point is reached where the increase in total volume due to the increased tunnel length is exceeding the reduction in the volume required to correctly tune the enclosure. The tunnel length for minimum volume is:

$$l=\frac{c}{\omega}-1.7r$$

Another limitation on the length of the tunnel is that it must not exceed 1/12th of a wavelength at the resonant frequency of the enclosure, otherwise the contained air would not behave purely as a mass.

We have seen that the reduction in size of a reflex cabinet is limited by the increase in mechanical impedance presented to the cone.

There are however, marketed enclosure designs which are based on the foregoing principle. These are extremely small, yet appear to have a substantial bass response.

It is evident from the expression for the resonant frequency of a vented enclosure that the enclosed air volume may become as small as we like, and the resonant frequency made low by having a very small vent and tunnel area. Such an enclosure has a very high mechanical impedance, thereby limiting the cone velocity at very low frequencies. Also, owing to the very resistive nature of the vent, the two lower resonances shown for a loudspeaker in a vented enclosure are highly damped and the upper resonance is prominent, resulting in an accentuated bass radiation around this frequency, hence the apparent bass "efficiency."

The amplitude and frequency of this upper resonance may both be reduced by facing the cone into a restricted aperture such as a slit, but this introduces serious irregularities in the response and will be discussed in a subsequent article.

The Tuned Pipe—This is based on the well-known organ pipe principle. In order to exclude modes of resonance other than the air column resonance the end of the pipe remote from the speaker should be either fully open or fully closed.

In the case of the open pipe resonances will occur at frequencies corresponding to all even numbers of quarter wavelengths and anti-resonances will occur at all odd numbers of quarter wavelengths. For the closed pipe the reverse is true.

One method of applying these properties to loudspeaker mounting, is to use an open pipe with the loudspeaker mounted at one end, the length of the pipe being such that its fundamental anti-resonance coincides with the cone resonance thus securing some of the advantages of a reflex enclosure.

A closed pipe may also be used in the same manner, in which case the length of the pipe need only be about half that of the open pipe. However, the impedance presented to the cone by this method is high, and a serious reduction in cone velocity may result at low frequencies. The radiation from the open end of the open pipe increases the radiation efficiency of this system to some extent.

The length of an open pipe for a given frequency of anti-resonance f is:

$$l=\frac{c}{2f}-1.7\sqrt{\frac{A}{\pi}}$$

where A is the cross-sectional area of the pipe.

The length of a closed pipe for a given anti-resonance frequency f is:

$$l=\frac{c}{4f}-0.85\sqrt{\frac{A}{\pi}}$$

Whilst these pipes are a little more simple to construct than reflex enclosures, their overriding disadvantage is the presence of all resonances and anti-resonances occurring at every quarter wave length, and it is virtually impossible to damp the enclosure and to absorb all the resonances without severely attenuating the required fundamental. A way of partially overcoming this is described in a patent by Voigt. This is to mount the speaker in the wall of a pipe which is closed at one end and open at the other, the position of the loudspeaker being 1/3rd of the pipe length away from the closed end. By this means, the first resonance above the fundamental (3rd harmonic) will be cancelled.

The Labyrinth—The labyrinth consists of a very long tube, usually folded and heavily lined with absorbing material with the loudspeaker mounted at one end. The labyrinth is probably the cleanest way of disposing of unwanted back radiation, which, having left the rear of the loud speaker cone at one end of the tube does not re-appear at the other. It does not really matter therefore whether this far end is open or closed.

The analogous circuit is that of a transmission line and is shown in Fig. 6. The sound energy due to the back radiation from the cone is largely dissipated in the resistive components R_1 and R_2, where R_1 is due to the viscous losses between the air in motion and the lining on the internal surfaces of the labyrinth, and R_2 is due to the absorption of sound energy at these surfaces.

As the frequency is increased, R_1 increases and R_2 decreases. Therefore if the labyrinth is to be effective at the lower frequencies the lining must be fairly thick. If however, this begins to take

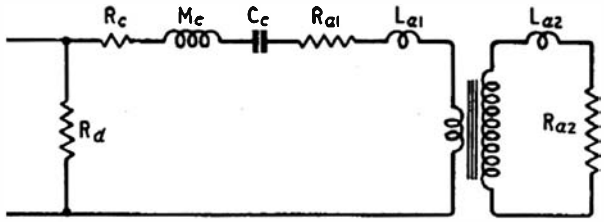


Fig. 7. Electrical analogue of a moving coil loudspeaker with horn loading. R_{a1} and L_{a1} represent the air load on the side of the diaphragm not coupled to the horn. R_{a2} and L_{a2} constitute the air load at the mouth of the horn.

up an appreciable part of the cross-sectional area of the labyrinth, the air loading on the rear of the cone, which is normally quite high in this type of enclosure, will become excessive, resulting in a severe reduction in the radiated power at these frequencies. The cross-sectional area should therefore be at least equal to the piston area of the cone, and to achieve the necessary dissipation of sound energy from the rear of the cone, the effective path length of the labyrinth should be as great as possible, the minimum length being set empirically at a quarter wavelength equivalent to the lowest frequency to be reproduced.

Under these conditions the impedance presented to the rear of the cone is quite high and mainly resistive so that the cone approaches constant-velocity operation and behaves in the manner previously described for this condition.

The Horn—Horn loading is the most efficient form of loudspeaker mounting and, if the horn were large enough, it would give a performance superior in every respect to any other form of loudspeaker mounting.

The action of the horn can be most readily grasped by consideration of the analogous circuit.

The major problem in all the systems so far discussed has been to compensate for the fall in R_a at low frequencies. The use of a transformer would be an obvious answer if this problem were an electrical one, and, applying this to the analogy, we have Fig. 7. Acoustically, such a transformer is analogous to the horn, which may be used to match the relatively high mechanical impedance of the loudspeaker cone to the radiation resistance, and, by making the mouth of the horn large, this resistance does not become so low at low frequencies.

From the analogy, since the effective radiation resistance reflected back to the primary of the transformer is very high, the cone operates under constant velocity conditions and no resonance is evident.

Below a certain frequency the acoustic resistance of a horn falls sharply and its reactance (mass) rises. This cut-off frequency is determined by the dimensions of the horn and, since size-for-size an exponential horn maintains its efficiency to a lower frequency than a conical horn, the former is more often used. The cross-sectional area (A_x) of the exponential horn at any distance x from the throat is given by:

$$A_x = A_0 \epsilon^{mx}$$

where A_0 is the throat area and m the flair constant.

The cut-off frequency is given by: $f_c = \dfrac{mc}{4\pi}$

The diameter of the mouth should not be less than a quarter wavelength at f_c, otherwise the horn will tend to exhibit the resonances similar to a tuned pipe.

Most text books on electro-acoustics deal very fully with the horn, and there is little point in our doing so here, especially since, due to its size, an adequately large horn is rarely encountered. Although many small folded horn designs are capable of impressive (if not accurate) reproduction, let it suffice to say that a horn capable of presenting a constant radiation resistance down to 30 c/s to the cone of a 12-in loudspeaker would be over 12ft long and have a mouth diameter of about 9½ft.

Conclusion—The different types of loudspeaker enclosures number as many as the possible combinations of L C R in series with the analogous cone circuit.

Some time ago, the thought arose that an excellent method of designing a loudspeaker enclosure would be to state the ideal velocity characteristics, and then determine an electrical impedance which, when placed in series with the analogous cone circuit, would produce these characteristics. It would then remain to transpose this impedance into acoustical terms and to evolve an enclosure having the required component values.

This line of development has been followed to a successful conclusion and will be described in the second part of this article.

LOUDSPEAKER ENCLOSURE DESIGN

By E. J. JORDAN*

2.—A Cabinet of Reduced Size With Better Low-frequency Performance

IN the first part of this article the features of performance and design of the principal methods of mounting a loudspeaker were reviewed. These may be briefly summarized in order of merit, as follows.

Full Horn — Acoustically this is the ideal method of loudspeaker mounting. It provides excellent air loading on the cone, is devoid of self-resonance and possesses a high radiation efficiency down to any desired frequency being limited only by the horn dimensions. The disadvantage of the horn is the very great size required for effective operation down to very low frequencies.

Absorbing Labyrinth — This again presents excellent resonance-free air loading on the loudspeaker cone and in this respect is comparable to the horn. It is effective down to any desired frequency, being limited, like the horn, by its dimensions. Unlike the horn however, the disadvantage of this system is the falling efficiency at low frequencies due to the approach to constant-velocity conditions although this may be partially compensated for in the amplifier. A labyrinth capable of good absorption down to very low frequencies is still rather big.

Reflex Enclosure — The advantage of the reflex cabinet is that excellent damping is applied to the loudspeaker cone at its resonance where it is most required. A further point in its favour is that it is relatively simple to construct. The bass response from a reflex enclosure will have an efficiency somewhat higher than that from a labyrinth and for a given bass extension, will be smaller, although it still makes a rather dominating piece of furniture in the drawing room. The response will not be so smooth as for a labyrinth due principally to the upper of the two resonances common to this type of mounting. If very much bass boost is applied the reflex enclosure will tend to sound boomy, also port radiation at the lower of the two resonances will tend to cancel that from the cone.

Wall Mounting or Large Flat Baffle – This type of loudspeaker mounting presents a lower impedance to the rear of the loudspeaker cone than any other.

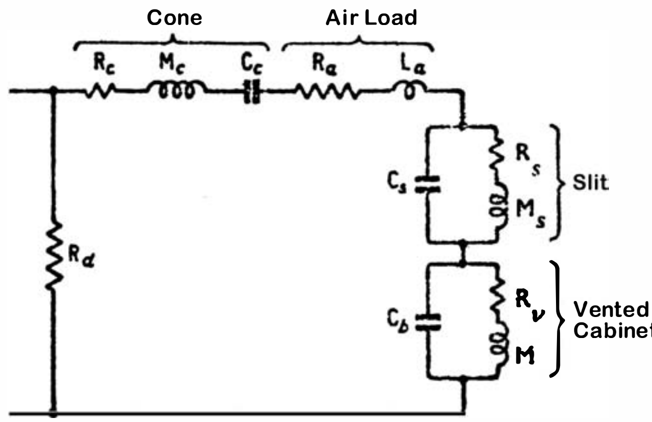


Fig. 8. Electrical analogue of a loudspeaker/cabinet system incorporating an additional restricted aperture in front of the cone. M_s and R_s are the mass and compliance associated with the slit and C_s is the compliance formed between the cone and the inside face of the orifice.

Therefore, with the exception of horn loading, this system has the highest efficiency amoung direct radiators. The low acoustic damping to the cone however makes necessary the use of a loudspeaker with a high degree of electromagnetic damping if excessive cone velocity is to be avoided, in which case the relative efficiency of the system at low frequencies is lost and its performance will be similar to that of a labyrinth.

Recent Trends – It has for years been the ambition of designers to produce a loudspeaker system having the performance of a horn and the dimensions of an orange crate. Many audio designers have examined the possibilities of small, compromise horn-type enclosures since these may be capable of very impressive reproduction. The writer however prefers to aim for accuracy. The horn cannot be effectively compromised and good reproduction from, say, 50 c/s to 30 c/s demands an enormous horn. In any case it is questionable whether such high efficiency is necessary from a given loudspeaker unit. The labyrinth will

*Goodmans Industries Ltd.

SYMBOLS

C_b = compliance of air in closed cabinet
C_c = compliance of cone suspension
C_s = compliance of air between cone and front baffle slit
L_a = $L_r(\pi r^2)^2$
L_r = acoustic radiation mass
M_c = mass of cone system
M_s = mass of air in slit
M_v = mass of air in vent or orifice
R_a = $R_r(\pi r^2)^2$
R_c = resistance due to resistance in cone
R_d = mechanical resistance due to voice coil damping
R_r = radiation resistance
R_s = viscous resistance of vent
R_v = total resistance component of vent = $R_r + R_s$
v = velocity of cone
Z_b = impedance due to loudspeaker mounting
Z_r = acoustic radiation impedance
ω = $2\pi f$

C.G.S. units for mechanical and acoustical quantities, and E.M. units for electrical, have been assumed throughout.

secure the same downward extension of bass and freedom from resonance as a horn many times its size. Admittedly the amplifier is called upon to supply a few more low-frequency watts, but for normal requirements this is well within the capabilities of any of the well-known 10 or 15-W amplifiers. Even if an additional bass boost circuit has to be fitted, the cost and trouble is still hardly comparable to that of horn construction.

Space-saving considerations give the reflex enclosure a very great advantage over the other systems mentioned; in addition the acoustic characteristics are very good, and the principle suggests itself as being more amenable to compromise than that of the horn. A great deal of experimental work has been directed therefore to reducing still further the size of a reflex enclosure and improving its performance.

We saw in the previous article that, if its size is reduced, the reflex enclosure will present a higher impedance to the rear face of the cone at all frequencies, and, due to the increased impedance of a smaller port, the upper resonant frequency will become unduly prominent. We mentioned also that facing the cone into a restricted aperture or slit would reduce the resonance. This may now be explained by considering the analogous circuit (Fig. 8). Here the impedance due to the mass and resistance components of the slit appears as the series M_s and R_s shown. Now the lower resonant frequency

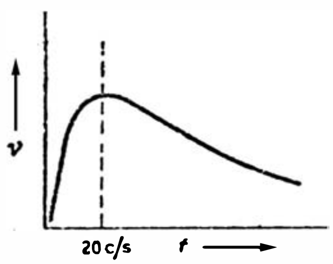


Fig. 9. General form of the velocity/frequency response of the cone required at very low frequencies.

will be substantially due to $R_cM_cC_cR_aM_sR_sM_vR_v$ in series and the upper resonant frequency due to $R_cM_cC_cR_aM_sR_sC_b$ in series. Since the impedance of M_s and R_s forms a greater proportion of the total mass reactance and resistance in the second case the upper resonance f_2 will be lowered both in frequency and amplitude to a greater extent that f_1 (see Fig. 11). A vertical slit also has the advantage of diffusing the higher frequencies horizontally due to its approaching a line source.

The condemning feature of the slit (or any other reduced orifice in front of the loudspeaker cone) is that in conjunction with the cavity (C_s) formed between the cone and the inside face of the material forming the slit, it constitutes a Helmholz resonator which makes itself heard very forcibly somewhere in the middle frequency (300 c/s-700 c/s) range. Standing waves also occur between the cone and the inside face, causing irregularities noticeable in the treble (1,000 c/s-5,000 c/s). Therefore, we may frown upon slits. :-)

It is better to form the impedance M_s and R_s behind the cone by fitting, for example, a cowling† over the rear of the loudspeaker which has an outlet of restricted area, or, as is described in a patent held by Murphy Radio, a corrugated cardboard cylinder is fitted over the rear of the speaker, so that the rear radiation must pass through the small tubes so formed.

These systems represent a very considerable improvement over the slit, although they still tend to introduce slight irregularities in the response. It is surprising, how efficiently even a cardboard drum can behave as a tubed pipe. Nevertheless it must be said the performance of these enclosures is very good for their size and at low frequencies is comparable to that of a full-sized labyrinth. Like the labyrinth they present a high resistive impedance to the rear of the loudspeaker cone; their efficiency is therefore low. It will be seen that M_s and R_s in the analogous circuit will tend to reduce the cone velocity at all frequencies. These components do therefore constitute a further loss of efficiency.

The reader should now be well acquainted with the principles involved in the design of the basic type of loudspeaker mounting and with the problems encountered if these designs are comprised. The question of size is a very important one; there is a demand for a really high-quality sound-reproducing system that is small enough to be unobtrusive in a small lounge or flat.

A good approach to the design of such a system would be to state exactly what was meant by " really high quality " and to define the acoustic properties of the system in terms of cone velocity. This can be expressed as a function of mechanical impedance, which in turn may be translated into an analogous electrical impedance. The problem then resolves itself into the solving of the electrical circuitry. This approach led to the design of an enclosure having the desired performance and, proceeding as above, we shall endeavour to show the derivation of this design.

Enumerating the principal qualities of an " ideal " enclosure we have:

(1) Frequency response extended down to at least 20 c/s.

(2) Complete absence of resonances above this frequency.

(3) Small size.

(4) Efficiency as high as possible in keeping with (2) and (5).

(5) Low distortion.

In order to satisfy requirements (1), (2) and (4) the cone velocity must increase progressively as the frequency is lowered to 20 c/s. Therefore, the enclosure must load the cone in such a way as to bring the effective cone resonance down to this frequency. There must be also a sufficiently high resistance component in order to satisfy requirement.

(5) By limiting excessive increase of cone velocity due to resonant conditions.

In the analogy, these conditions are fulfilled by the velocity curve shown in Fig. 9, and the corresponding analogous circuit shown in Fig. 10, where inductive and resistive elements are added to the cone circuit.

As we have seen, a convenient way of adding mass to the loudspeaker cone is to load it by means of restricted orifice or vent. It is preferable to couple this air mass to the rear face of the cone, and since, at the resonance of the system (neglecting here any compliance existing between this air mass and the cone) the radiation from the vent will be in antiphase with that from the front of the cone, in order to produce negligible cancellation, the vent

†Patent applied for by Goodmans Industries.

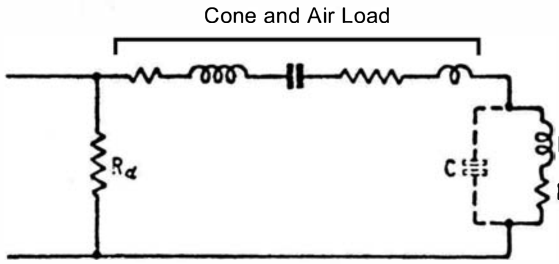


Fig. 10. *Analogue circuit elements added to cone to produce the response seen in Fig. 9.*

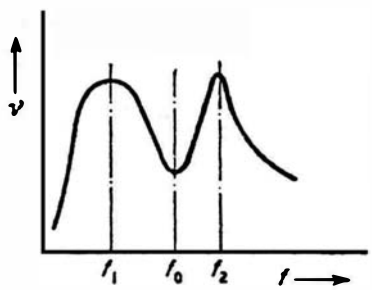


Fig. 11. *Velocity/frequency response resulting from the addition of 'C' in Fig. 10.*

area must be considerably less than the effective piston area of the cone. Therefore, for a given mass reactance a small vent is preferable to a larger vent with a tunnel. As the orifice is reduced, however, the resistance due to viscosity at its edges is increased relatively to the mass reactance, and, whilst to some extent this is desirable for requirement (5) above, a point is reached where the rise in velocity down to the required frequency due to the action of the added mass is severely reduced, resulting in an undue loss of radiated power at these frequencies. This conflicts with requirement (4) above. These considerations therefore fix the port dimensions within fairly narrow limits, quite irrespective of whether the mass reactance from these dimensions is sufficient to reduce the loudspeaker cone resonance from wherever it is down to the required low-frequency limit. Since the mass reactance of the orifice will increase with frequency, it will be necessary to decouple this mass from the cone at the higher frequencies. This requires a shunt capacitance C in the analogy, which may be of such value, that in combination with the mass reactance ωM will produce an effective mass reactance $\omega M'$ having the value required to lower the effective resonance of the series circuit, *i.e.*, the effective cone resonance by the desired amount. Since the capacitance C performs two functions, its value must be determined with both these in mind. For "decoupling" purposes the circuit must become capacitive as soon as possible above f_1 (Fig.11) which indicates that the resonance of the parallel section f_0 should occur a little above this frequency. We shall see later, however, that it is desirable for f_0 to occur above the free-air resonant frequency of the loudspeaker cone. The effect of C on the effective cone resonance may be seen by considering the susceptance of the parallel section, which is:

$$B = \frac{\omega^2 CM - 1}{\omega M}$$

and provided this expression is negative the circuit will behave as an effective inductive reactance of magnitude

$$\omega M' = \frac{\omega M}{1 - \omega^2 CM}$$

To lower the effective cone resonance to a frequency f_1 the sum of the above expression, and the effective reactance of the cone must be zero at this frequency.

$$\text{Effective reactance of cone, } X'_{cone} = \frac{\omega^2 M_c C_c - 1}{\omega C_c}$$

By implication $\omega M'$ is positive at ω_1 and X'_{cone} negative at ω_1.

$$\text{Equating we have } \frac{\omega^2 M_c C_c - 1}{\omega C_c} = \frac{\omega M}{1 + \omega^2 CM}$$

Transposing for C we have

$$C = \frac{C_c}{\omega^2 M_c C_c - 1} - \frac{1}{\omega^2 M}$$

Although a value of C may be found from this expression a lower limit is set to its value by its decoupling function. It is vital that the impedance of the parallel section be well decoupled at frequencies above about 50 c/s.

It is evident that the circuit we now have is analogous to a vented enclosure where the component values have been specially chosen to maintain the radiation efficiency down to 20 c/s. In the previous article we showed how a circuit of this type had three critical frequencies f_1, f_0 and f_2 which resulted in a velocity curve as shown in Fig. 11. In the present case f_1 is our required low frequency resonance and in respect of our second requirement for the "ideal" enclosure the resonances at f_0 and f_2 must be eliminated. (f_0 in the present case is not coincident with the cone resonance.)

We have seen that the resonance at f_0 is due

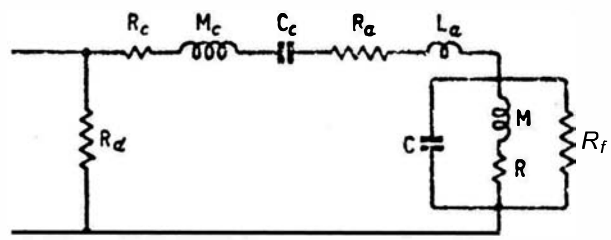


Fig. 12. *Complete analogue of the final design.* R_f *is an added acoustic resistance.*

to the parallel section where its impedance rises to some high value reducing the cone velocity at this frequency. This impedance rise may, of course, be limited by shunting this circuit with a low resistance R_f, Fig. 12, the low limit of R_f being set by its damping effect at f_1.

It has been found possible to choose values of M, C and R_f that are compatible with all the previous considerations and at the same time are such as to reduce the resonances at f_0 and f_2 to negligible proportions.

Let M and C have values producing a reactance characteristic which, relative to that of series components M_c and C_c will be as shown in Fig. 13. The three critical frequencies are shown, and it will be noticed that the reactance of the individual circuits at f_2 is much higher than at f_1. If the effective reactance of M and C in parallel is X_p and this is shunted by R_f, then we may replace this

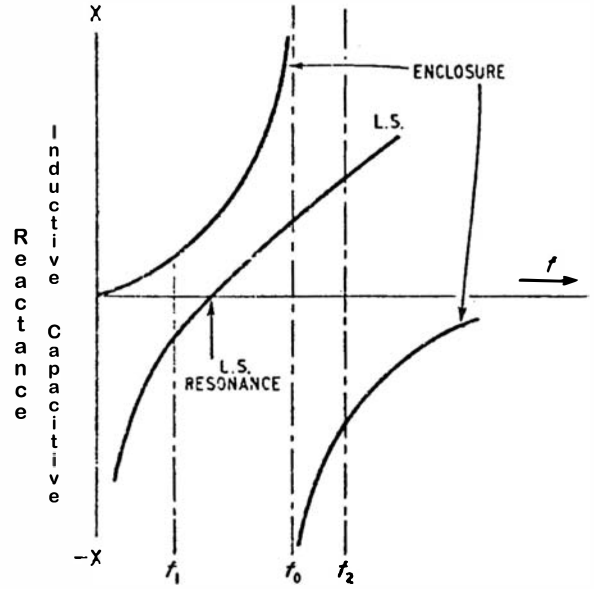


Fig. 13. Reactance characteristic of a loudspeaker in a vented enclosure with the resonant frequency of the enclosure adjusted to be higher than that of the speaker.

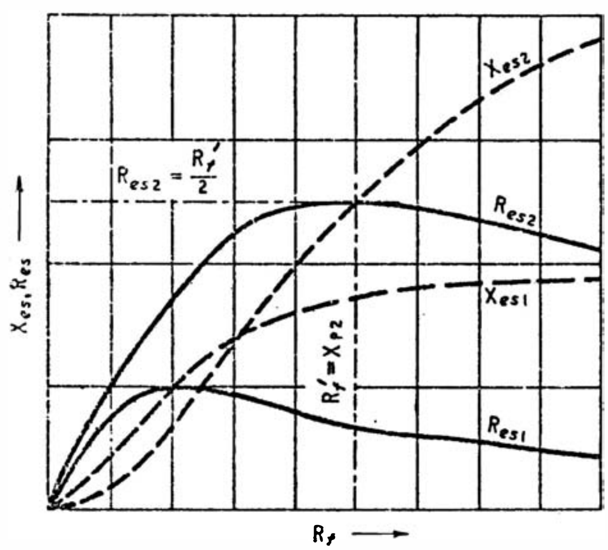


Fig. 14. Variation of X_{es} and R_{es} with R_f for two values of X_p when $X_{p1} < X_{p2}$.

arrangement by an equivalent series circuit consisting of a resistance R_{es} and reactance X_{es} which obey the well-known relationships:—

$$R_{es} = \frac{R_f X_P^2}{R_f^2 + X_P^2} \quad X_{es} = \frac{R_f^2 X_P^2}{R_f^2 + X_P^2}$$

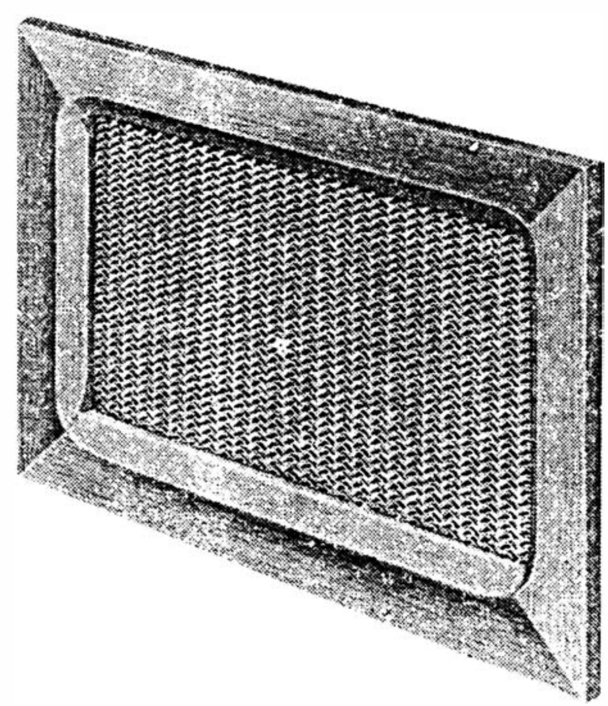
The effect on R_{es} and X_{es} of varying R_f is shown in Fig. 14. The curves have been plotted for two values of X_P, i.e. X_{P1} and X_{P2} corresponding to those shown at f_1 and f_2. It will be seen that the curve R_{es2} reaches a maximum at $R_f = X_P^2$ where its value is $R_f/2$. At this point it will be seen that X_{es2} and R_{es2} are equal and the Q of the circuit under these conditions is therefore 1.

If we now consider a lower value of X_P corresponding to X_{P1} at f_1 we see from the curves that for the value $R_f = X_{P2}$ the Q clearly greater than 1. It is evident from the curves that R_f has a range of values that will produce higher damping at f_1 than at f_2 (and also some values that will produce the opposite effect). The action of the enclosure vector may be summarized by considering the locus of its impedance, which is part of a spiral and is shown in Fig. 15. The presence of R_f will of course alter the actual values of the frequencies f_1 and f_2, but again careful choice of component values enable us to hold f_1 at 20 c/s. We care not for the predicament of f_2.

It was decided that the first prototype enclosure based on these principles should be designed for use in conjunction with the Goodmans Axiom 150 Mk. II loudspeaker. Accordingly the values of R_d, R_c, C_c, M_c and R_a in the analogy were determined from the physical constants of this loudspeaker and translated into acoustical terms. From this the dimensions of the enclosure and vent were determined, and an enclosure was constructed accordingly, the resistance being analogous to a resistive air leak in the enclosure walls. The impedance curves for this enclosure are compared in Fig. 16 with those of the reflex cabinet and a true infinite baffle when housing speakers identical to the above. The evidence is fairly conclusive. The effect of closing the air leak (removing R_f) is also shown.

There are a number of methods of forming the resistive air leak, all of them possessing varying degrees of manufacturing difficulty. One method is to make a number of very narrow slits in one or more of the enclosure walls. Another is to cover a relatively large aperture in an enclosure wall with a material of suitable porosity. In any event the resistance is due to the frictional component of the air leak and one of the principal practical difficulties has been to make this frictional component high relative to the mass component which is present in any aperture. In the analogous circuit this mass component appears as an inductance in series with R_f.

From the foregoing principles formulæ have been derived expressing the various cabinet dimen-



One of a range of acoustical resistance/mass units designed to match Goodmans loudspeakers in cabinets of specified volume.

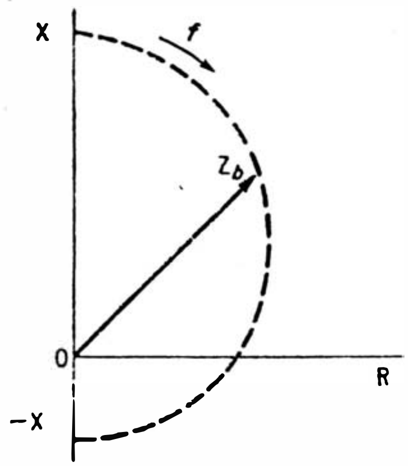


Fig. 15. Locus diagram showing the variation with frequency of the magnitude and direction of the enclosure's impedance vector.

sions in terms of the loudspeaker constant and the desired frequency characteristics. The application of these formulæ, however, demands a complete knowledge of the conditions under which they were being used, otherwise the results can be laughable. In acoustics all sorts of nasty things happen; resistance varies with frequency (but only sometimes) and component values vary with the weather. One is almost tempted to suggest that guesswork would yield as good results.

Fortunately this is not quite true, and in order to simplify the design of enclosures for their various loudspeaker systems Goodmans Industries have worked out the optimum enclosure volume for each system and have designed and marketed for each system a panel containing the acoustic components corresponding to R_f and M in the foregoing analogies. These panels are slightly inaccurately known as acoustic resistance units or ARUs but in fact the required mass component is also included so that all the home constructor needs to do is to make a box of the prescribed internal volume and cut two holes, one for the loudspeaker and one for the appropriate ARU, and having lined the enclosure and screwed these items into place, the enclosure will exhibit all the properties originally stated. The manufacturers have produced this unit, since they feel that in view of the foregoing discussions it is not possible to offer any simple formulæ or design that could be used by persons not familiar with this type of work to produce the required acoustic components with any degree of accuracy.

The performance of Axiom enclosures has been compared with that of other types. Listening tests have shown that the bass radiation is somewhat better than that from the reflex type cabinet at middle bass frequencies and considerably better at the low frequencies, thereby imparting a warm, well-balanced quality to the reproduction. Tests with an oscillator showed that a strong, pure 20-c/s fundamental note could be radiated without excessive cone movement. Transient curves taken showed a very short decay time, characteristic of non-resonant conditions. This is the more interesting when one realizes that the volume of this type of enclosure is about half that of a correctly designed reflex cabinet for the same speaker.

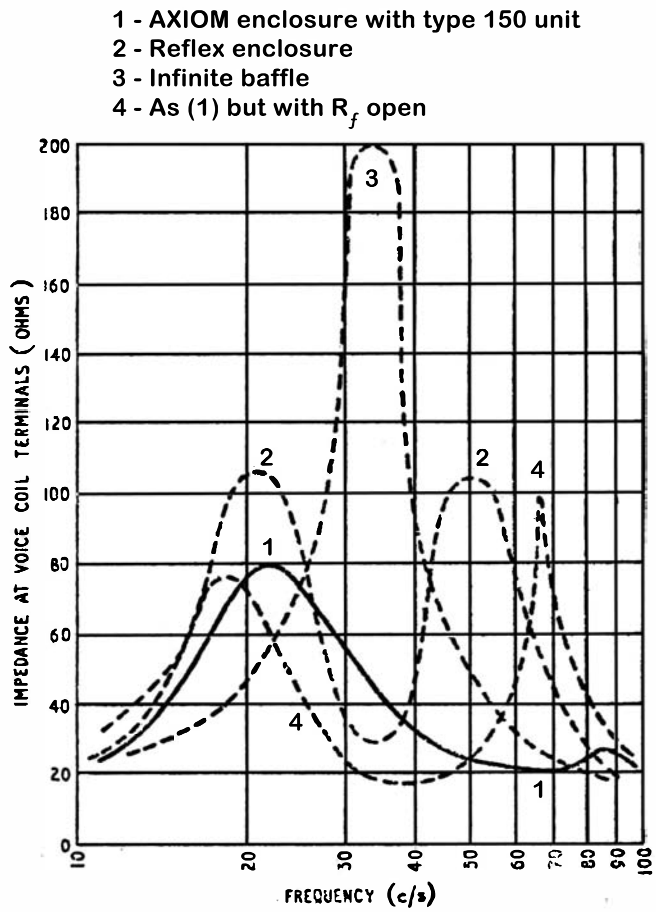


Fig. 16. Voice coil impedance curves of the Axiom 150 unit in an Axiom 172 enclosure, and the same speaker in various conventional mountings.

In addition to the qualities mentioned this type of enclosure has the following advantages:

(1) It is simple and cheap to construct.

(2) The dimensions of the enclosure (corresponding to C in the analogous circuit) are not extremely critical and may be varied up to $\pm$ 10%, if necessary for "styling."

(3) The enclosure can be of any shape and the acoustic resistance unit can be placed in any position relative to the speaker.

(4) The resonant frequency of the loudspeaker is not critical, although, if higher than the value for which the enclosure was designed, the bass extension will be reduced.

Theoretically the bass response of any enclosed loudspeaker will tend to fall, due to the damping applied to the cone reducing the condition of mass control. In the enclosure we have described however, the impedance applied to the cone governs its velocity in a predetermined manner thereby securing a higher efficiency which in practice made bass boosting unnecessary even when used in conjunction with loudspeakers having high electro-magnetic damping.

This enclosure design has been named "Axiom" after the range of high-fidelity loudspeakers manufactured by Goodmans Industries. Patent applications have been made.

Loudspeakers on Damped Pipes*

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First Locanthi's horn analog is shown to be well suited for modeling transmission-line loudspeaker systems. The circuit can accommodate arbitrary flare shapes and allows damping to be included as any combination of series and parallel losses. Four empirical parameters are then developed to simulate the effects of lining or stuffing. Finally three optimized transmission-line geometries are presented, which can be described in terms of generalized alignments using familiar Thiele–Small parameters. Using one of these new alignments, a transmission line can match the frequency response and efficiency of a comparable closed box, but with reduced cone excursion.

0 INTRODUCTION

The Acoustic Labyrinth loudspeaker enclosure was patented by Olney in 1936 and analyzed in a paper published the same year [1]. Olney proposed to correct the defects of simple open-back loudspeaker cabinets by taking a different approach altogether: ". . . the major problem was taken to be the elimination of cavity resonance, and the course pursued was the direct but drastic one of abolishing the troublesome cavity." The loudspeaker was to be mounted at one end of a folded tube lined with a material whose absorption coefficient increased with frequency. In theory this would provide useful summation of front and rear radiation at low frequencies while attenuating higher order resonances.

Versions of the labyrinth were produced by Stromberg-Carlson and others as late as 1950. Interest in the design as a high-quality loudspeaker enclosure was revived by Bailey in 1965, who used fibrous stuffing instead of absorptive lining [2]. It remains a favorite of loudspeaker experimenters and audio enthusiasts. To its devotees, the damped transmission line delivers a neutral, uncolored performance that cannot be matched by vented or closed boxes.

The length and shape of the pipe, the density and kind of material used for damping, and the optimum loudspeaker characteristics have been debated for more than 30 years. However, objective information is rare and is confined mostly to the frequency response measurements of a few successful designs. The goals of this project were to develop a computer analog capable of modeling transmission-line systems, to validate the model by testing a variety of designs, and to develop basic performance relationships similar to the Thiele–Small analysis of vented boxes.

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1 SYMBOLS USED

f_3 = −3-dB corner frequency of low-frequency rolloff
f_P = nominal quarter-wave pipe resonance frequency
f_0 = actual pipe fundamental resonance frequency
f_S = loudspeaker resonance frequency
f_L = frequency of lower impedance peak
f_H = frequency of first upper impedance peak
Bl = loudspeaker force factor, N/A
Q_{MS} = Q of loudspeaker mechanical system
Q_{TS} = total Q of loudspeaker
R_{ES} = dc resistance of voice coil
R_{MS} = resistive component of loudspeaker mechanical system
L_{ES} = inductance of voice coil
L_{MS} = inductive component of loudspeaker mechanical system
C_{MS} = capacitive component of loudspeaker mechanical system
V_{AS} = volume of air having compliance equivalent to loudspeaker cone suspension
V_P = internal volume of pipe (including coupling chamber)
ρ = density of air, = 1.2 kg/m^3
c = velocity of sound, = 344 m/s.

These are familiar symbols, in most cases identical to those used for vented-box analysis. The symbol f_P is a reference frequency based on the physical length of the air path, such as "a nominal 100-Hz pipe." The pipe's actual fundamental resonance f_0 is affected by a number of additional factors, including end correction, pipe geometry, and stuffing material.

2 BASIC ANALOG CIRCUIT MODEL

A damped pipe can be analyzed as a horn with losses. At least three earlier papers describe one-dimensional horn analogs capable of modeling arbitrary flare shapes. In one study [3] the flare is approximated as a series of exponential sections. In another, conical sections are used [4]. A third approach, originated by Locanthi, predates the others and was originally built as an analog transmission-line model using real inductors and capacitors [5].

Locanthi's analog includes the familiar mobility model of the loudspeaker itself, followed by an LC ladder in which series inductors represent air compliance and shunt capacitors represent mass. Each LC section is equivalent to a cylindrical element of specified diameter and length. If the lengths of individual elements are very small in relation to wavelength, then the analog is surprisingly accurate.

Fig. 1 shows the circuit as modified to model transmission-line loudspeaker systems. The model uses 32 sections so that arbitrary flares can be entered element by element in a fairly short time, yet there are enough sections to handle a reasonable bandwidth. The usable upper frequency limit for a 32-element transmission line is roughly 800 divided by its length in meters.

Three modifications were made to Locanthi's circuit. First, shunt resistances were added to model damping losses. These are shown as variable resistors because damping may be frequency dependent. Series resistances could be included to represent leakage losses, but their effect is negligible.

Second, since the loudspeaker may not be mounted at the end of the pipe but at some intermediate location, an optional 16-element closed stub was inserted at the pipe throat.

Third, a transformer between the loudspeaker and the horn throat has been omitted. The transformer was there to explicitly match the throat area to a larger or smaller driver cone area. Its effect can be duplicated by scaling the circuit values for eitherthe horn or the loudspeaker. If there is a coupling chamber between the cone and the pipe throat, it can be represented by an additional series inductor. In practice, the coupling chamber compliance is simply included in the value of L_1.

The easiest way to calculate loudspeaker circuit values is first to divide R_{ES} and L_{ES} by $(Bl)^2$. Then C_{MS} in farads is numerically equal to the moving mass in kilograms, and L_{MS} can be derived from f_S. With this information plus Q_{MS}, the value of R_{MS} can be found.

Calculating transmission-line values is not much more complicated. Let

s_d = driver cone area, m^2
S_0 = throat area, m^2
S_n = section area, m^2
K_s = S_n/S_0
x = section length, m.

Then,

$$L_n = \frac{K_s x}{\rho c^2 S_0}$$

$$C_n = \frac{\rho S_0 x}{K_s}.$$

If necessary, these values can then be impedance scaled to the ratio $(S_0/S_d)^2$. Detailed information about the derivation of the analog and the calculation of acoustic loads can be found in Locanthi [5].

Following Thiele–Small loudspeaker system analysis we assume that radiation loading is negligible in the (bass) frequency range of interest. In this range, however, acoustic loads for cone and pipe radiation can be accurately included as simple RC shunts if desired [6].

As with a vented-box analog, the net system output is equivalent to the complex sum (including phase reversal) of cone output and pipe output. In most practical transmission-line systems any effects of mutual coupling between the two are small enough to be ignored. The effects of pipe bends and folds are also ignored. In a typical transmission line, they can be expected to occur above the frequency range of useful pipe output.

The ability of this circuit to model a fairly compli-

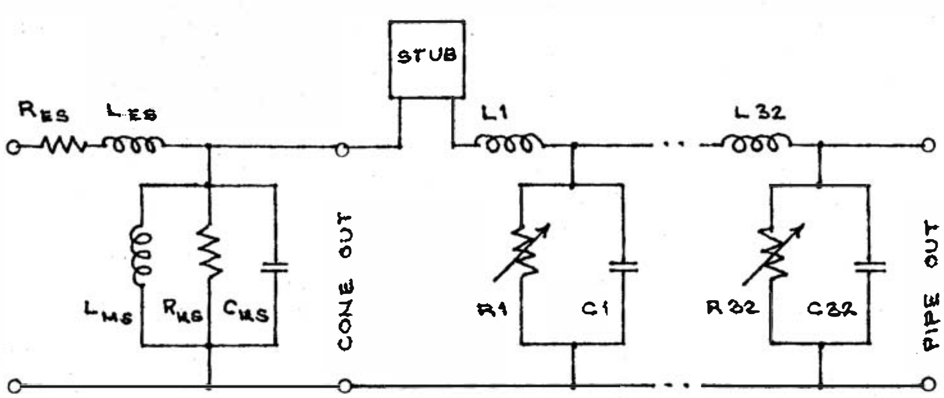


Fig. 1. Tranmission-line analog circuit.

cated, undamped transmission line is impressive. One of the test systems consists of a small loudspeaker mounted 0.14 m from the end of a 0.71-m tapered pipe. The measured cone output and pipe output are graphed in Fig. 2(a), and the corresponding analog circuit curves in Fig. 2(b).

Locanthi's horn analog has not received much attention, presumably because it is more complicated and less accurate than alternative computer models. However, a simple *RLC* ladder is easy to set up with any circuit modeling program and calculations are very fast. Moreover, using an electrical transmission line to model an acoustical transmission line seems particularly appropriate.

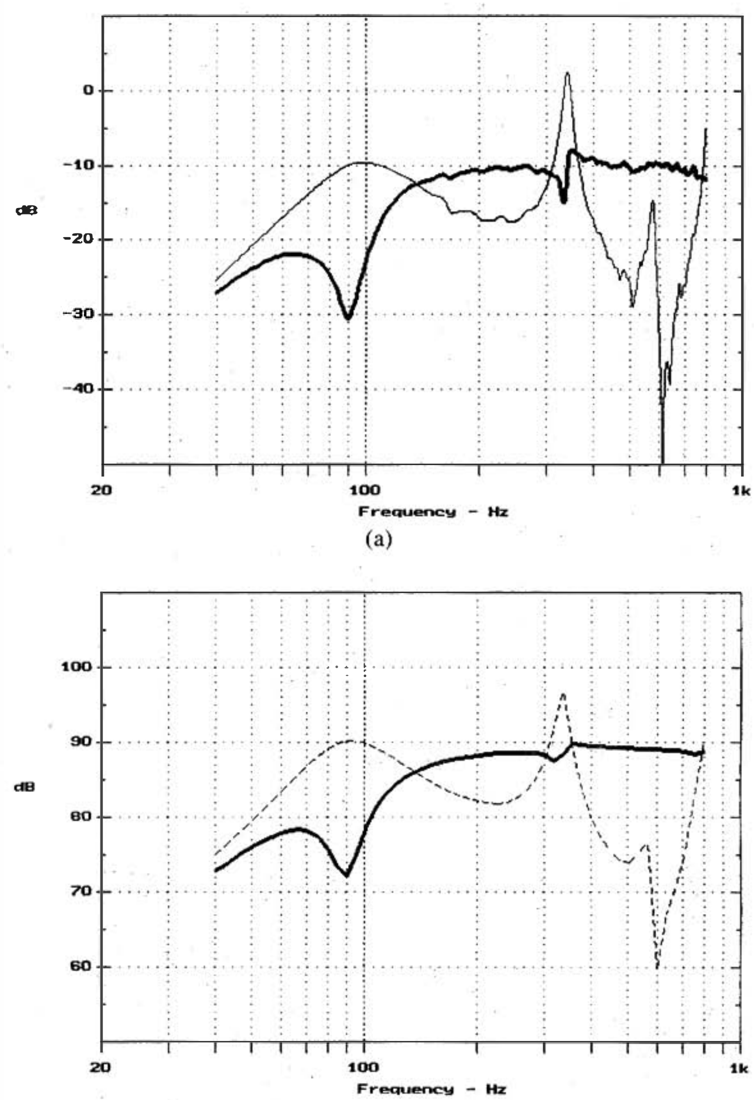
3 TEST PROCEDURE

The simplest transmission line is a straight pipe with a loudspeaker on one end. To check the accuracy of the analog and to study the behavior of damping materials, a number of these were built and tested. Four were cylindrical pipes varying in length from 0.6 to 1.8 m. The fifth was a reversible rectangular pipe with two slanted sides: a parabolic horn. Additional variants were built as the project went along.

To make response measurements, a given pipe was set horizontally on a trestle about 1 m above the floor. A calibrated Bruel & Kjaer 4134 microphone was connected to a TEF20 analyzer. Sweeps were run from 20 Hz to 1 kHz with a frequency resolution of 10 Hz, giving accurate readings down to about 25 Hz. Impedance curves were also run. Each test was saved to disk. The TEF system stores all measurements as sets of complex data points, preserving both magnitude and phase.

Frequency response measurements were made using near-field microphone placement [7] so that loudspeaker and pipe outputs could be measured separately with negligible flanking effects. By blocking the end of the pipe it was possible to measure leakage from the loudspeaker at the other end. Crosstalk in the 0.6-m pipe was about −25 dB. It was down more than 30 dB in the longer pipes.

Postprocessing allows the system response to be calculated as the complex sum of loudspeaker and pipe outputs. However, when a microphone is located very



(b)

Fig. 2. (a) Measured response of undamped test system. (b) Response of analog circuit model. Cone output (bold) and pipe output.

near a small sound source, a movement of only 1 or 2 mm can shift the measured sound level by more than 1 dB. A number of preliminary runs were made to be sure that the test setup delivered repeatable results. To verify that the cone and pipe data could indeed be summed, several response measurements were made with the microphone equidistant from both loudspeaker and pipe, at one apex of an equilateral triangle.

It was later learned that this test setup closely parallels that of Letts' 1975 study [8]. To the extent that the tests overlap, results are in close agreement.

4 BEHAVIOR OF DAMPING MATERIALS

The available technical literature includes a great deal of information about the acoustical performance of absorptive materials. However, Bradbury's 1976 paper [9] is one of the few relating directly to transmission-line design. His study postulates that fibers are set in motion as sound waves pass through the line. Aerodynamic drag analysis is then used to predict resistive and reactive effects from fiber size, mass, and packing density. The concept was later expanded by Leach and applied to closed-box loudspeaker systems [10].

In the 1980s Bullock developed a transmission-line computer simulation based on Bradbury's model, but comparisons with measured performance proved to be ". . . not satisfactory" [11]. The conclusions put forth in Bradbury's paper are also at odds with some of the test results to be described here. One reason may be that his formula for computing the drag coefficient was admittedly tentative. Also, it is not certain that fiber motion is really that important. For example, Hersh and Walker [12] reported excellent predictions of measured behavior, yet their analysis makes the simplifying assumption that fibers are stationary.

For practical loudspeaker system design, our concern is not with the composition of the damping material but rather its actual performance. Moreover, we are only interested in the low-frequency response of a limited range of pipe lengths. On that basis, tests were made with various kinds of lining and stuffing.

Most of the tests used varying densities of four stuffing materials:

1) Ordinary fiberglass thermal blanket. This is readily available with paper backing, which can be removed.

2) Polyester fiber stuffing, "Poly Fluff," a product of Western Synthetic Fiber Inc., Carson, CA.

3) Microfiber stuffing, Celanese "Microfill."

4) "Acousta-Stuf." This is a Nylon polyamide fiber sold for use in loudspeaker enclosures. It is available from Mahogany Sound, Box 9044, Mobile, AL 36691-0044.

These materials are easy to buy, easy to use, and perform well for this application. Numerous other substances were tested, including cotton puffs, steel wool, packing pellets, and plastic foam.

It seems prudent to focus on inert, nonorganic materials. However, long-fiber wool was chosen by Bailey as the ideal stuffing for transmission lines, and his preference was supported by Bradbury. Unfortunately bulk wool is not easy to find in the United States, so fluffy wool yarn was tested instead. It displayed no unusual properties, performing roughly the same as Acousta-Stuf, which is advertised as a superior substitute for wool. A similar comparison was observed between cotton puffs and microfiber.

Microfiber is light and fluffy. Acousta-Stuf is ropy and fairly heavy. For roughly equivalent damping, the packing densities of these two materials must differ by a factor of 2 or more. When this is taken into account, the behavior of all four materials is similar.

Even so, there are some differences in attenuation characteristics. For example, at higher packing densities fiberglass displays a somewhat sharper knee and more rapid high-frequency attenuation than the other materials. On the other hand, it seems to be more prone to unexpected response irregularities at low densities.

Fig. 3 compares the measured pipe output of a test system stuffed with 8 g/L of fiberglass and 16 g/L of Acousta-Stuf. The Acousta-Stuf curve is nicely rounded. In comparison, the fiberglass has a sag around

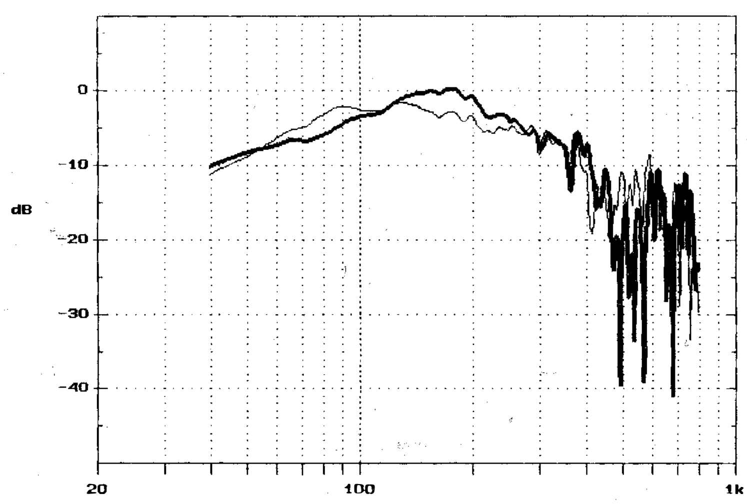


Fig. 3. Pipe output. 8-g fiberglass (bold) and 16-g Acousta-Stuf.

80 Hz, and a broad bump centered at 180 Hz. The remaining peaks and dips are characteristic of the test setup.

This is a typical example. The pipe output is always a little lumpy, and different materials have their own characteristic signatures. In transmission-line systems, if the pipe output is appreciable, then these small differences may be audible.

Most transmission-line literature recommends some optimum stuffing density regardless of pipe length. Common sense suggests that a 100-Hz short pipe should have the same stuffing density as a 50-Hz pipe twice as long. In reality, test results clearly demonstrate that the shorter pipe requires greater packing density for equivalent performance. Moreover, this is apparent from an examination of the analog circuit.

Consider a single transmission-line section. The values of L and C are proportional to the section length x. If the length of the pipe is doubled without changing its cross-sectional area, then L and C must also double. Viewing the section as a low-pass filter, its cutoff frequency has shifted down one octave with no change in impedance. Therefore, since R remains constant but x has doubled, damping per unit length must be halved.

A few tests were run using lining instead of stuffing. However, it became obvious that in pipes of small diameter, even highly absorptive lining cannot provide enough midfrequency attenuation to control passband ripple. Moreover, predicting the performance of lining involves the cross-sectional area and perimeter as well as the length, adding unwanted complications to a basic computer model.

5 MODELING DAMPING MATERIALS

The resistive component of damping is represented by shunt resistance. A frequency-related resistance is required even though fixed losses produce greater attenuation at higher frequencies. Fig. 4 shows analog circuit pipe output with fixed relative damping ranging from 0 to 10. The 1-m pipe is driven by a constant-velocity piston and terminated in its characteristic impedance. Typical absorptive materials exhibit somewhat steeper slopes, and a reactive component is also present.

It is well established that sound wave propagation through tangled fibers is slower than in free air, and is roughly proportional to some power of frequency. In a lightly damped pipe this shows up as a lowering of the nominal quarter-wave resonance frequency plus a smaller shift of the upper harmonics. In a nonresonant transmission-line system, however, damping has a much greater effect on low-frequency performance than propagation velocity. If velocity is set at a fixed value determined by passband ripple frequencies, then any remaining errors mostly affect the response below cutoff.

Four empirical parameters seem to be sufficient to model typical stuffing materials:

1) Fixed losses
2) Variable losses, corner frequency
3) Variable losses, slope
4) Relative sound speed.

The first three are used to calculate the values of shunt resistors at each frequency to be plotted. The last simply sets a scaling factor for capacitance values.

It can be argued that these are unscientific twiddle factors, but they enable the analog circuit described to deliver good approximations of transmission-line behavior. As an example, Fig. 5 shows the system of Fig. 2(b) with the addition of moderate damping. In this case, measured response curves have been omitted because they essentially duplicate the analog response.

6 BASIC SYSTEM BEHAVIOR

With no stuffing, a pipe resonates at odd multiples of its fundamental quarter-wave resonance. The loudspeaker cone is heavily loaded at these frequencies so that loudspeaker output is attenuated and pipe output is accentuated. To complicate the picture, the two are alternately in and out of phase at even multiples of the fundamental, resulting in a highly irregular system response.

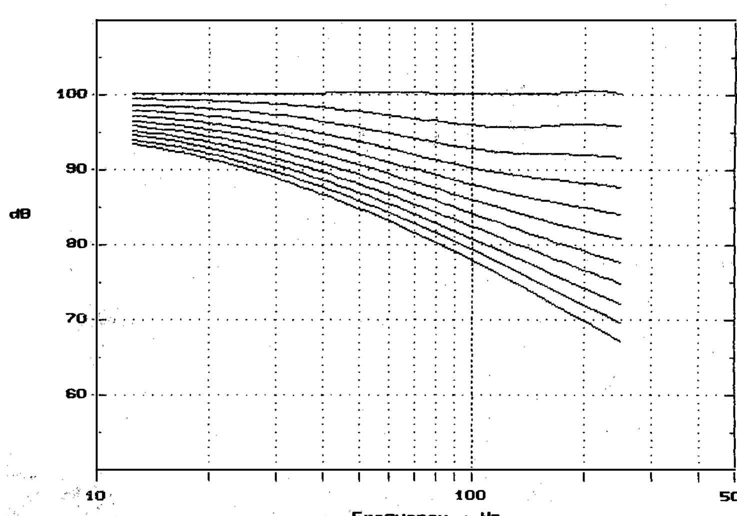


Fig. 4. Terminated pipe output for fixed damping. Relative damping from 0 (top) to 10.

This is clearly shown in Fig. 6, which is the analog response of a small automotive loudspeaker on a 0.78-m pipe. This is nominally a 109-Hz pipe, but it actually resonates at 100 Hz, which is also the loudspeaker's cone resonance. The light solid line represents cone output, the dashed line is pipe output, and the heavy solid line is the combined system response. Note that the upper resonances and antiresonances fall at exact 100-Hz intervals.

The dotted line at the bottom shows voice-coil impedance relative to dc resistance, plotted logarithmically. The impedance curve of this undamped transmission line is obviously similar to that of a vented box. The impedance minimum at 100 Hz is flanked by two peaks at about 64 Hz and 150 Hz. Additional peaks at higher frequencies will disappear as damping is added.

Fig. 7 illustrates what happens when the pipe is stuffed with light, medium, and heavy damping. Like Fig. 6, these are computer analog response curves, but they are all confirmed by actual test data.

When a small amount of damping is introduced, cone and pipe outputs still show resonances, and pipe attenuation is minimal [Fig. 7(a)]. The cone output suggests that quarter-wave loading has moved down to about 80 Hz. However, the fundamental resonance has all but disappeared from the impedance curve. The lower impedance peak no longer exists, and f_H has become a gentle bump. The cone and pipe outputs are additive down to about 85 Hz, and the low-frequency slope is reduced from 24 dB to about 18 dB per octave.

Moderate stuffing density, as shown in Fig. 7(b), results in a well-behaved transmission-line system with a sag of perhaps 2 dB around 300 Hz and gentle rolloff below 150 Hz. Below 100 Hz the slope is about 12 dB per octave. Although pipe output is well below cone output, the two are additive over more than two octaves. The only identifiable resonance in the impedance curve is f_H. The system is starting to behave very much like a closed box.

Still more stuffing results in a purist's transmission line, which effectively swallows up back radiation through the passband, as in Fig. 7(c). Going beyond this point is self-defeating since the output of the loudspeaker cone is progressively reduced by excessive damping.

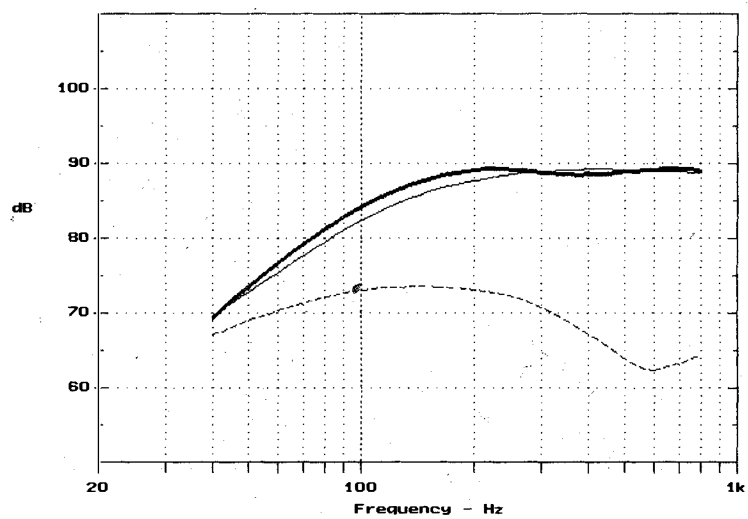


Fig. 5. System of Fig. 2(b) with medium-density stuffing.

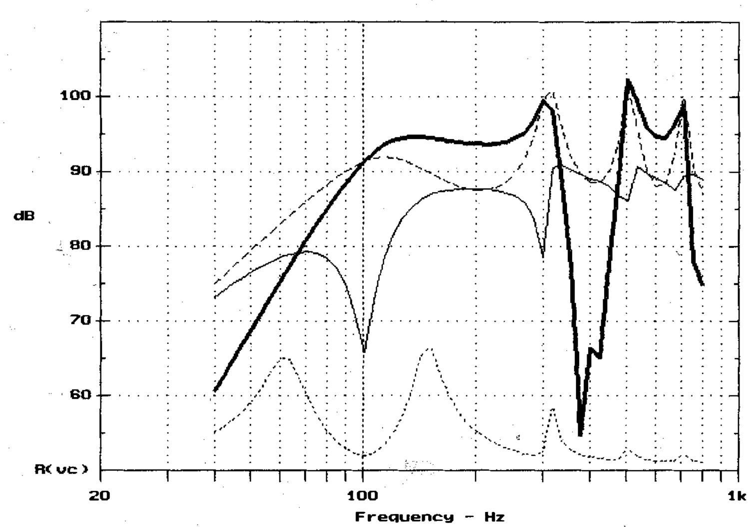


Fig. 6. Response of loudspeaker on undamped straight pipe. Impedance (bottom), cone output, pipe output, and system response (bold).

7 TRANSMISSION-LINE DESIGN FUNDAMENTALS

For highest possible efficiency with minimal passband ripple, it is apparent that damping should be negligible up through the second harmonic yet provide more than 20 dB of attenuation at the fourth harmonic. None of the materials tested even comes close. Rather than searching for a new kind of damping material, it seems more reasonable to look at the behavior of Fig. 7(b) and

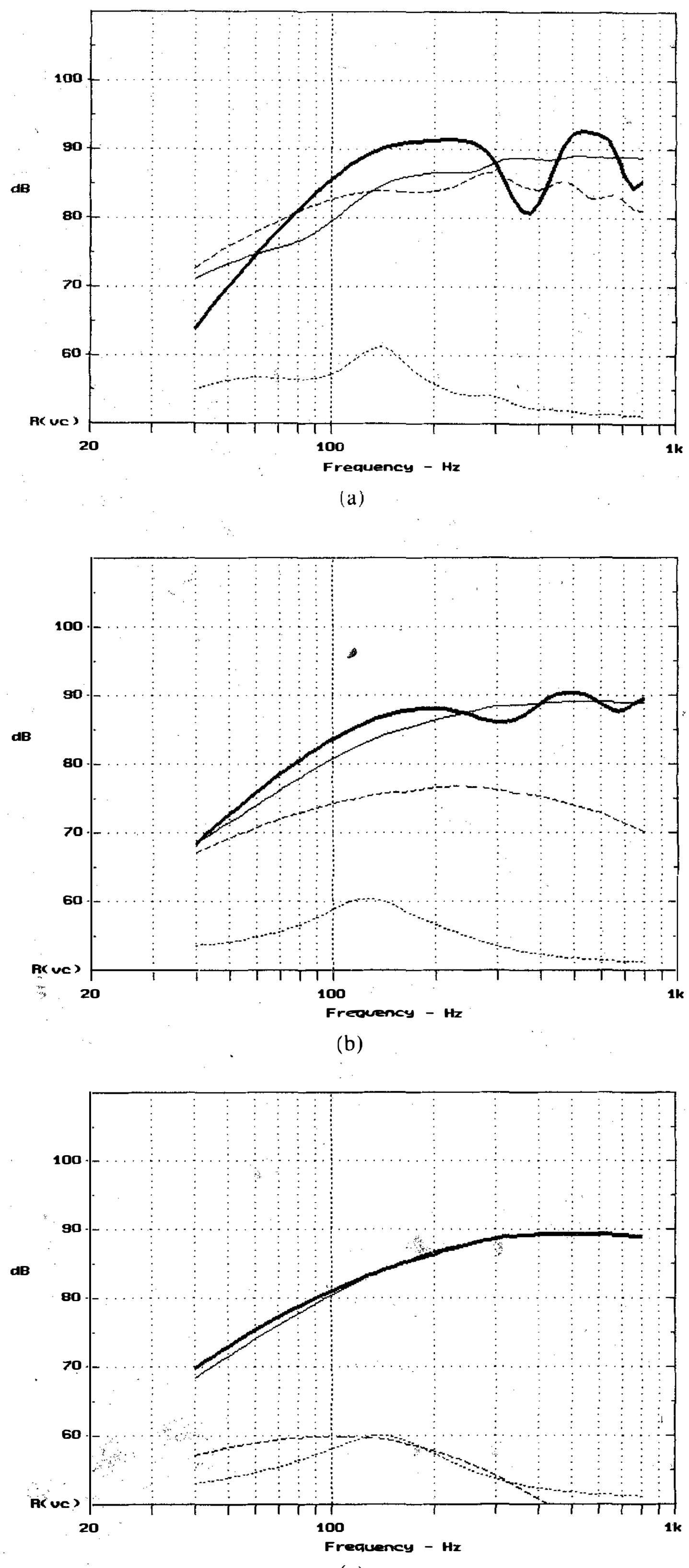


Fig. 7. Response of loudspeaker on straight pipe. (a) Light damping. (b) Moderate damping. (c) Heavy damping. Impedance, cone output, pipe output, and system response (bold).

see what changes might be made to flatten the response.

First Q_{TS} could be increased to lessen midrange sensitivity and thus level the response above 150 Hz. Also, as in a closed-box system, it appears that f_S should be substantially lower than f_3. Finally, to keep passband ripple within ±1 dB, the stuffing density must be increased slightly.

With some trial-and-error tweaking of the loudspeaker parameters, the response of Fig. 8(a) was achieved. Now f_3 matches f_P while f_S is an octave lower. Q_{TS} is about 0.5 and the pipe volume is one-half V_{AS}. Apart from selecting the right stuffing density, this is sufficient information to duplicate the response curve for any desired low-frequency cutoff. Notice that the pipe diameter is determined by the pipe length and V_P. The cone diameter per se is not a factor in low-frequency enclosure design, as Small proved more than 25 years ago [13].

In Fig. 8(a) the pipe output and cone output add constructively down to 40 Hz or lower. Therefore it might be possible to set f_3 as much as an octave below f_P by specifying the proper loudspeaker parameters, with no change in stuffing density. With the benefit of hindsight, this is a logical assumption. It is confirmed by computer modeling and test results. Fig. 8(b) shows how a nominal 109-Hz pipe can be "tuned" to 65 Hz. Efficiency goes down as well, just as one would expect from Thiele–Small analysis of box-type systems.

Some experimenters have reported a miraculous extension of the low-frequency response by using high-density stuffing in very short pipes. This is wishful thinking. In the real world, as the packing density is increased beyond its optimum value the system behaves more and more like an overdamped infinite pipe. For maximum efficiency it appears that f_3/f_P should be between 0.7 and 1.4.

The following simple alignment table summarizes the Thiele–Small relationships of Fig. 8:

	$\frac{f_3}{f_P}$	$\frac{f_S}{f_P}$	$\frac{V_{AS}}{V_P}$	Q_{TS}
Fig. 8(a)	1.0	0.50	2.0	0.46
Fig. 8(b)	0.6	0.33	1.0	0.36

For a nominal 100-Hz pipe the alignments shown can be realized with 32-g/L polyester stuffing. However,

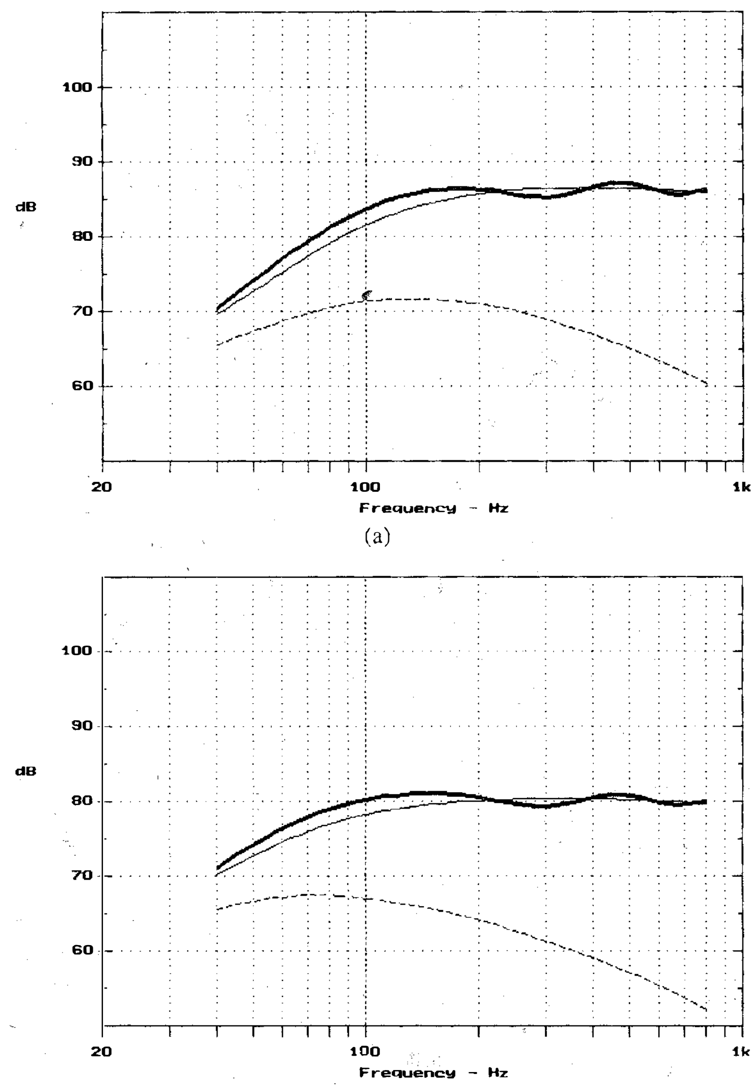


Fig. 8. (a) Response of straight pipe with improved alignment. (b) Response of extended low-frequency alignment. Cone output, pipe output, and system response (bold).

they can also serve as multipurpose alignments. The classic low-frequency rolloff of Fig. 7(b) can be approximated by halving Q_{TS}. To simulate an infinite pipe, the stuffing density should be increased by about 50%.

These examples show what can be done with traditional transmission-line design, but they are lossy. Efficiency is 2–5 dB less than for a comparable closed box. Fortunately the situation can be improved by considering something other than a simple straight pipe.

8 ALTERNATE GEOMETRIES

The computer analog made it easy to experiment with all sorts of geometrical modifications, including tuned stubs, abrupt discontinuities, tapered pipes, coupling chambers, and tricks with damping placement. Promising designs were built and tested. Five of these managed to deliver greater efficiency without sacrificing the traditional transmission-line performance. Fig. 9 shows these variant geometries:

- Tapering the pipe (reverse flare) lowers the fundamental resonance frequency without affecting the upper harmonics. The frequency range of constructive pipe output is broadened [Fig. 9(a)].
- Constricting the pipe exit (a vented pipe) has a similar effect [Fig. 9(b)].
- A coupling chamber lowers the fundamental resonance and increases the high-frequency attenuation of damping materials [Fig. 9(c)].
- An abrupt change in pipe diameter at one-third its length produces a reflection that offsets cancellation in the troublesome fourth-harmonic region [Fig. 9(d)].
- Mounting the loudspeaker at one-fifth the length of the pipe is even more effective in attenuating the pipe output near the fourth harmonic [Fig. 9(e)].

Of these, the tapered pipe, coupling chamber, and offset loudspeaker were chosen for additional analysis.

8.1 Tapered Pipe

Tapered tranmission lines go back to Bailey's design [2]. The reasoning seems to be that since energy decreases along the length of the pipe, space can be saved without making any difference in performance. In fact, tapering makes a big difference.

Tapering an undamped pipe can lower f_0 by more than one-third octave. Upper harmonics are almost unchanged. In a transmission-line loudspeaker system f_3 shifts down, giving a useful extension of the low-frequency bandwidth.

An area reduction between 1:3 and 1:4 seems to work best. If the pipe throat is too large, cross modes can be a problem. If the mouth is too small, excessive air turbulence may result. The taper can be linear or conic, or approximated by cylindrical sections. These variants influence the pipe output, but not enough to appreciably affect the overall system response. Fig. 10 shows the performance of an optimized system, normalized to a low-frequency cutoff of 100 Hz.

8.2 Pipe with Coupling Chamber

Coupling chambers have also been used in many transmission lines. The idea seems to have evolved empirically. Technical explanations range from better impedance matching to suppression of pipe resonances. The latter is close to the truth.

The loudspeaker cone is coupled to the pipe throat by the springiness of the air in the chamber. At mid and high frequencies the throat impedance is largely resistive, and the resulting low-pass action adds another 6 dB per octave of high-frequency rolloff. This can easily be seen by comparing Figs. 10 and 11. Both systems are 9-L nominal 125-Hz pipes stuffed with the same packing density. The coupling chamber is also stuffed. In Fig. 11 the coupling chamber accounts for 3 L and a slimmer pipe contains the remaining 6 L.

The system response of Fig. 11 closely matches that of Fig. 10. Above 200 Hz, however, the pipe output rolls off more rapidly and passband ripple is reduced even though the ripple frequencies have moved down. Also, in the 100-Hz region the cone excursion is slightly less.

For the system to function as modeled there must be a clear demarcation between pipe and coupling chamber. On the other hand, if the chamber is too large, then we have restored the cavity that the transmission line was supposed to eliminate. A good compromise is to make the chamber volume one-third of the total volume.

8.3 Offset Loudspeaker

Quarter-wave stubs are sometimes used in duct silencers to suppress specific frequencies. In a damped transmission line the effect is more akin to a shelving filter. Fig. 12 shows how this geometry can be used to

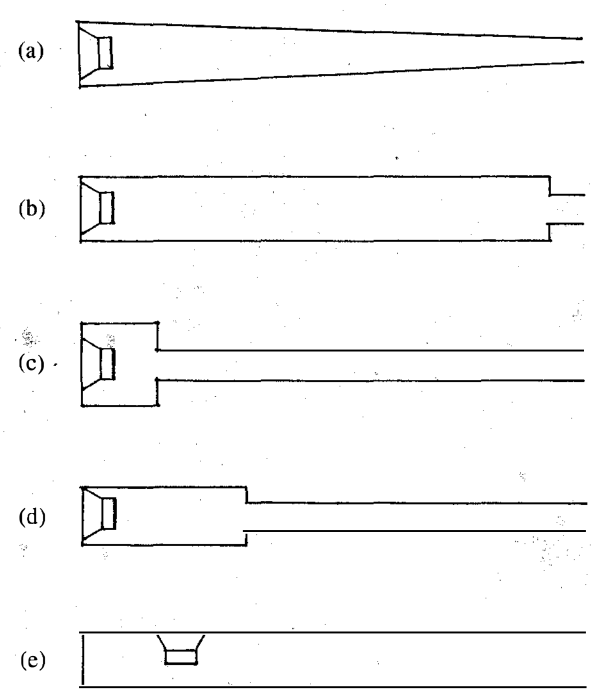


Fig. 9. Alternate pipe geometries. (a) Tapered. (b) Vented. (c) Chamber. (d) Stepped. (e) Offset loudspeaker.

achieve performance at least as good as in the previous examples. A straight pipe is used with the loudspeaker located at one-fifth the length of the pipe.

This system has the same volume, the same stuffing density, and the same low-frequency cutoff as the previous two examples. However, f_3 is now higher than f_P. A longer, thinner pipe is required for comparable performance.

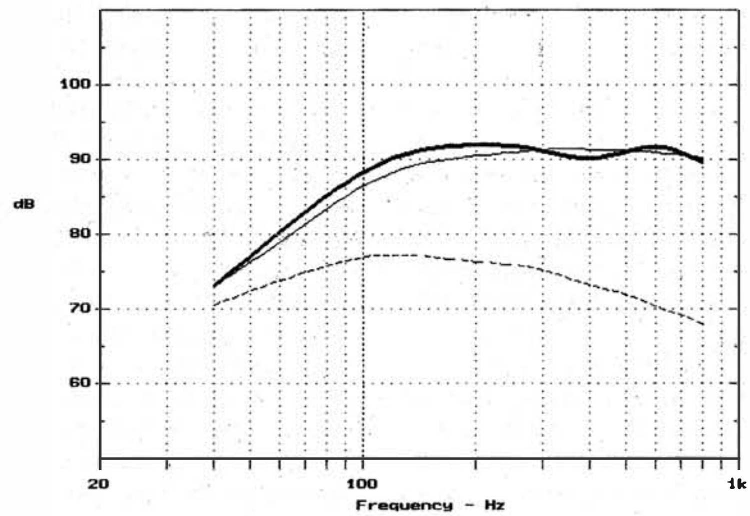


Fig. 10. Tapered pipe transmission-line response. Cone output, pipe output, and system response (bold).

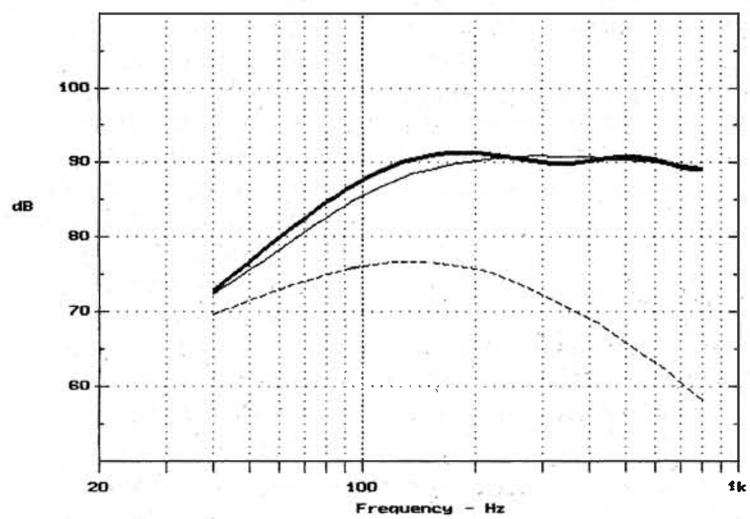


Fig. 11. Coupling chamber transmission-line response. Cone output, pipe output, and system response (bold).

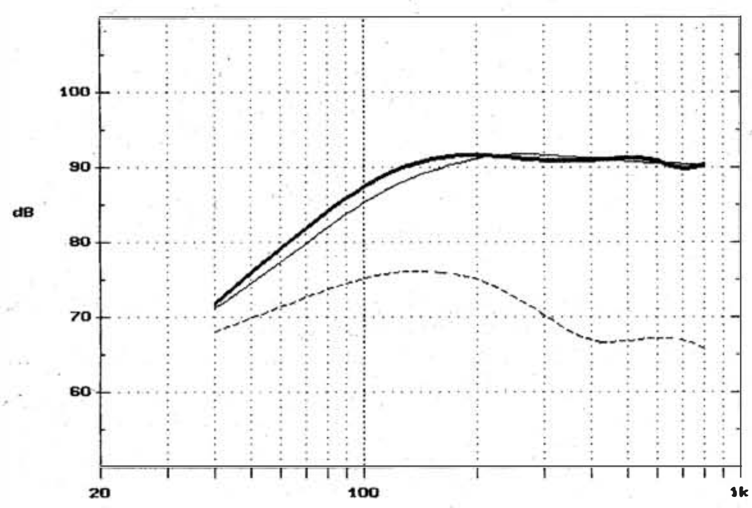


Fig. 12. Offset loudspeaker transmission-line response. Cone output, pipe output, and system response (bold).

8.4 Alignment Table

Alignments based on the Thiele–Small parameters are listed in Table 1 for all three systems. Three sets of values for each design offer a reasonable spread of loudspeaker choices. In reality, all three alternates represent the same loudspeaker mechanism with different cone suspension compliances.

8.5 Combinations

It is possible to combine various pipe geometries. As a case in point, a coupling chamber can drive a tapered pipe. Although it has been used in at least two commercial transmission-line designs, this combination provides no reduction in pipe volume and slightly degrades the low-frequency performance.

Other combinations are similarly disappointing. For example, a tapered pipe with an offset loudspeaker can be made to squeeze out another decibel of efficiency, but at the cost of greater cone excursion. The net result is a decrease in the maximum low-frequency output.

8.6 General Comments

The optimized transmission lines described are characterized by second-order low-frequency rolloff with minimal passband ripple. The efficiency can match that of an equivalent closed-box system, however the pipe output contributes 2–3 dB in the low-frequency region. Since loudspeakers are displacement limited at low frequencies, the net result is a corresponding increase in maximum output.

9 STUFFING SPECIFICATIONS

In theory, separate system alignments would be required for different stuffing materials, different packing densities, and different pipe lengths. However, the general trend for all materials is an increase in sound attenuation with an increase in frequency. As previously noted, if appropriate packing densities are chosen for the four materials studied, then their damping characteristics are similar over a moderate range of frequencies.

There is little concern about the pipe output at frequencies well below f_3 because this region is out of the passband. For acceptable passband ripple, the pipe output must be at least 15 dB below the cone output at frequencies well above f_3 and will continue to drop at higher frequencies. It follows that for a specified cutoff frequency, the damping characteristics must be matched only over a range of two octaves or less. However, densities for pipes of various lengths must be specified separately.

Table 2 is a cross reference chart to be used in combination with Table 1. It shows the equivalent densities of four stuffing materials over a range of useful pipe lengths. The information is derived from several sets of measurements for each material and should be reasonably accurate for fiberglass, Acousta-Stuf, and polyester. Fewer tests were made with microfiber and a fair amount of interpolation is included. Also, tests made with very low packing densities show appreciable variations.

Which material is best? Each material has its own damping characteristics, and even with close matching the differences may be audible. On this basis the choice is arbitrary, but there are other factors to consider. Probably the most important is consistency.

Polyester pillow stuffing seems to be fairly generic, but there is no guarantee that a batch from another manufacturer will be the same as Poly Fluff. Fiberglass thermal blanket delivers consistent performance at packing densities greater than 15 g/L. Its unpacked density is about 10 g/L (0.6 lb/ft^3), and its physical properties are held to close tolerances. On the debit side, it is nasty stuff to work with.

Table 1. Optimized alignments for three practical systems.

Design		f_3/f_S	f_3/f_P	f_S/f_P	V_{AS}/V_P	Q_{TS}
Tapered	I	2.0	0.8	0.40	3.10	0.36
(nom. 4:1)	II	1.6	0.8	0.50	2.00	0.46
	III	1.3	0.8	0.63	1.20	0.58
Coupling chamber	I	2.0	0.8	0.40	2.14	0.31
	II	1.6	0.8	0.50	1.35	0.39
	III	1.3	0.8	0.63	0.84	0.50
Offset loudspeaker	I	2.0	1.2	0.60	3.10	0.36
	II	1.6	1.2	0.74	2.00	0.46
	III	1.3	1.2	0.94	1.20	0.58

Table 2. Packing densities (g/L) for various pipe lengths for tapered, offset, and coupling chamber alignments.

Length (m)	f_P (Hz)	Acousta-Stuf	Polyester	Fiberglass	Microfiber
0.61	140	27.0	29.0	14.5	10.5
0.91	94	21.0	22.5	11.0	9.0
1.22	71	16.0	17.5	9.5	7.5
1.83	48	12.0	13.5	—	5.5
2.44	36	8.0	10.5	—	4.3

Acousta-Stuf is more expensive than fiberglass or polyester but its characteristics are closely specified. As delivered, it is lumpy and must be thoroughly teased, especially for low packing densities. Otherwise, it is easy to use and delivers predictable results.

Microfiber is very light and fluffy. Once packed to the desired density it seems to stay in place, but loose wisps will drift around for days. If the brand name Celanese Microfill is used, then its acoustical properties should be as predictable as those of fiberglass or Acousta-Stuf.

All of these materials can be tricky to use in long, large pipes requiring low packing densities. Partitioning a fat pipe into two or more thin pipes will help keep the stuffing in place and make the structure more rigid. Using thick lining instead of stuffing is another alternative, but is outside the scope of this study.

10 DIRECTIONAL EFFECTS

Letts [8] seems to be the only researcher to have noticed the unusual directional properties of transmission lines.

If the dimensions of a sound radiator are very small in comparison with the wavelength, then it is assumed to behave like a point source. Its coverage pattern is omnidirectional, constrained only by large adjacent surfaces. A small sealed or vented loudspeaker system is essentially omnidirectional at frequencies below 200 Hz or so.

In contrast, a loudspeaker on a lightly damped straight pipe is a unidirectional gradient source at low frequencies. Its coverage pattern is the same as that of a cardioid microphone. If the pipe output is less than the cone output, the directional effects are less pronounced but still in evidence. The pipe output must be at least 15 dB below the cone output for the directivity to be determined by the loudspeaker alone.

All of the system response curves in this paper are on-axis curves, that is, they represent a response at some point equidistant from the loudspeaker and the pipe mouth. If the pipe output is appreciable, then the off-axis response and the total power response may both be quite different from the on-axis curve. Such differences are minimized if the loudspeaker and the pipe mouth are very close together.

Since the low-frequency tonal balance heard in a typical listening room is dominated by generally reflected sound, it follows that a loudspeaker on a straight pipe may indeed sound different than one on an otherwise identical folded pipe.

11 CONCLUSION

This study has attempted to demystify the nonresonant transmission-line design pioneered by Bailey: a loudspeaker is mounted on a pipe stuffed with tangled fibrous material of uniform density, providing sufficient damping to control the passband ripple yet allow useful reinforcement of the cone output at low frequencies.

With a few small modifications, Locanthi's horn analog was shown to be an excellent tool for modeling transmission-line loudspeaker systems. However, to derive usable parameters for real-world damping material it was necessary to test a number of pipes with different materials of varying densities.

Based on test results, four empirical parameters were found sufficient to approximate the performance of damped transmission lines. Three of these define a frequency-dependent resistive component. Surprisingly, the relative propagation velocity can then be set to a constant value even though, in reality, it is also frequency dependent.

For a pipe of given length, different materials require different packing densities to achieve desired damping. Once this is done, the passband performance is essentially the same for any of the materials tested.

The pipe length establishes a usable range of cutoff frequencies, typically a one-octave band centered at f_P. Within that range, f_3 is controlled by the loudspeaker parameters in relation to pipe length and volume. Damping remains unchanged.

In contrast to a basic cylindrical pipe, at least four other geometries allow lighter damping, which results in higher efficiency. Systems can be scaled to any cutoff frequency and any practical efficiency by using simple alignment tables. Optimized alignments were developed for three alternate geometries. Allowing for ±1-dB passband ripple, these new alignments approximate the response of an equal-volume closed box, but with reduced cone excursion and correspondingly greater maximum low-frequency output.

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Application of Electric Circuit Analogies to Loudspeaker Design Problems*

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Electric circuit analogies are derived for three types of loudspeaker systems: direct radiator in an infinite baffle, direct radiator in a reflex enclosure, and horn loudspeaker. The data are in good agreement with data taken from experimental ascoustica units.

Editor's Note: It is with pleasure that we publish this 1952 tutorial paper from the *IRE Transactions on Audio*. Mr. Locanthi has simply substituted the new I.S. units for the old c.g.s. system.

INTRODUCTION: Electrical engineers and physicists are frequently concerned with the task of obtaining solutions to electromechanical problems. These people are generally well versed in electric circuit theory. By transforming all of the mechanical "constants" of an electromechanical system to their equivalent electrical quantities, an electric circuit analogy is developed from which the qualitative performance may be quickly judged. Furthermore, quantitative data may be obtained by making appropriate measurements in the electric circuit.

It will be demonstrated in the paper that electric circuit analogy data are in good agreement with data taken from the following experimental acoustical units: 1) direct radiator in an infinite baffle, 2) direct radiator in a reflex enclosure, and 3) horn loudspeaker.

Especially in view of the good agreement between the electric circuit analog data and those obtained from an experimental horn loudspeaker, which was treated partly as a lumped parameter system and partly as a distributed parameter system, there can be little doubt as to the power of this type of analysis.

Electric circuit analogies are derived in this paper for the three types of loudspeaker systems described. The analog computer at the California Institute of Technology was used to obtain data from the electric circuit analogies.

Electric circuit analogies for electromechanical systems have been known for at least 50 years [1]. It is the purpose of this paper to demonstrate the application of electric circuit analogies to certain loudspeaker design problems. In particular, the direct radiator in an infinite baffle, the direct radiator in a "reflex" type enclosure, and the horn loudspeaker will be discussed. Throughout the discussion, the mobility analogy will be used (i.e., voltage represents velocity and current represents force). The only major assumptions that will be made to facilitate the analysis are 1) the cone or diaphragm has no resonant modes and moves as a uniform piston, and 2) the resistance losses in the suspension of the moving system are negligible compared to the radiation losses.

CASE I: DIRECT RADIATOR IN AN INFINITE BAFFLE

A diagram of a direct radiator in an infinite baffle is shown in Fig. 1. Let

* Presented at the 1952 IRE National Convention, March 3-6, New York. Reprinted from *IRE Transactions on Audio*, vol. PGA-6, March 1952.

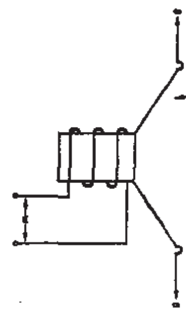


Fig. 1. Direct radiator loudspeaker in infinite baffle.

M = total effective mass of moving system (i.e., cone, voice coil, and effective air mass), kg
R_a = radiation resistance in mechanical ohms, N · s/m
C_m = effective compliance of suspension and back enclosure, if any, m/N
i = applied current to driving coil, amperes
l = wire length of driving coil, meters
T = flux density in region of driving coil tesla
R = dc resistance of driving coil, ohms
W_a = acoustic power radiated, watts
x = displacement of moving system, meters
L = inductance of driving coil, henrys
E = applied voltage to driving coil, volts

In all the following analogies a voltage generator of zero source impedance will be assumed. However, it should be clear that an amplifier of finite source impedance may be represented by an amplifier of zero source impedance in series with the appropriate impedance. Insofar as the overall system performance is concerned, the amplifier source impedance may be considered as an additional element of the series electrical impedance $(R_1+j\omega L_1)$.

The following two equations then describe the system [1], [2]:

$$L\frac{di}{dt} + Ri = (E-E_1) \quad (1)$$

$$\frac{M}{(Tl)^2}\frac{dE_1}{dt} + \frac{R_a E_1}{(Tl)^2} + \frac{1}{C_m (Tl)^2}\int E_1\,dt - i = 0. \quad (2)$$

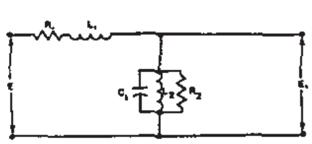


Fig. 2. Direct radiator loudspeaker in infinite baffle.

The electric circuit which satisfies (1) and (2) is shown in Fig. 2, where

$$R_1 = R \qquad L_1 = L \qquad C_1 = \frac{M}{(Tl)^2}$$

$$R_2 = \frac{(Tl)^2}{R_a} \qquad L_2 = C_m (Tl)^2 \qquad E_1 = (Tl)\frac{dx}{dt}$$

$$W_R = \frac{E_1^2}{R_2} = \dot{x}^2 R_a \quad \text{watts.}$$

The parallel combination of C_1, L_2, and R_2 may be said to represent the inverse mechanical impedance of this moving system multiplied by $(Tl)^2$ As the factor (Tl) is increased, the impedance of the parallel combination of C_1, L_2, R_2 increases as $(Tl)^2$ relative to the series electrical impedance of R_1 and L_1. Clearly, then, if (Tl) is increased far enough, E_1 will equal E for a relatively wide range of frequencies. The bandwidth for which $E_1 = E$ increases as $(Tl)^2$. If one refers to Fig. 3, which shows the radiation resistance per square meter of a piston, set in an infinite baffle, as a function of the ratio of piston diameter to wavelength of radiated sound, he will see that $E_1 = E$ is not a desirable condition for all frequencies. $E_1 = E$ implies a constant velocity condition and clearly, in this case, for D/λ less than 0.5, the acoustic output will drop at the rate of 6 dB/octave. It might be interesting to note that $D/\lambda = 0.5$ for a 15-inch radiator at approximately 440 Hz.

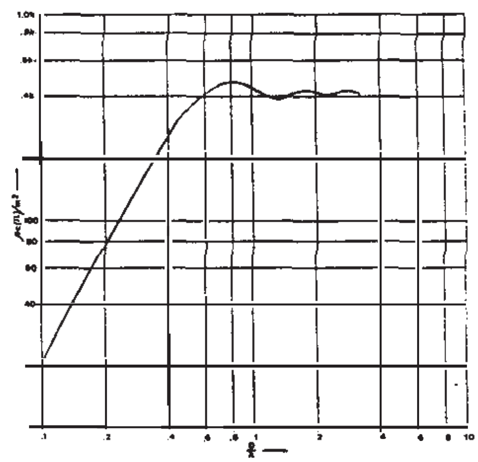


Fig. 3. Piston vibrating in infinite baffle.

Below $D/\lambda = 0.5$, the velocity of the radiator must increase inversely as the frequency if the acoustic power radiated is to be independent of frequency. The usual method for providing this frequency–velocity characteristic is to make use of the mechanical "resonance" of the loudspeaker. The insertion loss of this electromechanical equalizer is determined by the smallest D/λ down to which the response is to be uniform and the maximum variation in the low-frequency response which can be tolerated. If one sets the maximum variation in

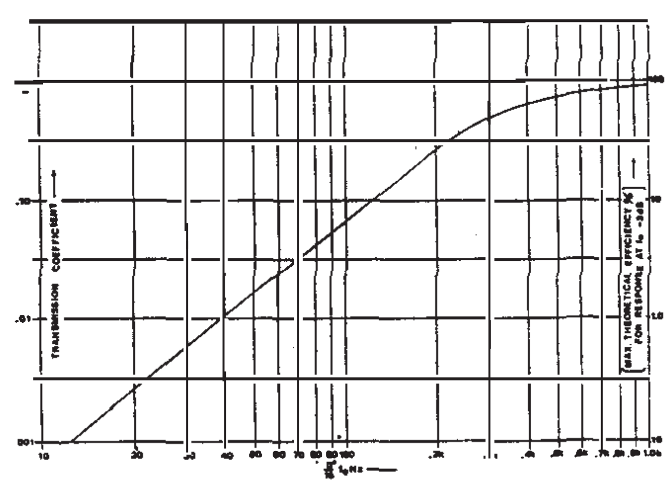


Fig. 4. Maximum transmisssion coefficient and theoretical efficiency, direct radiator loudspeaker.

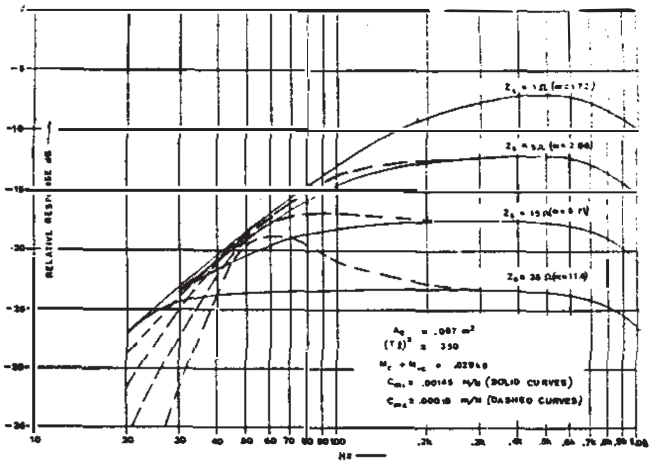


Fig. 5. Relative response 15" diameter, direct radiator loudspeakers.

low-frequency response at $\pm$ 1.5 dB, he may use the data in Fig. 3 and the concept of insertion loss to produce the curve shown in Fig. 4. The midrange (i.e., $D/\lambda = 0.5$) insertion loss is plotted as a function of $D/15f_0$, where D is the diameter of the piston radiator in inches and f_0 is the low frequency at which the response is to be down 3 dB for infinite suspension compliance.

Fig. 5 shows the relative acoustic output as a function of frequency for several values of α

$$\alpha = \frac{R_1 \times 10^2}{(Tl)^2}$$

and for two different suspension compliances. The larger compliance was chosen to provide a system resonance in the infinite baffle of 20 Hz, while the smaller compliance was chosen to provide a system resonance of 55 Hz in the infinite baffle. The ordinate zero dB represents a midrange efficiency of 100 percent. It should be observed that only a slight improvement in response for the $\alpha = 5.71$ curve was produced by using a mechanism having the smaller suspension compliance. For the $\alpha = 10.44$ curve, the rise of 5 dB as system resonance would be held to be objectionable by many listeners.

Note the excellent correlation between the data shown in Figs. 4 and 5, even though two widely different system resonances were used. Fig. 4 is for some systems slightly pessimistic from an efficiency point of view.

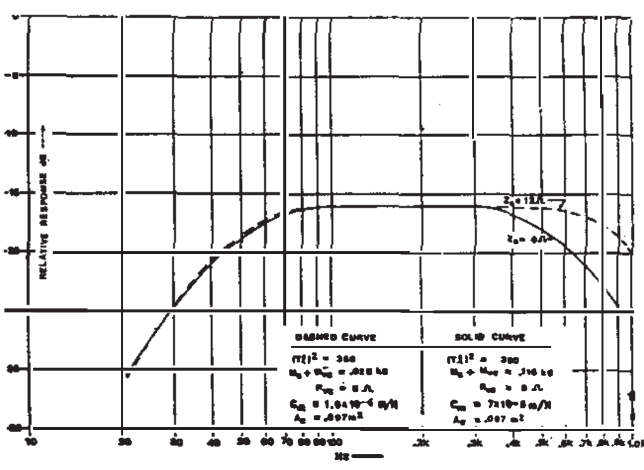


Fig. 6. Relative response of two different 15" direct radiator loudspeakers.

However, the extent to which this is so is small.

Fig. 6 shows the relative response curves obtained for two different 15-inch loudspeaker mechanisms possessing the same electromechanical coupling coefficients and the same system resonances (when mounted in an infinite baffle). The mechanisms differed in only one respect: the cone plus voice coil mass of one unit was four times heavier than that of the other. The heavy unit was driven by an amplifier which presented a zero source impedance while the lighter unit was driven by an amplifier which presented a 12-ohm source impedance.

CASE II: DIRECT RADIATOR IN A REFLEX ENCLOSURE

A diagram of a direct radiator in a reflex enclosure is shown in Fig. 7. The port in a reflex enclosure may be treated as a zero length tube; the usual end conditions for both ends of a tube apply at the port. In most cases, the inner surface of the enclosure is covered with sound-absorbing material. At such low frequencies as are likely to be encountered near the Helmholtz resonance of the port, the absorption coefficient of the lining may be neglected. The major loss for the port is then radiation into the space away from the enclosure.

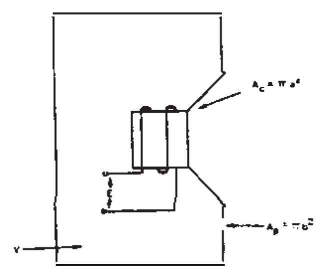


Fig. 7. Reflex enclosure loudspeaker.

The effective air mass acting at the port is approximately twice that due to the reactive component of the radiation impedance of one end of the tube, while the resistance offered by the port is the real part of the radiation impedance of the open end of the tube [3]. For constant pressure in the enclosure, the particle velocity at the port is proportional to the cone velocity and to the ratio of the cone area to the port area. The dimensions of the reflex enclosure are assumed to be small compared to the wavelength of sound for frequencies in the neighborhood of the Helmholtz resonance. The absorption coefficient of the material which lines the enclosure should be of sufficient magnitude at the resonant modes of the enclosure above the port resonance to damp them out.[1]

The port is coupled to the back of the cone by the compliant air of the enclosure. Let

ρ_0 = density of air, kg/m^3
$\pi a^2 = A_c$ = cone area, m^2
$\pi b^2 = A_p$ = port area, m^2
V = volume of enclosure, m^3
c = velocity of sound, m/s.

[1] For a discussion which treats the variation of C_v from the uniform pressure case see [4].

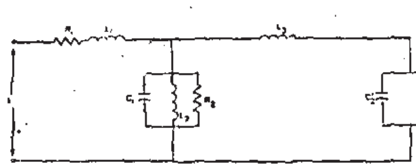


Fig. 8. Reflex enclosure loudspeaker without mutual impedance.

Then the compliance of the cone against the enclosure with the port closed satisfies the following equation:

$$C_v = \frac{V}{\rho_0 c^2 A_c^2}.$$

Let M_{ap} represent the total air mass effective at the port, and R_{ap} the radiation resistance of the port. If one assumes that there is no coupling between the port and the cone outside the enclosure, he obtains the electric circuit analogy shown in Fig. 8 [5], where

$$R_1 = R \qquad L_1 = L \qquad R_3 = \frac{(Tl)^2}{R_{ap}}$$

$$R_2 = \frac{(Tl)^2}{R_a} \qquad L_2 = C_m (Tl)^2 \qquad C_2 = \frac{M_{ap}}{(Tl)^2}$$

$$R_3' = \left(\frac{A_p}{A_r}\right)^2 R_3 \qquad L_3 = C_r (Tl)^2 \qquad \frac{N_r}{N_p} = \frac{A_r}{A_p}.$$

$$C_1 = \frac{M}{(Tl)^2} \qquad C_2' = \left(\frac{A_r}{A_p}\right)^2 C_2$$

It has been the author's experience that the impedance versus frequency curves for the analogy in Fig. 8 do not agree well with those of experimental units. The major differences appear to have been the frequencies at which the two low-frequency impedance maxima occur. The principal source of the discrepancy seems to be the omission of external coupling between the cone and port.

An approximate determination of the mutual impedance between the piston and the port may be determined in the following manner.

The pressure at a point p, distant R from the center of the vibrating piston and in the same plane (see Fig. 9), may be obtained by solving the following equation:

$$p(y) = \frac{i\rho_0 \omega u_0 e^{i\omega t}}{2\pi} \iint_s \frac{ds}{h} e^{-ikh}$$

The solution may be approximated by expanding e^{-ikh}/h in an infinite series and integrating the first four terms [6].

$$\frac{Zm'}{A_p} = \rho_0 f \pi a^2 \left[k - 1/6\, k^3 \left(R_0^2 + \frac{a^2}{2} \right) \right] + i\rho_0 f \left[4 \frac{a^2}{R_0} B\left(\frac{a^2}{R_0^2}\right) - \frac{2k^2 a^2 R_0}{9} \left\{ \left(5 + 3\frac{a^2}{R_0^2} \right) K\left(\frac{a^2}{R_0^2}\right) - \left(1 + 7\frac{a^2}{R_0^2} \right) D\left(\frac{a^2}{R_0^2}\right) \right\} \right]$$

where B, K, and D are the complete elliptic integrals defined and tabulated in [7].

The inclusion of the mutual impedance terms modifies the electric circuit analogy shown in Fig. 8 to the extent shown in Fig. 10, where

$$R_1 = R \qquad L_1 = L$$

$$L_2 = C_m (Tl)^2 \qquad C_1 = \frac{M}{(Tl)^2} - C_3$$

$$L_3 = C_r (Tl)^2 \qquad C_2' = \frac{M_{ap}}{(Tl)^2} \left(\frac{A_r}{A_p} \right)^2 - C_3$$

$$\frac{1}{R_2} + \frac{1}{R_4} = \frac{R_a}{(Tl)^2} \qquad C_3 \cong \frac{2\rho_0 A c^2}{\pi^2 R_0 (Tl)^2} B\left(\frac{a^2}{R_0^2} \right)$$

$$R_3' + \frac{1}{R_4} = \frac{R_{ap}}{(Tl)^2} \frac{A_r^2}{A_p} \qquad R_4 \cong \frac{(Tl)^2}{A_c^2 \rho_0 f k}.$$

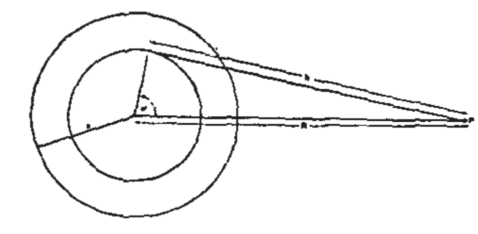


Fig. 9. Geometrical configuration.

It was found that this approximation modified the analogy shown in Fig. 8 so as to bring the impedance measurements of the electric circuit analogy into good agreement with those of experimental units. The inclusion of the mutual impedance into the electric circuit analogy for the reflex enclosure has been observed to produce the following differences from the electric circuit analogy which does not include the effect of the mutual impedance.

1) A reduction of the resonance frequency, which occurs above the port resonance, by 4 to 7 Hz out of an average of 65 Hz for several designs considered;

2) an increase of the resonance frequency, which occurs below the port resonance, by 2 to 4 Hz out of an average of 25 Hz for the several designs considered;

3) a sharper cutoff in response below the port resonance frequency.

No difference in the frequency at which the port resonance occurs has been observed between the electric circuit analogy which includes the effect of the mutual impedance and the electric circuit analogy which does not include the mutual impedance.

Figs. 11–14 show computed response curves for two different loudspeaker mechanisms mounted in two different reflex enclosures. Fig. 15 shows the computed relative acoustic output as a function of frequency for the same loudspeaker mechanism when mounted in two different types of enclosures. The amplifier source impedance was adjusted in each case to provide the same de-

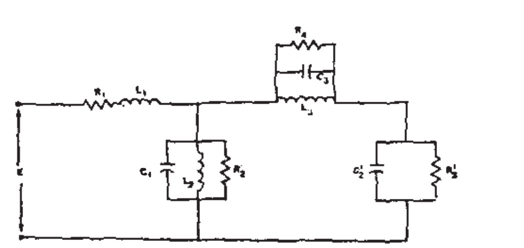


Fig. 10. Reflex enclosure loudspeaker with mutual impedance.

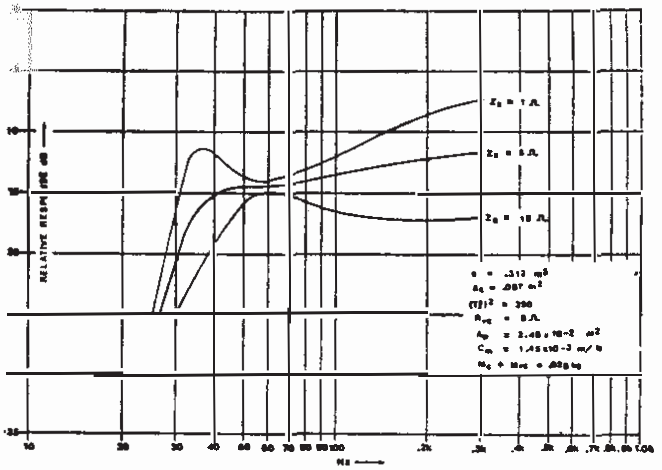


Fig. 11. Response curves (solid, Fig. 5) in reflex enclosure 0.312 m³ and port area 0.0249 m².

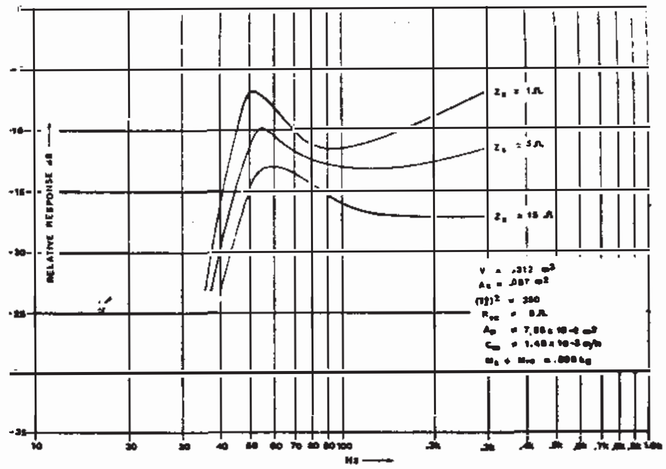


Fig. 12. Response curves (solid, Fig. 5) in reflex enclosure of 0.312 m³ and port area 0.0755 m².

gree of relative flatness in response. In the lower part of Fig. 15 are plotted the relative cone displacements as a function of frequency required to produce the response curves shown above. Note that a 6-dB improvement in overall efficiency is easy to attain with the reflex enclosure over the infinite baffle design and with no greater peak-to-peak displacement down to 40 Hz.

Many designs which provide a relatively high frequency port resonance (i.e., 70 to 100 Hz) produce considerable low-frequency distortion when driven at frequencies from 30 to 65 Hz. Below the port resonance, the acoustic loading of the cone or diaphragm is very low, large cone excursions ensue, producing very little radiation at the fundamental driving frequency and considerable harmonic distortion.

The electric circuit analogy is, of course, applicable to modified Helmholtz resonators in which a tube of finite length is used. The total mass effective at the port is then that due to both end corrections plus that due to air contained within the tube. The Helmholtz resonance frequency satisfies the following equation:

$$f_h \cong \frac{1}{2\pi}\sqrt{\frac{1}{M_{ap}C_r}} = \frac{bc}{2}\sqrt{\frac{3}{V(3\pi l+16b)}}$$

where all parameters are as before and l is the axial port length.

In the case of enclosures of small volume, the area of the port required for a given low resonance frequency may be so small that friction losses in the port may exceed radiation losses. Heavy, large diameter dummy cones with soft suspensions may be substituted for the air mass in the port to obtain low-frequency Helmholtz resonances in small enclosures.

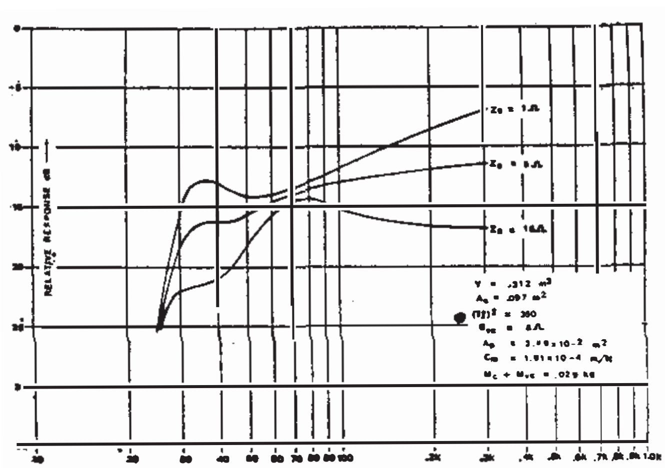
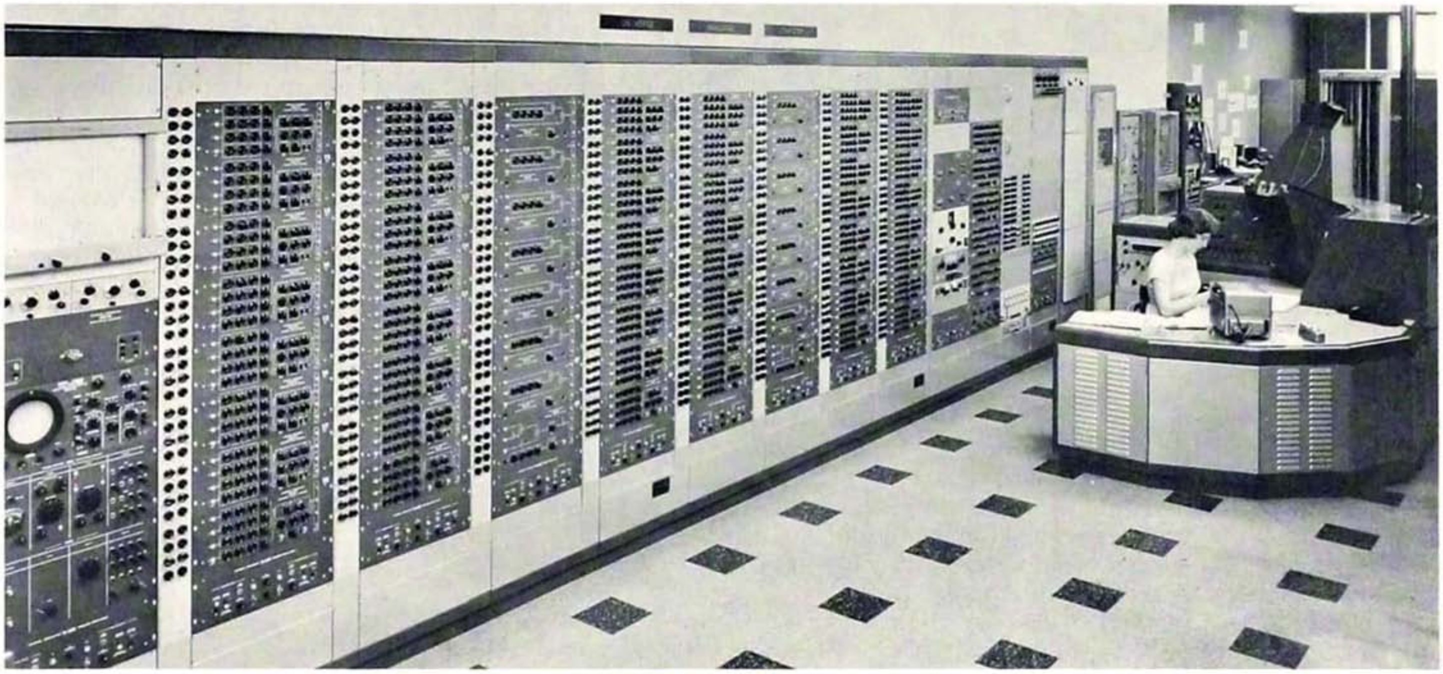


Fig. 13. Response curves (dashed, Fig. 5) in reflex enclosure of 0.312 m³ and port area 0.0249 m².

THE LIGHTNING EMPIRICIST

Advocating electronic models, at least until livelier instrumentalities emerge

Volume 11, Number 1 January 1, 1963



Analog Installation at Hercules Powder Co., Magna, Utah, U.S.A.

INTENTIONALLY UNCONVENTIONAL ANALOGUERY

YOUR orthodox present-day analog computing machine is a thing of pots and patches. That is to say it presents to the viewer a pattern made up of one or more patch bays and a large number of ten-turn potentiometers. Patch cords, generally plugged into boards which occupy the patch bays, establish the sequence of operations comprising the system to be simulated and solved. The settings of the potentiometers are made in accord with parameters which determine the quantitative nature of the system under study.

Perhaps the most striking feature of this Hercules installation is the absence of pots and patches, both being replaced in a uniform way by rotary, manual switches. The switch settings, which may be recorded in a perfectly natural way, embody and define the interconnections as well as the parameters of the system being investigated. A number of advantages follow from this pervasive application of switching throughout the machine, as we shall explain in a moment.

Other features include: a conceptual framework which is mathematical, rather than electronic; provision for *panceleric* performance, whereby the same apparatus operates periodically or aperiodically with a computing time ranging from below 0.05 seconds up to 50 seconds or beyond; and a large-screen electronically calibrated display which permits direct comparison of many variables and convenient photography.

Mathematical Framework

We love operational amplifiers as much as the next analog enthusiast. Many are involved in this installation. The fact is we have probably manufactured most of the operational amplifiers now being used on this planet. But we deprecate the practice of asking the user of computing machines to think in terms of such amplifiers. In this machine, for example, the user deals only with mathematical concepts and operations. His basic such operations comprise: linear combinations of one to four terms; time integrals of such combinations; multiplication and division; bounds; and functions of variables. These, and certain other minor operations, are collected into three principal operational modules, described elsewhere in these pages. Note in particular that the concept of linear combination includes, as special cases, the ideas of adding, subtracting, inverting, and averaging.

The user, in dealing with his differential equations, or functional block diagrams, or signal flow charts, has enough to worry about without electronic folklore. To him such facts as the habitual minus sign involved in one-amplifier transformations must appear as an irrelevant trick of fate. By offering a simple environment for the user's deliberations, his attention may center on the important matters of normalising, scaling, ranges of parameters, etc.

Switched Coefficients

Multi-turn potentiometers belong to our culture. They are available from many makers, with knobs and scales of surpassing cleverness. We like them. We have used them in certain instruments. We do not advocate them, however, for a general purpose machine such as this. Among other reasons are the following:

(a) They are hard to set.
(b) They are hard to read, especially from over two feet away.
(c) They are subject to loading errors.
(d) They waste power.
(e) They do not lend themselves to conveniently calibrated conductance settings.

The usual procedures in which potentiometers are manipulated to establish coefficients do not involve the dial calibration except as an approximate guide or as means whereby one can return to previously held positions. Since the fractional rotation itself is unusable as an accurate coefficient measure, other means are used to measure the fractional transmission of signal through the potentiometer, the latter being manipulated manually or automatically until the desired transmission results. Conversely, an experimental setting must be followed by such a measurement to formalize a coefficient value. This seems too bad, and inferior to a means of coefficient setting wherein the numerical value may be established or read out directly.

The preferred method of handling coefficients in this machine, covering in fact substantially all the computing parameters, involves switch setting by decades for numerical value and switch setting for algebraic sign. The settings as made have validity independently of load. They are easily and quickly made and readable from a distance. They are economical as to electrical power. This is not the place for a detailed account of the means employed to accomplish this end, which is simple if mildly subtle. A three-terminal network is involved, whose transfer conductance progresses arithmetically through as many decades as are required for precision.

Voltages also may be incorporated by the same switching structures, both for sign and for numerical value, as fractions of reference voltages, in decades of conductance switches. Such voltages, indeed, may be looked upon as parameters of the mathematical description.

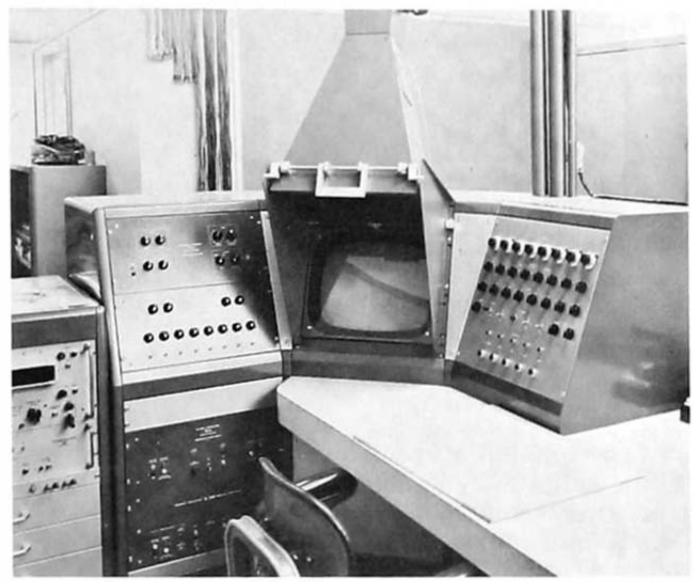
Switched Interconnections

Patch cords are simply replaceable lengths of covered wires which connect the outputs of selected computing sections to the inputs of others, and which thus determine the order of internal operations, or "program." Patch boards are traditionally provided into which these cords can be plugged, and which in turn are bodily inserted — as gigantic multiple plugs — into patch bays which communicate electrically with all relevant elements of the machine.

The idea of the patch board, of course, is to enable *storage* of the program, or pattern of interconnections. While it does this indeed, it fails to store the parameters, which still must be established by setting potentiometers. As all initiates will agree, however, the dominant feature of patch boards is the wild and vermiform tangle of cords which proliferate on them in the case of problems which are more than trivial. These tangles are picturesque but not particularly efficient, and there is evidence that they conduce errors and that such errors are hard to track down.

In the present installation, and in some earlier experimental structures, all wires are removed from the front of the machine. Instead, every interconnection is made by a system of switches very much like that applied for parameters. A pair of decade switches located near each input on each modular panel allows choice, as the variable to be applied to that input, of any output variable in the machine. Each such pair of switches may be said to embody an "inverse address," for evident reasons. Experience has borne out that this technique of switch programming is resistant to errors and easy to operate. It looks like the interconnection method of the future.

Problems for solution are set up on standard tabular sheets, which formally resemble the modular panels of the machine itself. The entries are simply the switch settings and are uniform in style for interconnections and coefficients. These tables naturally serve for storage of problem embodiments for future reference.



Accessories, Including Display

At a central point, thoughtfully arranged about and upon an operator's desk, are instruments attending control of and access to the computing machine proper. Included are: timing apparatus for determining the speed of operation of the machine, relating to the set-run-hold cycle of computation; recorders such as Visicorders and fm tape machines; generators of standard time functions; and, of course, one or more human operators of appropriate sort.

We are pleased that a role of importance was given to our electronically calibrated, large-screen, display system. Hercules put this system into cabinetry of their own choosing. From the central desk, switches select any of the variables on the machine for plotting on the cathode ray display. Advantages of our display include, along with voltage and time calibration to good accuracy, the ability to compare solutions by superimposing them on the same coordinates, monitoring up to eight solutions, and cross-plotting of one such solution against up to seven others.

Appreciation

The Hercules installation is called by them the UNI-VERSE COMPUTER, presumably to reflect its universality of application. It is our policy not to judge the spelling of customers. The fact is we are very proud to have our apparatus chosen for this ambitious and forward-looking installation. While we naturally have heard of some of the computing applications, we cannot describe them here without impropriety. We can say that the machine is in use on a full schedule and that some of the problems being solved are those formerly submitted to the digital facilities at Hercules. For problems of the latter type, the analog equipment permits the user more intimate contact for changes in solution conditions — changes which could not have been foretold before solution was attempted. This points up the superiority in general of analog methods for exploratory studies.

The following individuals among others, at Hercules, have contributed to making successful the installation and its subsequent applications: Messrs. D. J. Cholley, F. M. Wanlass, K. S. Cook, and L. Godfrey. We want especially to thank Mr. David J. Cholley, Analog Analyst at Hercules, with whom we have had the closest contact. His courage and faith supported our efforts in more ways than we like to admit. It is hard to see how the project would have turned out so well without him. This is how he looks when not displeased. He prepared for his present activities at Mount Union College, Alliance, Ohio, taking the degree of Bachelor of Science.

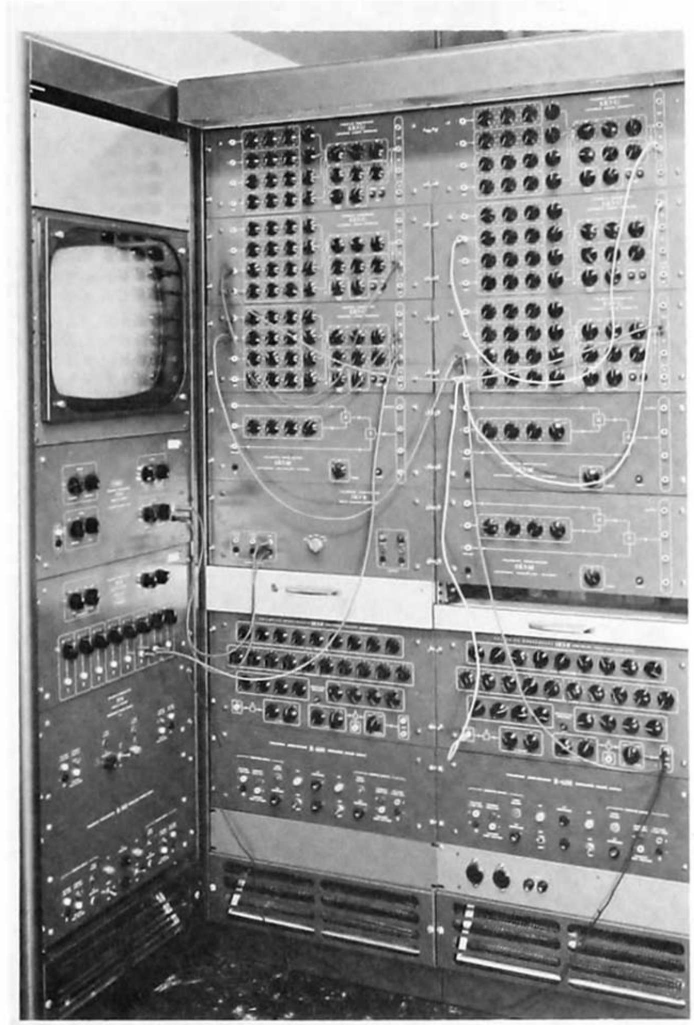
Incidentally, we have reprinted, and have available for interested readers, a pamphlet written by Mr. Cholley in connection with this installation, having the title "A Particular Application of FM Tape Used With An Analog Computor."

Complement of Philbrick Equipment in Hercules Installation as Shown in Photographs

48 Model SK5-U Universal Linear Operators
16 Model SK5-M Multiplier-Dividers
5 Model SK5-F Arbitrary Function Generators
3 Model SK5-R Relay Control Components
1 Model 6009 Operational Manifold
9 Model R-600 Regulated Power Supplies
3 Model R-500 Regulated Power Supplies
1 CS2 Central Signal Component
1 CRM Central Response Component
1 5934 Multi-Channel Calibrated Display System
1 Camera Accessory for Display System

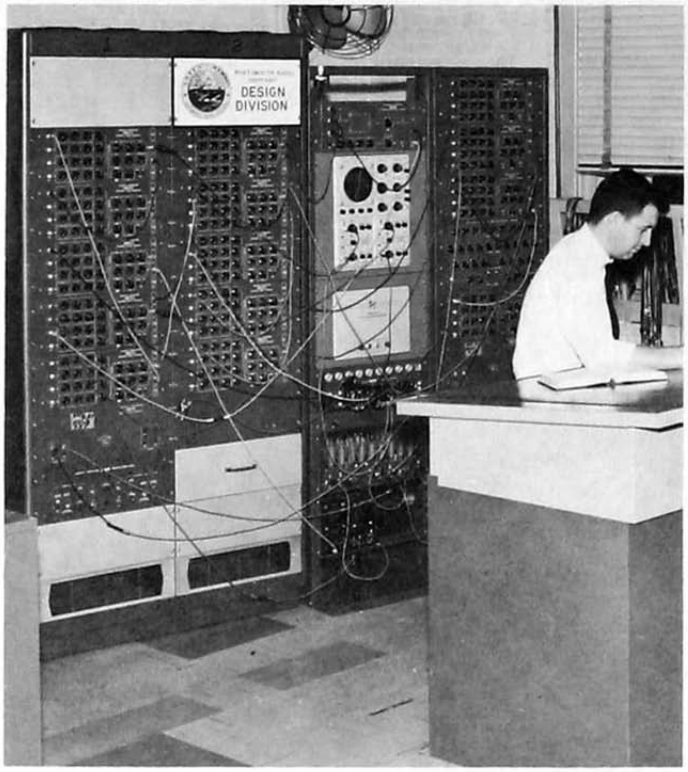
Selected Additional Installations

Since this issue is principally devoted to the nature and applicability of the SK5 modules concept, there is included below a group of photographs and brief descriptions which cover representative computing facilities in which the SK5 devices are given central roles.



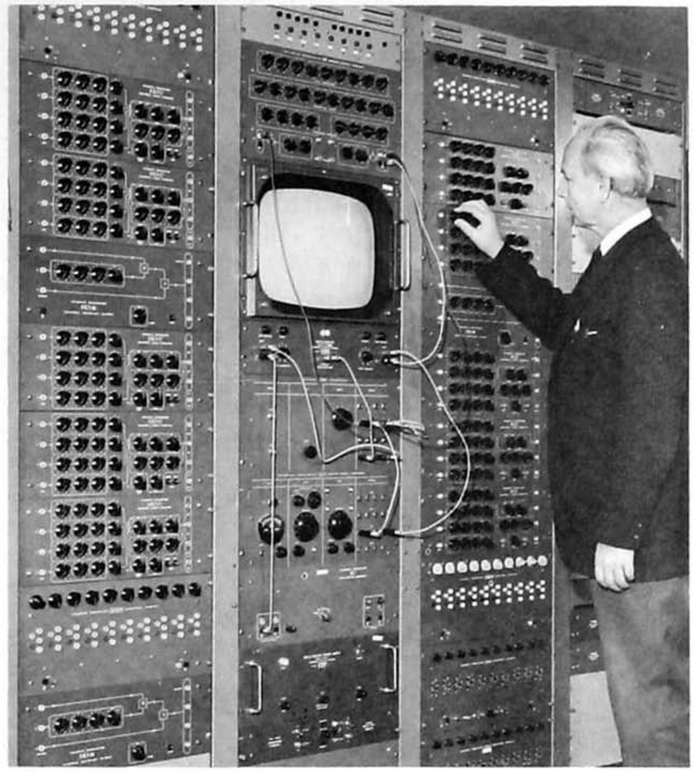
The new installation in the Physiology Department of Western Reserve University Medical School in Cleveland is to be used for various research problems principally in the field of endocrinology. The computor will model transient diffusion, mixing, and reaction kinetic phenomena to help determine the mechanisms involved in the biosynthesis of hormones. In charge of the computor is Dr. Thomas Hoshiko, Assistant Professor, whose research is concerned with ion transport through membranes. Others concerned with the computor include Dr. George Sayers, Head of the Department of Physiology; Dr. Howard Sachs, whose research concerns the biosynthesis of Basopressin; Mr. Gerald Lower, Electronics Technician; and Mr. Barry Lindley, Graduate Student.

The system shown consists of six SK5-U's, three SK5-M's, two SK5-F's, and a 5934 Calibrated Display.



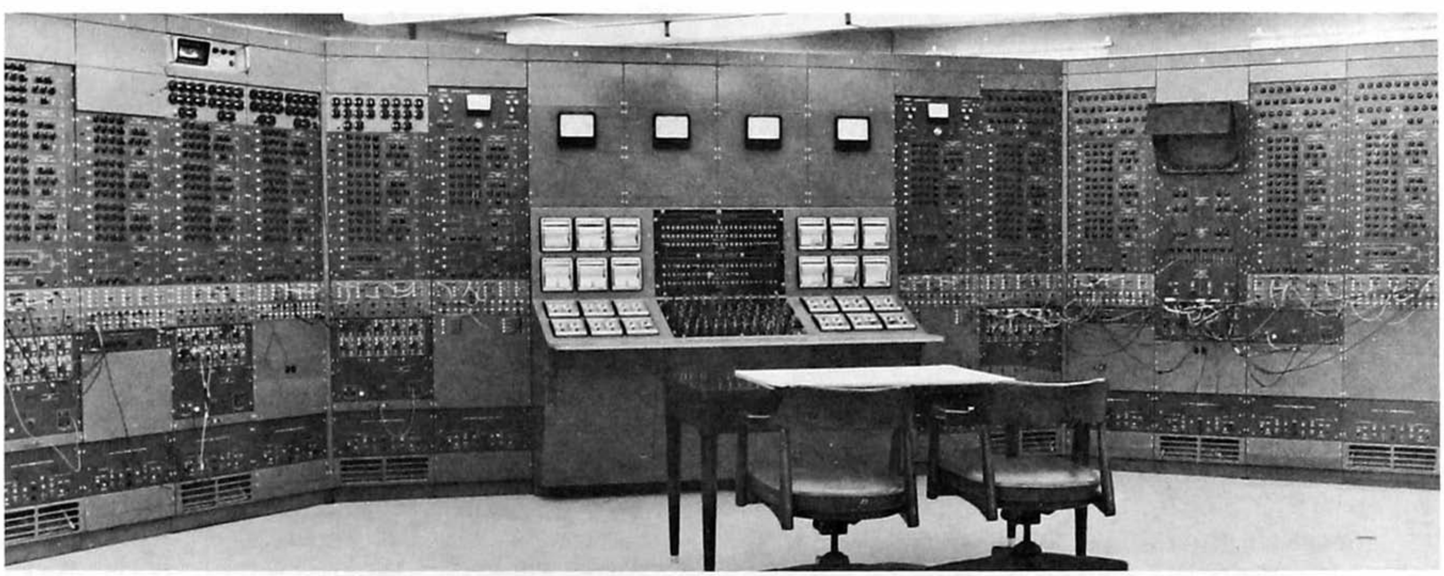
The SK5 Computor at the Portsmouth Naval Shipyard, New Hampshire, is in the Computing Branch of the Design Division which is in the Planning Department under the guidance of Richard Wade, the Supervisory Engineer, and Robert Perkins, the Electronic Engineer.

This facility is made available to all engineers and scientists connected with the yard on an "open shop" basis. Typical problems solved include torpedo firing, mechanical shock, emergency surfacing & emergency flooding of submarines, vibration and design of dash-pots.



Dr. Heinz W. Kasemir, shown in photo above, is the scientist in charge of the U. S. Army Electronics Research and Development Laboratory, Fort Monmouth, New Jersey, Oakhurst Field Station.

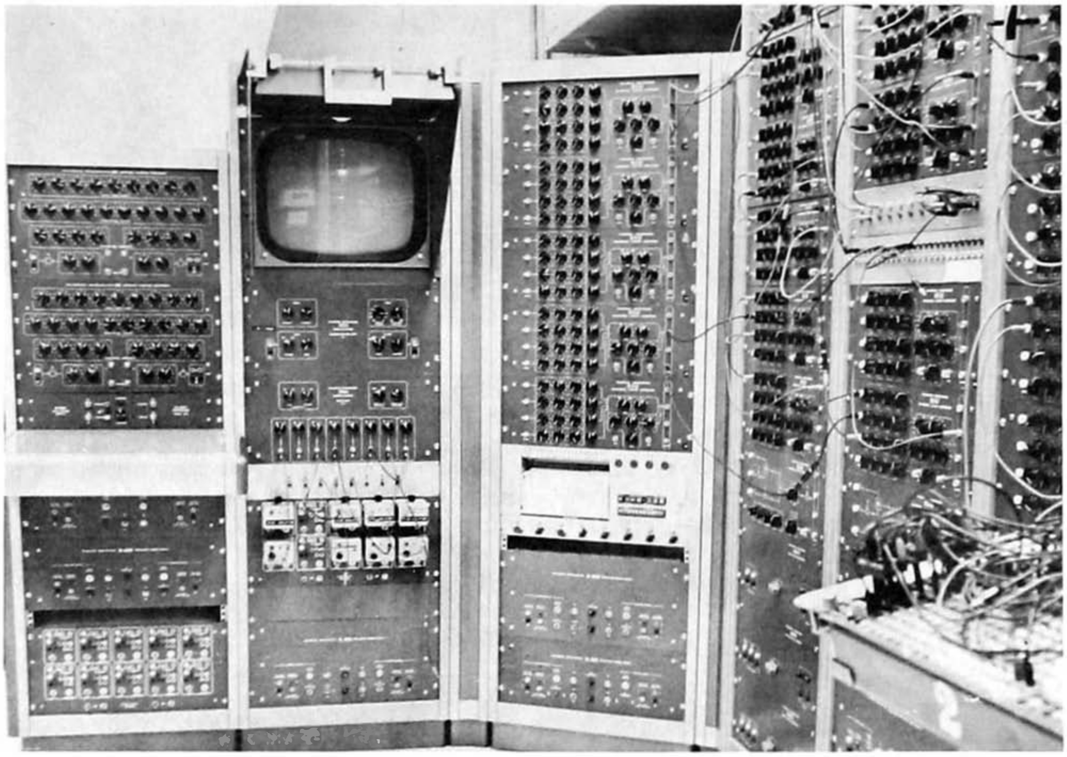
This installation is used for such typical purposes as solving complex differential equations bearing on problems of atmospheric electricity, and in the development of sophisticated electronic systems for making frequency analyses of the speed of lightning strokes and altitudinal variations of air-earth electrical currents.



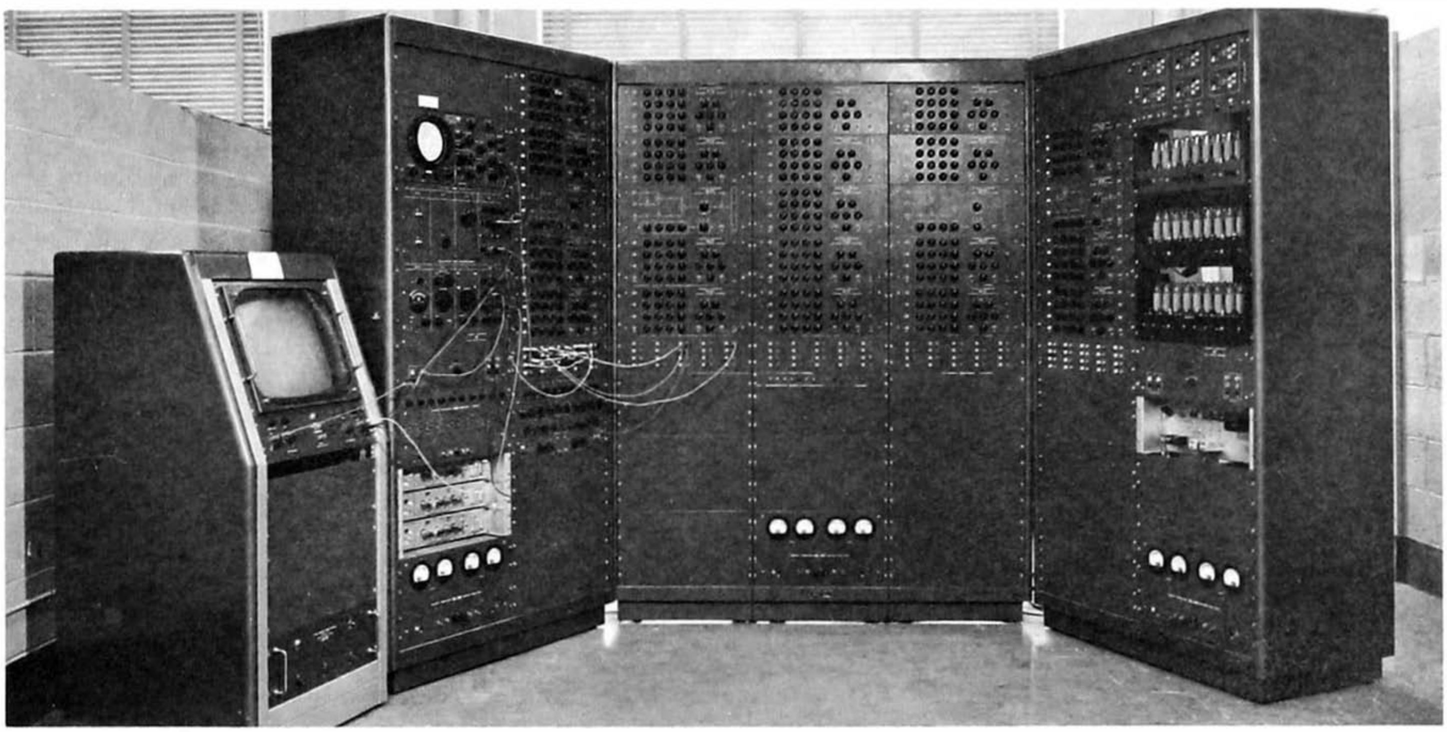
The Dow Chemical Company installation at the Process Control Laboratory, Freeport, Texas, is used for the study of process control systems. Engineered by the Project Engineering Department of The Foxboro Company, it consists of K5-U computing modules and twenty-four Foxboro electronic controllers and pen recorders.

The Texas Division of Dow Chemical Company had spent two years developing analog techniques, and procured this computor built to fill their needs. The processes are modeled with the K5 computor and controlled with hardware similar to that used in actual processes. This enables the engineer or process man to test and evaluate the design of a plant before it is built and to train operating personnel.

This controller/simulator is under the supervision of Porter Hart, Director of the Process Control Laboratories, and Bernard Poettker, Senior Research Physicist. The K5 portion of this installation consists of 32 K5-U's, 13 K5-M's, 5 K7-A10's, 5 RC's, 4 FF's, and a 5934 Display System.



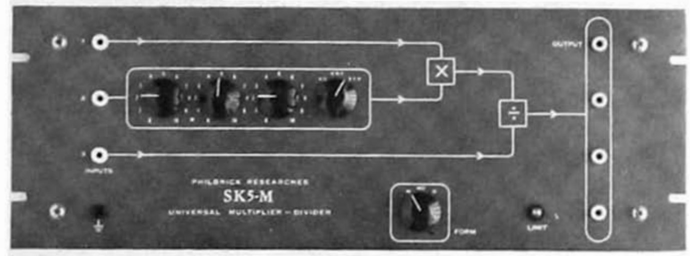
Shown at the left is an installation at the Latter Day Saints Hospital in Salt Lake City which is used for the reduction of biological data and the simulation of biological systems. Dr. Homer Warner, who is in charge of this area has developed many novel techniques for solving physiological problems. His plans for the future include coupling the K5 computor to a digital system in order to perform convolution and correlation. He is ably assisted by Sanford Topham and Wayne Wiscomb. Included are 19 K5-U's, 5 K5-M's, 3 RC's, 2 K7-A10's, 3 FF's, and a 5934 Display System & Camera.



The Analog Computing Facility at the Massachusetts Institute of Technology is located in the Engineering Projects Laboratory. Conceived and promoted into tangible form by Professor Henry M. Paynter of the Mechanical Engineering Department,* it may be used by staff and students of all departments. It is available on an open-shop basis and is in active use on a twenty-four hour, seven-day-a-week schedule. Because it is modular in assembly, the computor has been used successfully during peak load periods, even in the midst of the bi-weekly, three-hour period reserved for maintenance and calibration. Since the computor was installed in 1958, none of the equipment has been out of service for more than a day. The installation includes 20 K5-U's, 4 K5-M's, 2 RC's, and a 5934 Display System (which does not appear in this photo), as well as some earlier manifolds, utility packaged amplifiers, and regulated power supplies.

*A long and fruitful cooperative relationship has existed between this Company and the above-named Department of MIT, particularly through association with Doctor Paynter himself. His inspiration and acknowledged mastery of engineering analysis have supported our convictions on the power of models.

Philbrick SK5 Series of Computing Modules



MODEL SK5-U UNIVERSAL LINEAR OPERATOR

The SK5-U is the basic module of an SK5 assemblage, providing capability to compute a linear combination of four input variables, or alternatively the time integral of such a combination. All parametric and internal functional changes are made with multiposition switches which provide both convenience and flexibility. Direct mathematical programming eliminates much of the confusion and drudgery associated with conventional analog computing.

The SK5-U applies an independent arbitrary coefficient (of either sign) to each of four input variables. The coefficients are repeatably and rapidly set with decade switched conductive networks (rather than potentiometers) to three significant figures with less than 0.2% error. Their values range from −10.00 to +10.00 in steps of 0.01.

The output may be the sum of these weighted variables multiplied by ten raised to an arbitrary power (—1, 0, 1, or 2) and added to an adjustable arbitrary three-digit index. Alternatively, the output may be selected to be the time integral of the weighted sum multipled by an arbitrary power of ten (0, 1, 2, 3, or 4) starting from an initial condition determined by the three-digit index. Set, hold, run logic can be manually or automatically controlled independently of or in synchrony with other integrators.

The availability of five integration rates independently selectable for each SK5-U enables systems consisting of both slow and fast sub-systems to be conveniently modelled. Most problems may be solved repetitively at a rapid rate convenient for optimizing parameters with an oscilloscopic display, or singly at a conveniently slow rate for electro-mechanical input or output devices.

The SK5-U output is non-linearly constrained to remain between precisely fixed upper and lower bounds. These bounds, in addition to providing protection against chopper stabilized amplifier saturation, may be used to achieve a variety of elementary non-linear operations such as absolute value, arbitrary bounds, dead zone, hysteresis, and crossing detection.

The SK5-U has two chopper stabilized amplifiers. Normally, one is used as an inverter to achieve the coefficient sign option. However, these amplifiers may also be used independently, thus increasing the computing capability of a single SK5-U with the sacrifice of arbitrary sign for the coefficients. This feature is of arbitrary sign for the coefficients.

MODEL SK5-M UNIVERSAL MULTIPLIER-DIVIDER

The SK5-M is an analog multiplier-divider of superior performance and reliability, designed for use in computing, controlling, and measuring applications. The output is proportional to the product of two input variables divided by a third. The constant of proportionality is switch selected in the ranges 0.100 to 1.000 in steps of 0.001 or 1.00 to 10.00 in steps of 0.01.

A mode switch automatically applies 100 volts to the denominator input, in the Multiply mode, or 100 volts to one of the numerator inputs, in the Divide mode, if one does not require simultaneous multiplication and division. Square rooting of a numerator input is accomplished by feeding back the output into the denominator input using the Divide mode.

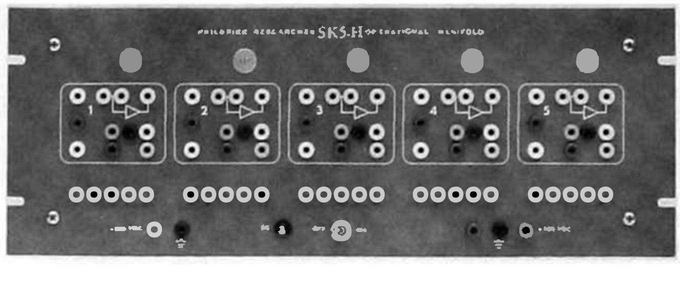
As is the case of all SK5 modules, the SK5-M has nominal output range of ±100 volts. The output is bounded at ±110 volts to prevent saturation of the chopper stabilized amplifiers employed in the circuit, thus assuring virtually instantaneous recovery from any output over-voltage condition such as might result from a small denominator input.

The numerator input variables may be either positive or negative; but should the denominator input be negative, the SK5-M output will bound at ±110 volts.

The SK5-M employs a triangular wave method for multiplication similar to that employed in the K5-M and MU/DV Multipliers. An inherent property of this method is the reduction of error as one or both of the numerator terms approaches zero. The SK5-M static accuracy is comparable with that of a good servo-multiplier (less than 100 millivolts including eight hour drift) yet the SK5-M has a bandwidth comparable to many quarter-square multipliers.

The SK5-M may be used to measure directly the mean square value of a signal having frequency components as high as 3 kcps with no apparent dynamic error. This may be accomplished by inserting a suitable capacitor in parallel with the feedback resistor of the output amplifier using jacks provided on the rear panel.

As in the case of the Model SK5-U module, the inputs and output are available on a rear panel connector as well as on front panel jacks, thus facilitating either direct front panel patching or remote patching or switching.



MODEL SK5-H OPERATIONAL MANIFOLD

The SK5-H Operational Manifold contains five high gain, wide band dc chopper stabilized and boosted operational amplifiers each capable of driving a 10 K load from −100 to +100 volts. For each amplifier, there are input and output terminals as well as servicing signals made available on conveniently spaced front panel jacks. As is the case in all SK5 computor modules, these terminals and signals are available on rear panel connectors as well. This arrangement facilitates either direct front panel patching or remote patching and switching.

Individual boxes (Model U2), which plug into these front panel jacks, are available in both committed and uncommitted forms. The committed U2 boxes provide standard analog operations such as:

- Addition
- Multiplication by an Arbitrary Coefficient
- Differentiation
- Function Generation
- Integration
- Selection
- Bounding

These enhance the capabilities of an SK5 computor by providing, at low cost, operations which cannot be performed by an SK5-U, M, or F, or which do not make efficient use of the capabilities of these modules.

The uncommitted U2 boxes enable convenient assembly of special operational networks which involve a single amplifier. The use of these boxes is beneficial in any application where hum and noise must be small because the aluminum case properly grounded provides shielding against ambient electrostatic and electromagnetic fields.

The SK2 amplifiers in SK5-H are particularly well suited for critical applications. The dc open loop gain of these amplifiers is greater than 10^9 while noise, offset, and leakage are typically no more than $100\mu v$, $30\mu v$, and 10^{-10} amps. respectively. Outputs are bounded at about ±105 volts (passively) to prevent saturation of the chopper stabilized amplifiers, thus assuring virtually instant recovery from any output over voltage. Under usual circumstances, the amplifiers can drive large capacitive loads. For instance, when the load resistance to ground is 100 K or greater, a load capacitance as large as $1000\mu\mu F$ can be driven at the maximum amplifier rate, ±6 volts/μsec., without special stabilization.

Premium quality components are used throughout to achieve high reliability consistent with many on-line control and critical measurement applications. Furthermore, as with the other SK5 modules, the SK5-H has an individual induced draft fan of capacity adequate to maintain the chassis and panel at almost ambient temperature.

MODEL SK5-F ARBITRARY FUNCTION COMPONENT

The SK5-F performs an arbitrary static non-linear operation on a single input variable. In addition to having non-interacting break-point and slope adjustments, the SK5-F provides adjustable parabolic smoothing at each break point, thus approximating the desired function with a sequence of linear and tangent parabolic segments. Many smooth functions may be approximated using the SK5-F with an error not exceeding 0.2% of full scale at any point along the function.

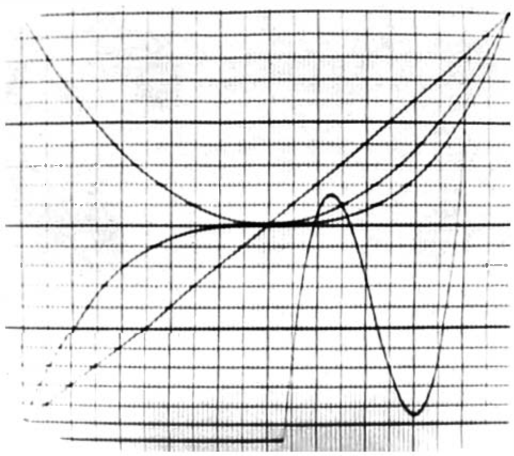
Although the SK5-F resembles its predecessor, the FF, many notable improvements have been made. Slope increments as large as ±10 may be adjusted accurately without interaction with previous adjustments. The chopper stabilized amplifiers have limit protection so that they cannot saturate during the function programming. Bandwidth is independent of the function programmed (3db at 2 kcps), except that it may be nearly doubled if the maximum slope increment can be limited to ±5 by feeding back the output to the null input. Also the null input may be used to program functions on the SK5-F which may be available as voltage functions of time.

Premium SK2 series amplifiers and high quality components are used in each critical location to achieve good reliability and repeatability. Furthermore, the module is cooled with an induced draft fan of sufficient capacity to prevent significant temperature gradients.

Inputs and outputs are available on front panel jacks and on a rear panel connector so that either local front panel patching or remote switching or patching may be employed.

Typical of the U2 plug-in components used with the SK5-H Operational Manifold (at left) is the U2-A Adder shown here.

The available operators are enumerated in the SK5-H section; or the user may build his own specials in uncommitted cases of this type, which are available in kit form and require only the simplest mechanical modification. This can be done in any laboratory or model shop.



Candid Shot of our Celebrated Graph Paper

The 5934 Multichannel Calibrated Display System is a convenient monitor for such instrumentalities as an analog computor, providing an accurate readout for repetitive or triggered behaviour. Up to eight variable voltages may be plotted simultaneously in correct phase and voltage relationship against a highly accurate coordinate system. The 5934-0 Display System Camera provides a 4 x 5 photographic record similar to the one reproduced. This reproduction is a negative, the customary photograph having white lines on a black field.

The coordinate grid consists of 21 horizontal lines at 10 volt increments from -100 to $+100$ volts and 101 vertical lines at 1% intervals of the selected display period. Each coordinate and signal voltage is sampled every 62.5 microseconds. A vertical flying scan system displays these samples on a large cathode ray tube. Periodically, an entire vertical scan is brightened to provide the time calibration lines. Since the grid and signals are displayed simultaneously and by the same electron beam, drift, distortion, phase, and parallax errors are eliminated. The resulting accuracy of the coordinate grid relative to the signals permits voltage and time to be read with an accuracy comparable to that of the best recorders, at speeds useful for instant evaluation of repetitive analog solutions.

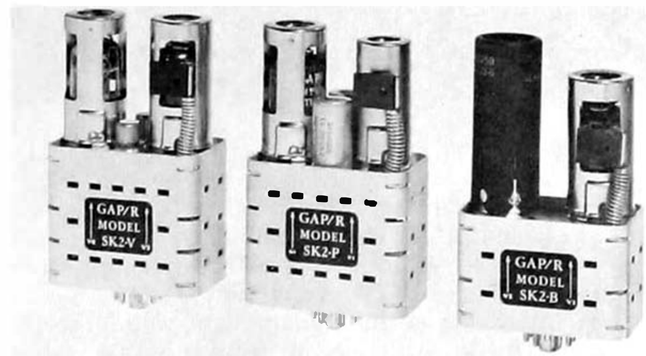
The 5934 produces an output signal which may be used to synchronize the computor to the repetitive display period. Alternatively, the start of the 5934 display period may be triggered periodically or aperiodically with an external signal enabling the display interval to be synchronized with an input device such as a tape recorder.

Although the usual application of the 5934 involves the plotting of computor variables vs. time, the cross plot mode permits up to seven input variables and the voltage calibration to be plotted simultaneously against the eighth. This is useful in making frequency response measurements and in phase plane analyses.

The figure above demonstrates the rather unusual use of the 5934 to present the solution of an algebraic cubic equation. The independent variable is swept from its minimum to maximum value as ramp function of time. The ramp is generated by integrating a constant. Integrating again yields the square of independent variable and a third integration yields the cube. By weighting and summing these functions and adding a constant in an SK5-U, the desired cubic equation is programmed such that when the SK5-U output is zero, the cubic equation is satisfied.

HIGH-PERFORMANCE AMPLIFIERS

The SK2 Series of octal plug-ins used together form an amplifier with outstanding characteristics for commercial applications. Their superior performance led to their utilization in the SK5 computing modules.



Model SK2-V Differential Operational Amplifier is an octal plug-in designed for computing, controlling, and instrumentation. All SK2 amplifiers, have an all-metal case and plug and feature our prize-winning exoskeleton construction. The SK2-V has ± 100 volts output at ± 3.0 ma, although much higher voltages are possible.

Low drift, low input error, and unusually high gain enable the SK2-V to perform very accurate computing operations. High speed computing is possible too for the open-loop gain, greater than 100,000 at dc, reaches unity at about 1.0 mcps.

Model SK2-B Booster Amplifier, also an octal plug-in, was designed especially for use with Philbrick operational amplifiers (such as SK2-V, K2-XA, and K2-W).

SK2-B follows an SK2-V where output currents greater than 3 ma are required, the SK2-B being used to drive the load, allowing the SK2-V to run essentially unloaded. This combination will drive loads up to ± 20 ma at ± 100 volts and will maintain the high performance of the SK2-V.

Model SK2-P Stabilizing Amplifier is an octal plug-in chopper amplifier specifically designed to stabilize operational amplifiers such as the SK2-V. The drift rate of the SK2-V, under optimum conditions, is typically about 5 mv per day, but can be many times greater (perhaps 25 or 50 mv) if proper precautions are not observed. When combined with the SK2-P, these drift rates can be reduced to a level usually below 100 microvolts per day. Dependability has been stressed as the major objective in the design of the SK2 series amplifiers. Dissipation has been kept so low that even without any heat sink the case is only reasonably warm.

THE LIGHTNING EMPIRICIST

Beginning with this issue, THE LIGHTNING EMPIRICIST will be published at quarterly intervals by Philbrick Researches, Inc., at 127 Clarendon Street, Boston 16, Massachusetts. This Journal began in June, 1952, and has appeared "aperiodically," in its earlier format, seven times since then. The staff of Philbrick Researches will serve in the various editorial and production capacities as needed. Comments and contributions will be welcome and should be addressed to Editor, LIGHTNING EMPIRICIST.

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An AES Paper Presented October 27, 1950

Direct Radiator Loudspeaker Enclosures

HARRY F. OLSON*

A comprehensive analysis of the effect of cabinet configuration on the sound distribution pattern and overall response-frequency characteristics of loudspeakers.

The principal factors which influence the performance of a direct-radiator loudspeaker are the mechanism itself, the acoustical impedance presented to the back of the mechanism by the enclosure, and the outside configuration of the enclosure. The major portion of the work involving cabinet research, development, and manufacture has been directed towards the acoustical impedance presented to the back of the loudspeaker mechanism by the enclosure. The volume of the cabinet and the internal damping means play the most important role in determining the acoustical impedance presented to the back of the loudspeaker. In other words, most of the considerations concerning the design of cabinets for direct-radiator loudspeakers have involved the volume or overall dimensions of the cabinet which—together with the mechanism—determines the low-frequency performance. The third factor, namely, the exterior configuration of the cabinet, influences the response of the loudspeaker system due to diffraction effects produced by the various surface contours of the cabinet. The diffraction effects are usually overlooked and the anomalies in response are unjustly attributed to the loudspeaker mechanism. Therefore in order to point up the effects of diffraction, it appeared desirable to obtain the performance of a direct-radiator loudspeaker mechanism in such fundamental shapes as the sphere, hemisphere, cylinder, cube, rectangular parallelepiped, cone, double cone, pyramid, and double pyramid. It is the purpose of this paper to present the results of the diffraction studies made upon these fundamental shapes. The response-frequency characteristics of a direct-radiator loudspeaker mechanism mounted in these different housings yield fundamental information regarding the effect of the outside configuration of the cabinet upon the performance of this combination. From this study it is possible to evolve a cabinet shape which has the least effect in modifying the fundamental performance of a direct-radiator loudspeaker mechanism.

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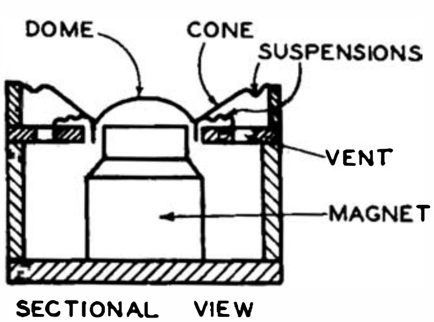


Fig. 1. Sectional view of the loudspeaker mechanism.

Characteristics of the Sound Source

In the experimental determination of the performance of direct-radiator loudspeaker mechanisms in various shaped enclosures, some consideration must be given to the radiating system. These considerations include the directional characteristics of the sound source and the sound power output characteristics of the sound source as a function of the frequency.

In order to obtain the true diffraction effects which are produced by the different enclosures, the radiation emitted by the sound source must be independent of the direction. Since the diaphragm of the direct-radiator loudspeaker mechanism used in these tests is relatively very small, it can be assumed that it is a piston source. The directional characteristics of a piston source are given by

$$R_\alpha = \frac{2J_1\left(\dfrac{2\pi R}{\lambda}\sin\alpha\right)}{\dfrac{2\pi R}{\lambda}\sin\alpha} \qquad (1)$$

where R_α = ratio of the pressure for an angle α to the pressure for an angle $\alpha = 0$,

J_1 = Bessel function of the first order,

R = radius of the piston, in centimeters,

α = angle between the axis of the piston and the line joining the point of observation and the center of the piston, and

λ = wavelength, in centimeters.

The upper frequency limit for this investigation will be placed at 4000 cps. The reason for selecting this limit is that the enclosures which will be used are relatively large. For example, the linear dimensions are eight to ten wavelengths at 4000 cps. It will be stipulated that the radiation from the cone of the loudspeaker mechanism at this frequency shall be down not more than 1.0 db for $\alpha = 90$ deg. as compared to $\alpha = 0$ deg. This insures a reasonably nondirectional sound source even at the upper end of the frequency range, that is, at 4000 cps. Of course, at lower frequencies the response discrepancy with respect to angle is much less. To satisfy the above requirements, the diameter of the diaphragm or cone must be ⅞ in. Accordingly a small direct-radiator loudspeaker mechanism employing a cone ⅞ in. in diameter was designed, built, and tested. A sectional view of the loudspeaker mechanism is shown in *Fig.* 1. Measurements indicated that the directional performance agreed with that predicted by equation (*1*).

The next consideration is the sound

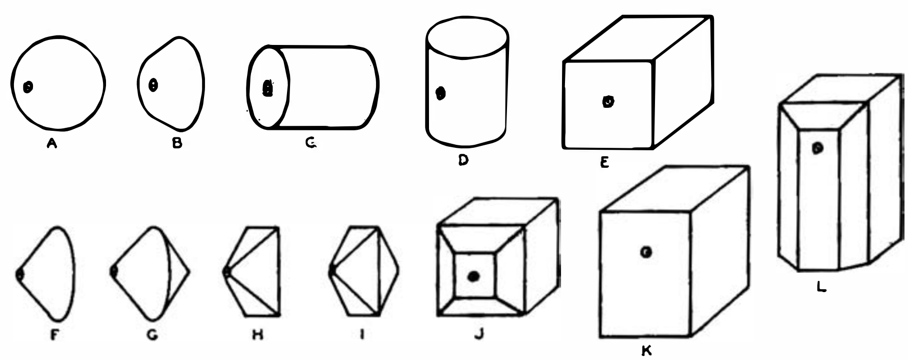


Fig. 2. Direct-radiator loudspeaker mechanism enclosures. The small circle with the dot in the center represents the speaker unit.

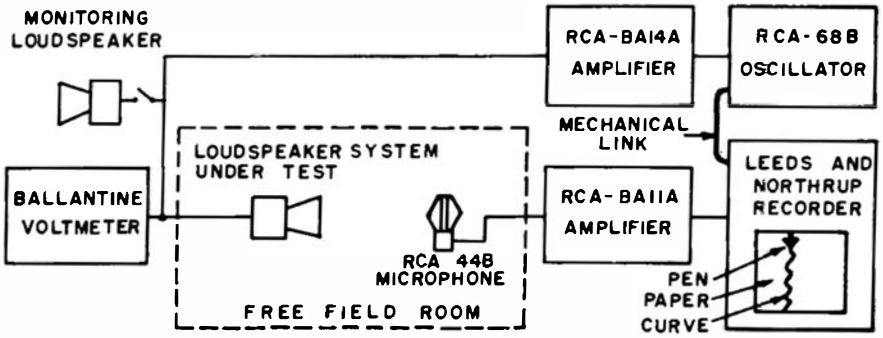


Fig. 3. Schematic diagram of the apparatus for obtaining the response-frequency characteristics of loudspeakers.

power output calibration of the sound source. The sound power output of a piston sound source radiating into 4π solid angles and operating in the frequency region in which the diameter of the piston is less than one-quarter wavelength[1] is given by

$$P_T = \frac{\rho\omega^2}{4\pi c} S^2 \dot{x}^2 = \frac{\rho\omega^2}{4\pi c} \dot{X}^2 \qquad (2)$$

where ρ = density of air, in gms./cu. cm.,
c = velocity of sound, in cm./sec.,
$\omega = 2\pi f$,
f = frequency, in cps,
S = area of the diaphragm, in sq. cm.,
$\dot{x}$ = r.m.s. velocity of the diaphragm, in cm./sec., and
$\dot{X}$ = r.m.s. volume current produced by the mechanism, in cu. cm./sec.

Equation *(2)* shows that the sound power output P_T of the sound source will be independent of the frequency f, if the velocity $\dot{x}$, of the piston is inversely proportional to the frequency. The characteristics depicted in this paper have been reduced to a sound source of this type, namely, that when it radiates into 4π solid angles the sound power output will be independent of the frequency. Since the directivity pattern of the sound source is independent of the frequency, the sound pressure, under these conditions, will also be independent of the frequency.

It may be mentioned in passing that, in the case of a direct-radiator loudspeaker mechanism operating in the frequency range below the ultimate acoustical radiation resistance, the velocity of the cone must be inversely proportional to the frequency in order to obtain constant sound power output, because the acoustical radiation resistance is proportional to the square of the frequency. In order to obtain this type of motion, the system must be mass controlled, which is the natural state of affairs in the direct-radiator type of loudspeaker mechanism above the fundamental resonant frequency of the system. In other words, the performance characteristics depicted in this paper are, for all practical purposes, the characteristics which will be obtained if conventional direct-radiator loudspeaker mechanisms are used in these enclosures.

[1] If the upper frequency limit is placed at 4000 cps, the diameter of the 7/8-in. cone will be less than one-quarter wavelength in the frequency range below 4000 cps.

Fig. 4. Equipment set-up for obtaining the response-frequency characteristics of loudspeakers.

Enclosures

The enclosures used in these experiments are depicted in *Fig. 2*. The sheet metal sphere shown at (A) is 2 ft. in diameter. The loudspeaker mechanism is mounted with the cone approximately flush with the surface. The sheet metal hemisphere shown at (B) is 2 ft. in diameter with the back closed by a flat board of hard wood. The loudspeaker mechanism is mounted upon the zenith of the hemisphere with the cone of the loudspeaker mechanism approximately flush with the surface. The sheet metal cylinder shown at (C) is 2 ft. in diameter and 2 ft. in length. The ends of the cylinder were closed by plywood boards of hard wood. The loudspeaker mechanism is mounted in the center of one end with the cone of the loudspeaker mechanism mounted flush with the surface. The cylinder shown at (D) is of the same size as that of (C). In (D) the cone of the loudspeaker mechanism is mounted approximately flush upon the cylindrical surface midway between the ends. The sides of the wood cube shown at (E) are 2 ft in length. The loudspeaker mechanism is mounted in the center of one face with the cone flush with the surface. The base of the sheet metal cone shown at (F) is 2 ft. in diameter. The height of the cone is 1 ft. The base of the cone is closed by

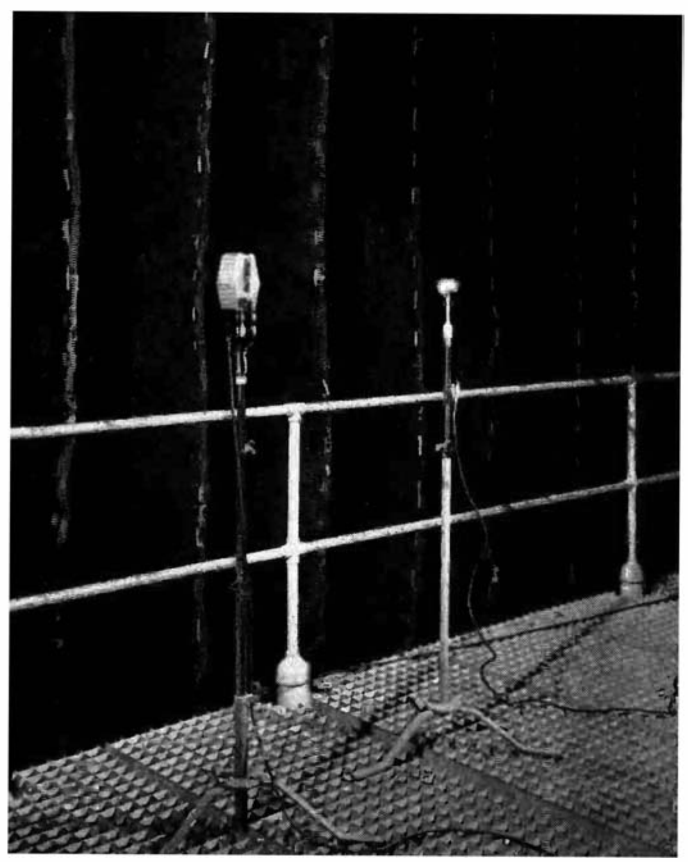


Fig. 5. Microphone and small direct-radiator loudspeaker mechanism of Fig. 1 under test in the free-field room.

a board of hard wood. The loudspeaker mechanism is mounted in the apex of the cone. The cone was truncated to accommodate the small loudspeaker mechanism. The double cone of (G) consists of two cones, each of the same size as that of the single cone of (F), with the bases placed edge to edge. The loudspeaker mechanism is mounted in the apex of one of the cones. The length of the edges of the square base of the wood pyramid shown at (H) is 2 ft. The height of the pyramid is 1 ft. The base of the pyramid is closed by a board of hard wood. The loudspeaker mechanism is mounted in the apex of the pyramid. The pyramid was truncated to accommodate the small loudspeaker mechanism. The double pyramid of (I) consists of two pyramids, each of the same size as that of the single pyramid of (H), with the bases placed edge to edge. The loudspeaker mechanism is mounted in the apex of one of the pyramids. The truncated pyramid of (J) is mounted upon a rectangular parallelepiped. The length of the edges of the truncated surface is 1 ft. The height of the truncated pyramid is 6 in. The lengths of the edges of the rectangular parallelepiped are 1 ft. and 2 ft. The loudspeaker mechanism is mounted in the center of the truncated surface. The lengths of the edges of the rectangular parallelepiped of (K) are 2 ft. and 3 ft. The loudspeaker mechanism is mounted midway between two long edges and 1 ft. from one short edge. At (L) a rectangular truncated pyramid is mounted upon a rectangular parallelepiped. The lengths of the edges of the rectangular parallelepiped are 1, 2, and 3 ft. The lengths of the edges of the truncated surface are 1 ft. and 2½ ft. The height of the truncated pyramid is 6 in. One surface of the pyramid and one surface of the parallelepiped lie in the same plane.

Measurement Apparatus and Techniques

The small loudspeaker mechanism of *Fig.* 1 was mounted in the enclosures shown in *Fig.* 2. In obtaining true diffraction effects it is important that reflection effects produced by room in which the response-frequency characteristic is obtained be reduced to a negligible minimum. Therefore, all the response-frequency characteristics depicted in this paper were obtained in the free field room[2, 3] of the Acoustical Laboratory of the RCA Laboratories. A schematic diagram of the apparatus used for obtaining the response-frequency characteristics, along with detailed designations of the components are shown in *Fig.* 3. The complete recording system—including the RCA-44B velocity microphone, BA1A amplifier, and Leeds and Northrup Speedomax recorder—was calibrated by the free

[2] H. F. Olson, *J. Acous. Soc. Am.*, Vol. 15, No. 2, p. 96, 1943.

[3] Olson, *Elements of Acoustical Engineering*, D. Van Nostrand Company, New York, 2nd Edition, 1947, p. 359.

FIG. 6

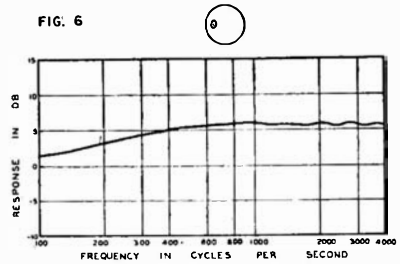


FIG. 7

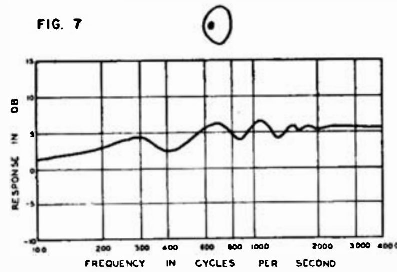


FIG. 8

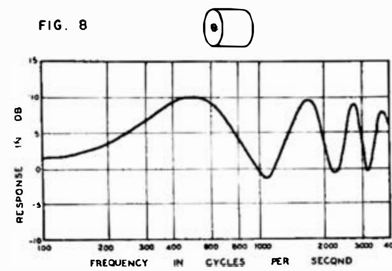


FIG. 9

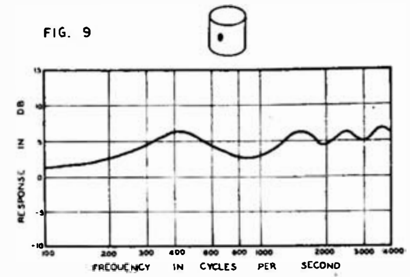


FIG. 10

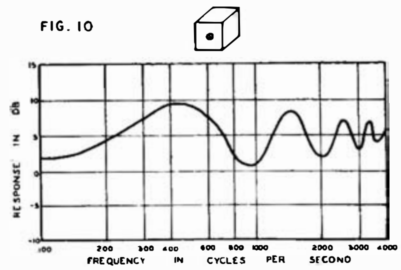


FIG. 11

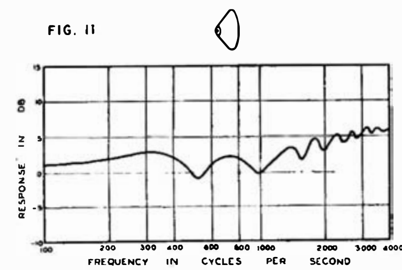


FIG. 12

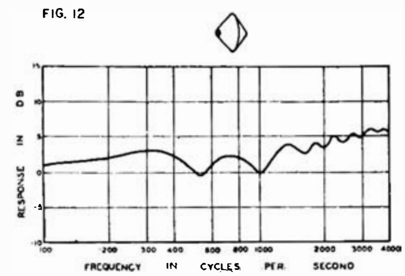


FIG. 13

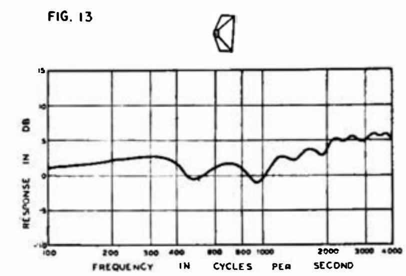


FIG. 14

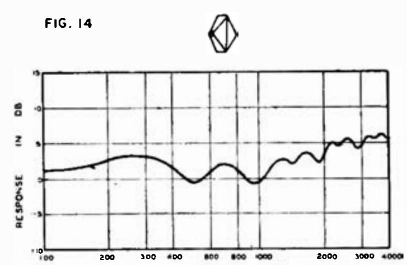


FIG. 15

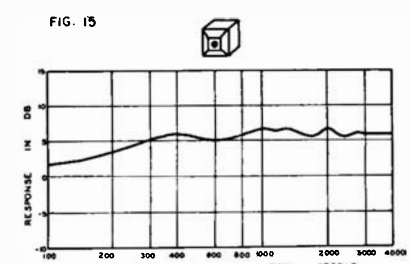


FIG. 16

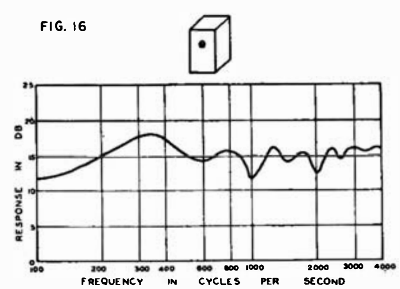
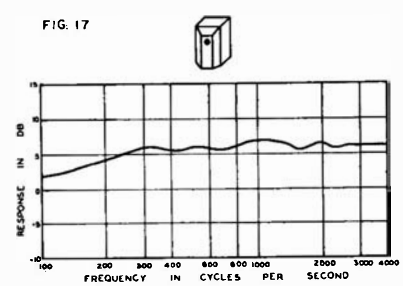


FIG. 17



Figs. 6 to 17. Response-frequency characteristic of a small direct-radiator loudspeaker mechanism mounted in the enclosures of Fig. 2.

field reciprocity method.[4, 5] A sinusoidal input was applied to the loudspeaker system under test by means of the combination RCA-68B beat-frequency oscillator and a BA-14A amplifier. The voltage applied to the loudspeaker system was measured by means of a Ballantine voltmeter. The measuring apparatus is shown in *Fig.* 4, and the microphone and the small loudspeaker mechanism of *Fig.* 1 under test in the free field sound room are shown in *Fig.* 5. The response-frequency characteristics illustrated and described in the sections which follow were obtained by means of the apparatus and arrangements described above.

Sphere

The first consideration will be the combination of the direct-radiator loudspeaker mechanism of *Fig.* 1 and the spherical enclosure as shown at (A) in *Fig.* 2. The axial response-frequency characteristic thus obtained was corrected so that the volume current produced by the mechanism was inversely proportional to the frequency, as previously described. The response-frequency characteristic[6, 7] of the combination of a small direct-radiator sound source in which the volume current is inversely proportional to the frequency and a large spherical enclosure is shown in *Fig.* 6. It will be seen that the response is uniform and free of peaks and dips. This is due to the fact that there are no sharp edges or discontinuities to set up diffracted waves of a definite phase pattern relation with respect to the primary sound emitted by the loudspeaker. The diffracted waves are uniformly distributed as to phase and amplitude. Therefore, the transition from radiation by the loudspeaker mechanism into 4π solid angles to radiation into 2π solid angles takes place uniformly with respect to the frequency. It will be noted that the sound pressure increases uniformly in this transition frequency. The ultimate pressure is 6 db higher than the sound pressure where the dimension of the sphere is a small fraction of the wavelength.

Hemisphere

The axial response-frequency characteristic of the loudspeaker mechanism of *Fig.* 1 mounted in the hemispherical enclosure of (B), *Fig.* 2, is shown in *Fig.* 7. The sharp discontinuity at the boundary of the spherical and plane surfaces produces a strongly diffracted wave. There is a phase difference between the primary and diffracted waves which results in peaks and dips in the response-frequency characteristic corresponding to in and out of phase relationships between the primary and diffracted sound. A physical explanation of the phenomena is as follows: The sound flows out in all possible directions from the sound source. The sound which follows the contour of the spherical surface encounters a sudden change in acoustical impedance at the intersection of the plane and spherical surface. A reflected wave is sent out at this point in all possible directions. The distance from the diaphragm of the loudspeaker mechanism to the circular diffracting edge is $\pi/2$ feet. The distance between the plane of the diaphragm and the plane containing circular diffracting edge is 1 ft. Therefore, the difference in path between the primary and the diffracted wave at the observation or measurement point on the axis is $(\pi/2+1)$ ft. The sound wave which follows the contour of the spherical surface encounters a decrease in acoustical impedance at the boundary of the spherical and plane surfaces, and the diffracted or reflected wave suffers a phase change of 180 deg. Therefore, when the distance $(\pi/2+1)$ ft. corresponds to odd multiples of one-half wavelength, there will be maxima of response because the primary and diffracted waves are in phase. The maxima will occur at 215, 645, 1075, etc. cps. It will be seen that this agrees with experimental results. When the distance $(\pi/2+1)$ ft. corresponds to multiples of the wavelength, there will be minima in the response because the primary and diffracted waves are out of phase. The minima will occur at 430, 860, 1290, etc. cps. It will be seen that this agrees with the experimental results.

Cylinder

The axial response-frequency characteristic[8] of the loudspeaker mechanism of *Fig.* 1 mounted in the center of one end of the cylinder of (C), *Fig.* 2 is shown in *Fig.* 8. The sharp boundary at the intersection of the plane and cylindrical surface introduces a strongly diffracted wave. The distance from the mechanism to the circular boundary is 1 ft. Therefore, since the diaphragm and the edge lie in the same plane, the path difference between primary and diffracted wave is 1 ft. Following the explanation of the preceding section, there should be maxima of response at 550, 1650, 2750, 3850, etc. cps, and there should be minima of response at 1100, 2200, 3300, etc. cps. It will be seen that there is remarkable agreement with the experimental results of *Fig.* 8. It is also interesting to note that the variations in response are very great, being of the order of 10 db.

The axial response-frequency characteristic of the loudspeaker mechanism of *Fig.* 1 mounted in the cylindrical surface of the cylinder of (D), *Fig.* 2, is shown in *Fig.* 9. Again the sharp boundary between the cylindrical and the plane surfaces produces a diffracted wave. However, the path difference between the primary and diffracted wave is not confined to a single discrete distance. Therefore, the maxima and minima of response are not as pronounced as in the case of (C), as shown in *Fig.* 8. From the response frequency characteristic of *Fig.* 9, it would appear that the effective distance between the primary and diffracted wave is about 1.17 ft. As would be expected, this means that the forward portion of the diffracting edge plays the predominant part.

Cube

The axial response-frequency characteristic[9] of the loudspeaker mechanism of *Fig.* 1 mounted in the center of one of the faces of the cube (E), of *Fig.* 2, is shown in *Fig.* 10. The sharp boundary at the edges of the cube produces a strongly diffracted wave. The average path between the mechanism and the

[4] H. F. Olson, *RCA Review,* Vol. 6, No. 1, p. 36, 1941.

[5] Olson, *Elements of Acoustical Engineering,* D. Van Nostrand Company, New York, 2nd Edition, 1947, p. 345.

[6] The response-frequency characteristics depicted in this paper were obtained on enclosures having the dimensions given. The response-frequency characteristics for enclosures of other dimensions can be obtained by multiplying the ratio of the linear dimensions of the enclosure given in this paper to the linear dimensions of the new enclosure by the frequency of the response-frequency characteristic given in this paper. For example: if the linear dimensions of the new enclosures are two times those of the enclosures described, the frequency scales of *Figs.* 6 to 17 inclusive should be multiplied by one-half.

[7] The theoretical and experimental sound pressures on a sphere as a function of the frequency for an impinging plane wave of constant intensity have been investigated by G. W. Stewart, *Phys. Rev.,* Vol. 33, No. 6, p. 467. 1911, S. Ballantine, *Phys. Rev.,* Vol. 32, No. 6, p. 988, 1928 and Muller, Black and Dunn, *J. Acous. Soc. Am.,* Vol. 10, No. 1, p. 6, 1938. The results reported by these investigators agree with those depicted in *Fig.* 6. This is to be expected from the reciprocity theorem which states that under appropriate conditions the source and observation points may be interchanged without altering the response frequency characteristics of the system. See Olson, *Elements of Acoustical Engineering,* D. Van Nostrand Company, New York, N. Y., 1947, p. 21.

[8] The theoretical and experimental sound pressures on the center of the face of a cylinder as a function of the frequency have been investigated by Muller, Black and Dunn, *J. Acous. Soc. Am.,* Vol. 10, No. 1, p. 6, 1938. The results reported by these investigators agree with those depicted in *Fig.* 8. This is to be expected from a consideration of the reciprocity theorem. See footnote 7.

[9] The theoretical and experimental sound pressures on the center of a face of a cube as a function of the frequency have been investigated by Muller, Black and Dunn, *J. Acous. Soc. Am.,* Vol. 10, No. 1, p. 6, 1938. The results reported by these investigators agree with those depicted in *Fig.* 10. This is to be expected from a consideration of the reciprocity theorem. See footnote 7.

edges is about 1.2 ft. Therefore, since the diaphragm and the edges lie in the same plane, the path difference between the primary and diffracted waves is 1.2 ft. Following the explanations of the preceding sections, there should be maxima of response at 460, 1380, 2300, 3200, etc. cps, and there should be minima of response at 920, 1840, 2760, etc. cps. There is reasonably good agreement with the experimental results of *Fig.* 10.

Cone

The axial response of the loudspeaker mechanism of *Fig.* 1 mounted in the apex of the cone, (F) of *Fig.* 2, is shown in *Fig.* 11. The sharp boundary at the base of the cone produces a diffracted wave. The distance from the mechanism to this edge is 1.3 ft. The distance between the plane of the diaphragm of the mechanism and the plane of the base is 0.95 ft. Therefore, the difference in path between the primary and diffracted waves is 2.25 ft. Following the explanations of the preceding sections, there should be maxima of response at 250, 750, 1250, etc. cps. and there should be minima of response at 500, 1000, 1500, 2000, etc. cps. There is very good agreement with the experimental results of *Fig.* 11. Another interesting fact is that the average magnitude of the response does not increase as rapidly with frequency as in the case of the examples in the preceding sections. This is due to the fact that the free space subtended by the loudspeaker mechanism is 2.6 steradians as compared to 2π steradians for most of the other systems considered in the preceding sections. Therefore, the ultimate sound pressure occurs at a higher frequency than in the case of enclosures in which the loudspeaker subtends 2π steradians.

The axial response of the loudspeaker mechanism of *Fig.* 1 mounted in the apex of the double cone, (G) of *Fig.* 2, is shown in *Fig.* 13. The sharp boundary at the bases of the cones produces a diffracted wave. The phase differences between the primary and diffracted waves are the same as those of the single cone. The performances of the single and double cone are about the same, as will be seen by comparing *Figs.* 11 and 12.

Pyramid

The axial response of the loudspeaker mechanism of *Fig.* 1 mounted in the apex of the pyramid, (A) of *Fig.* 2, is shown in *Fig.* 13. The sharp boundary at the base produces a diffracted wave. The average distance from the mechanism to this edge is 1.6 ft. The distance between the plane of the diaphragm of the mechanism and the plane of the base is 0.95 ft. Therefore, the difference in path between the primary and diffracted waves is 2.55 ft. Following the explanations of the preceding sections, there should be maxima of response at 220, 660, 1100, etc. cps, and minima at 440, 880, 1320, etc. cps. There is very good agreement with the experimental results of *Fig.* 13. The shape of the response-frequency characteristics is similar to that of the cone of a preceding section. As in the case of the cone, the ultimate response occurs at a relatively high frequency.

The axial response of the loudspeaker mechanism of *Fig.* 1 mounted in the apex of the double pyramid, (I) of *Fig.* 2, is shown in *Fig.* 14. The sharp boundary at the base of the pyramid produces a diffracted wave. The phase differences between the primary and diffracted waves are the same as those of the single cone. The performance of the single and double cone are about the same, as will be seen by comparing *Figs.* 13 and 14.

Truncated Pyramid and Rectangular Parallelepiped Combination

From the preceding examples, it will be seen that wide variations in the response-frequency characteristics occur when there is a sharp boundary or edge upon the surface of the enclosure which produces a strongly diffracted wave. The diffracted wave is further accentuated when all paths from the mechanism to the boundaries or edges are the same. The truncated pyramid and rectangular parallelepiped combination shown at (J) in *Fig.* 2 is designed with the object of reducing sharp boundaries on the front portion of the enclosure. Furthermore, the distances from the mechanism and the edges are not all the same. The response frequency characteristic of the loudspeaker mechanism of *Fig.* 1 mounted in the enclosure (J) is shown in *Fig.* 15. It will be seen that the response is quite uniform and free of large maxima and minima. This bears out the idea that the reduction of sharp boundaries on the surface of the enclosure and the elimination of equal path lengths from these boundaries to the mechanism will yield smoother response frequency characteristics.

Rectangular Parallelepiped

The rectangular parallelepiped in all its possible variations in dimensions is the most common direct-radiator loudspeaker enclosure. One of the obvious reasons for this state of affairs is that this shape is the simplest to fabricate. This is unfortunate, because the rectangular parallelepiped produces diffraction effects which adversely modify the response-frequency characteristic of a direct-radiator loudspeaker mechanism. The response-frequency curve of *Fig.* 16 was obtained with the loudspeaker mechanism of *Fig.* 1 mounted in the rectangular parallelepiped of (K), *Fig.* 2. The pronounced minima in the response at 1000 and 2000 cps are due to shorter distances from the mechanism to the upper and side edges. The minimum in response at 500 cps is due to the longer distance from the mechanism to the lower edge. The variations in response, due to diffraction effects by the cabinet, are of the order of 6 to 7 db. The response frequency characteristic of *Fig.* 16 is typical of the response obtained with this type of enclosure. Therefore, this cabinet shape is unsuitable for housing a direct-radiator loudspeaker mechanism, because of the wide variations in response produced by diffraction from the sharp edges of this cabinet.

Rectangular Truncated Pyramid and Parallelepiped Combination

From the data given the preceding sections is is possible to devise many cabinet shapes which will reduce the effects of diffractions in modifying the response frequency characteristics of the loudspeaker mechanism.

An example of the application of the principles outlined in this paper is shown at (L) in *Fig.* 2. In this cabinet the diffraction effects have been ameliorated by the reduction of abrupt angular discontinuities on the surface of the cabinet and the elimination of equal paths from these discontinuities to the mechanism. At the same time a practical exterior configuration has been retained which is not undesirable from an esthetic standpoint. The response-frequency characteristic of the loudspeaker mechanism of *Fig.* 1 mounted in the enclosure (L) is shown in *Fig.* 17. It will be seen that the response-frequency characteristic is quite smooth.

Conclusions

The response-frequency characteristics, which depict the performance of a direct-radiator loudspeaker mechanism in various enclosures of fundamental shapes, show that the outside configuration plays an important part in determining the response as a function of frequency. For example, in some of the enclosures the variation in response produced by diffraction exceeds 10 db.

All of the response-frequency characteristics depicted in this paper were taken on the axis of the loudspeaker mechanism and enclosure combination. In this connection, it should be mentioned that the variations in response are mitigated for locations off the axis. The reason for using the axial response is that the reference response-frequency characteristic of a direct-radiator loudspeaker is always taken on or near the axis. Practically all serious listening to direct-radiator loudspeakers is carried out on or near the axis.

The response of a loudspeaker in an enclosure will be modified by the directivity pattern of the mechanism, because the diffraction effects are influenced by the direction of flow of sound energy from the diaphragm. However, the performance in the frequency range in which the dimensions of the cone are less than a wavelength will not be markedly different.

The experiments described in this paper show that the deleterious effects of diffraction can be reduced by eliminating all sharp boundaries on the front portion of the enclosure upon which the mechanism is mounted, so that the amplitude of the diffracted waves will be reduced in amplitude and by making the distances from the mechanism to the diffracting edges varied so that there will be a random phase relationship between the primary and diffracted sound waves.

Measurement of the Propagation of Sound in Fiberglas

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As an aid to acoustical designers, this paper presents some experimentally determined values of the attenuation coefficient and the speed of propagation of sound in Fiberglas. The propagation parameters α and β in the equation for a plane wave $p=p_0\epsilon^{j\omega t-(\alpha+j\beta)x}$ are determined by comparing the amplitudes and phases of the sound pressures existing simultaneously at separate points in the Fiberglas. The principle of measurement is similar to that used by Beranek and Scott involving a longitudinal movable probe in a long narrow metal duct filled with rockwool. However, in the redesigned apparatus, edge effects were reduced by using a tube of larger cross section; wall vibrations were damped by laminated rubber and Masonite walls; and the shunting effect of the longitudinal probe hole was eliminated by the use of transversely inserted microphones.

Experimental data were obtained within one db for frequencies between 50 and 1000 cycles for two samples of Fiberglas with nominal weights of 9 lb/cu ft PF (hard) and $4\frac{1}{4}$ lb/cu ft TWF (soft). The direction of propagation of the sound was normal to the surface of the blanket. For both samples the attenuation increases almost as the square root of the frequency, but the speed of propagation is found to increase almost as the $\frac{1}{4}$ and $\frac{1}{3}$ root of the frequency for the hard and the soft samples, respectively.

INTRODUCTION

THE design of acoustical systems employing homogeneous porous absorbing material requires data on the characteristic impedance and the propagation parameters of sound in the material. These quantities can be used in determining the amplitude and phase of waves reflected from or transmitted through the material. This paper is concerned with the measurements of the propagation parameters only: the attenuation coefficient and the velocity of sound waves in Fiberglas at low audiofrequencies at room temperature.

The principle of these measurements is similar to that used by Beranek[1] and by Scott.[2] To a first approximation, the sound is assumed to propagate through the Fiberglas as a single plane wave having the attenuation and phase parameters α and β defined by the following equation for the sound pressure

$$p=p_0\epsilon^{j\omega t-(\alpha+j\beta)x}.$$

The parameters α and β are determined directly by comparing the amplitudes and phases of sound pressures existing simultaneously at different points in the Fiberglas.

APPARATUS

The apparatus containing the Fiberglas, as shown in Fig. 1, consisted of a square duct 6 inches on a side and 8 feet long with a loudspeaker mounted on one end and 17 microphone ports cut in the removable upper side. The sound pressures were measured simultaneously by two rochelle salt microphones thrust transversely into the Fiberglas through these circular ports sealed with rubber stoppers. A wave analyzer and an axis-crossing type phase meter were used to measure the amplitude and phase angles of the microphone outputs.

This apparatus differs from its predecessors[1,2] in the precautions taken to: (1) damp the vibrations of the containing duct; (2) reduce the edge effects; (3) eliminate the shunting due to the longitudinal hole for the probe; and (4) avoid the difficulties of inserting and handling a long probe. The details follow. To reduce the resonant vibrations of the duct, the laminated walls were made of 5 dense Masonite sheets bonded with rubber cement which acted in shear to dampen flexural vibrations. The speaker housing was attached to the duct by a sponge rubber gasket one inch thick and four heavy rubber bands which reduced the transfer of vibrations from the speaker housing to the duct walls. The effects of floor vibrations were reduced by putting the duct and the speaker housing on a one inch thick layer of sponge rubber.

As the Fiberglas is fairly stiff, the friction with the duct walls impedes the motion of the fibers and further stiffens the skeleton near the walls. To decrease the diaphragm effect due to clamping, the duct cross section was enlarged at the expense of losing data at high frequencies.

The stiffness of the Fiberglas prevented the easy insertion or the sealed withdrawal of a probe. To prevent leaks and to avoid the long probe, the microphones themselves were inserted directly into the sound field through the rubber stoppers. When the microphone was withdrawn, the cut hole was refilled with Fiberglas. The microphone leads and unused ports were sealed with rubber stoppers fitted with Masonite disks. To prevent the sound from reaching their flexible back covers, the microphones were mounted in heavy brass cases two inches in diameter sealed with insulating rubber gaskets. Open-mesh screens protected the faces of the microphones. In order to decrease the disturbance due to the presence of the microphones, the minimum

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[1] L. L. Beranek, J. Acoust. Soc. Am. 19, 420–427 (1947).
[2] R. A. Scott, J. Acoust. Soc. Am. 58, 165–183 (1946).

TABLE I. Experimental results.

	9 lb/cu ft PF Fiberglas (hard sample)	4½ lb/cu ft TWF Fiberglas (soft sample)
Dc flow resistance	36 400 newton sec/meter^4	13 800 newton sec/meter^4
Compressibility of skeleton	1.32×10^5 newtons/meter^2	2.5×10^4 newtons/meter^2
Density	135 kg/m^3	63 kg/m^3
Speed of propagation	32 $f^{0.26}$ meters/sec	23.5 $f^{0.34}$ meters/sec
Attenuation constant α	0.50 $f^{0.48}$ nepers/meter	0.39 $f^{0.48}$ nepers/sec
Phase constant β	0.20 $f^{0.74}$ radians/meter	0.27 $f^{0.66}$ radians/meter
Attenuation in db/meter	4.3 $f^{0.48}$ db/meter	3.4 $f^{0.48}$ db/meter
Porosity	0.965	0.985

f = frequency in cycles/sec.

projected area of the microphones—2.7 square inches—was obtained by orienting the planes of their diaphragms at right angles to the plane of the wave fronts.

Standing waves inside the duct were made negligible by the losses in the Fiberglas which attenuated the waves travelling toward and reflecting from the free end of the duct. Although cross-modes would exist above 1100 cycles in the empty duct, the high attenuation in the Fiberglas damped them out.

EXPERIMENTAL PROCEDURE

After the 6-in. squares of 1-in. Fiberglas had been stacked face-to-face in a single column in the duct, the apparatus was ready for operation. During the measurements, one microphone A remained fixed in the first port (nearest to the source), while the other one B was moved successively to ports farther from the source. Phase difference and relative amplitude were determined for each position of microphone B. The two microphones and their amplifiers were compared by placing the microphones A and B in turn in the first port in the duct while the electrical input to the speaker was being used as the reference.

As only the relative amplitudes and phases were required, the characteristics of the measuring apparatus, which remained the same for each pair of observations, canceled out. Consequently, slow variations of gain and phase shift with frequency in the amplifiers and microphones could be tolerated. In order to watch for distortion at the high input sound pressure of about 110 db above 0.0002 dyne per square centimeter, the wave shape was continuously monitored with an oscilloscope.

Low frequency noise intensified by room resonances limited the phase-meter observations to the two feet of Fiberglas nearest the source. The noise level was about 60 db below the output of the speaker. When an electronic filter was used to increase the signal-to-noise ratio, it was then found that anomalies began to appear near the middle of the duct. Measurements made with a vibration pickup showed these anomalies to be due to wall vibrations which were now of the same order of magnitude as the attenuated sound coming directly from the source through the Fiberglas. Consequently, only the measurements made within two feet of the source were accepted.

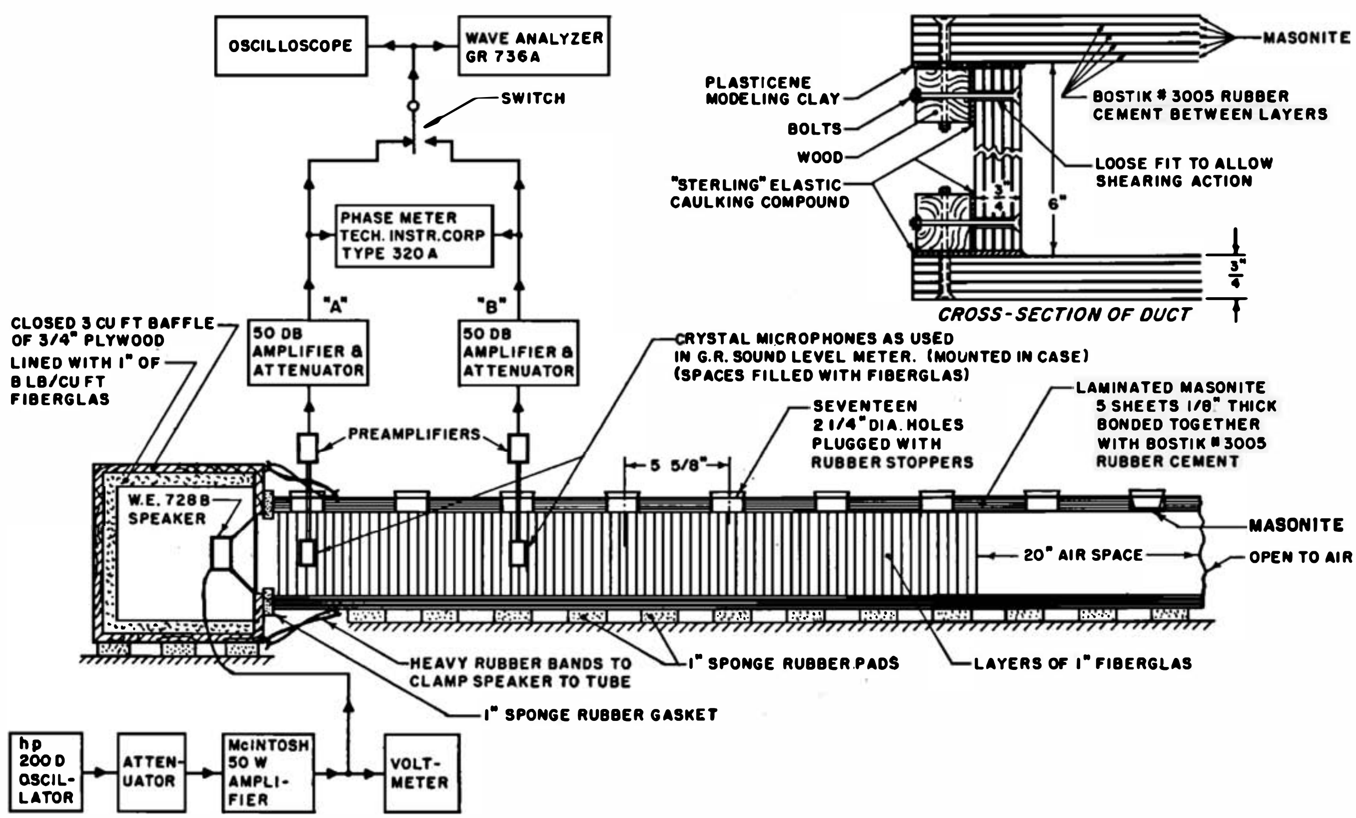

Fig. 4. Arrangement of the duct and measuring apparatus. The insert shows the details of the corners of the duct.

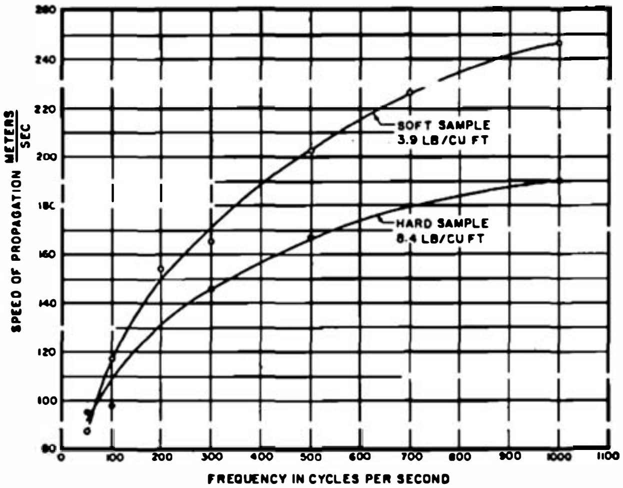

FIG. 2. The speed of sound in Fiberglas as a function of frequency for propagation normal to the blanket surface.

ERRORS AND DIFFICULTIES

The 1-db total error in reproducing relative measurements was due mainly to variations in the degree of packing of the space around the microphone (0.7 db), the uncertainty in the exact positions of the microphones (0.25 db), and drifts in the apparatus between readings. The method of measurement involves the following difficulties and their effects: (1) Leakage between the edges of the Fiberglas and the duct walls probably will decrease α and β. (2) Friction at the walls will tend to stiffen the Fiberglas to produce clamped-edge diaphragms. (3) Mechanical vibration of the microphone due to contact with the Fiberglas may introduce a variable quantity into the output of the microphone. (4) Vibrations of the duct walls limit the range of amplitude measurements to 40 db for the hard sample and to 60 db for the soft sample. (The difference in range is probably due to the tighter coupling of the stiffer Fiberglas to the duct walls.) (5) The presence of the microphones introduces discontinuities into the field (net increase of 0.2 db).

CONCLUSION

Tests were made on two samples of Fiberglas in the range from 50 to 1000 cps. The properties of these two samples determined by regular methods[3] and the results of our measurements at normal incidence are listed in the table.

As it can be seen from the table, the attenuation coefficient α (and consequently the attenuation in db/meter) in both samples vary almost as the square root of the frequency. However, in order to obtain the same attenuation at one kilocycle, the thickness of the soft sample should be almost 1.5 of the hard sample. The velocity-frequency relationship is manifested differently in that it varies as the $\frac{1}{4}$ and $\frac{1}{3}$ root of the frequency

[3] L. L. Beranek, *Acoustic Measurements* (John Wiley and Sons, Inc., New York, 1949).

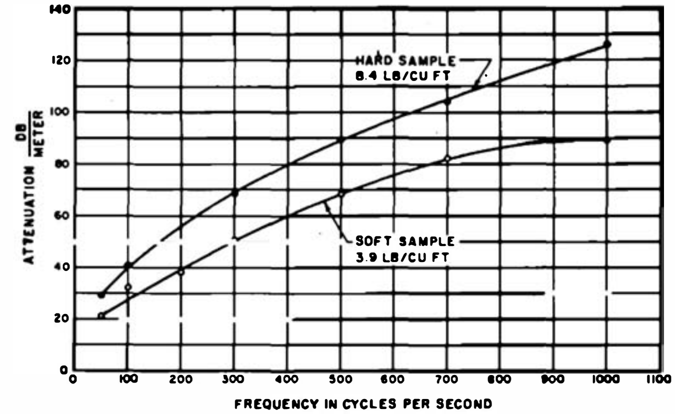

FIG. 3. The attenuation of sound in Fiberglas as a function of frequency for normal propagation.

for the hard and soft samples respectively. Figures 2, 3, and 4 show graphs of speed, attenuation, and the parameters α and β as functions of frequency for the two samples of Fiberglas.

It should be noted that the simplified presentation of the experimental data may mask the detailed mechanism of the energy transfer in the Fiberglas. Recent papers[4–7] indicate that the relationships are much more complex.

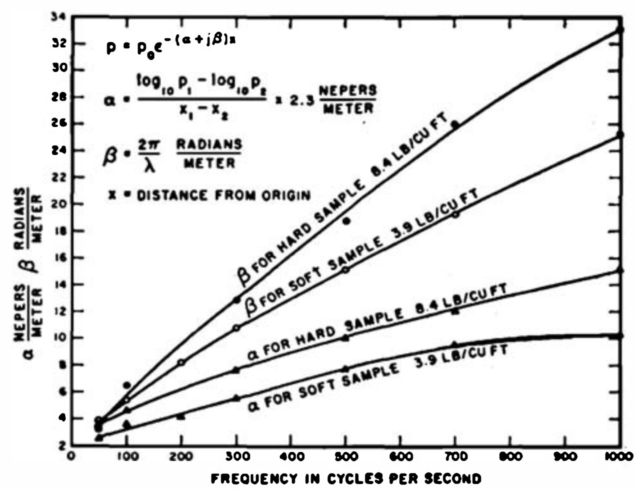

FIG. 4. The propagation parameters α and β plotted against frequency for sound travelling normally to the surface of the Fiberglas blanket.

ACKNOWLEDGMENT

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[4] P. M. Morse and R. H. Bolt, Revs. Modern Phys. **16**, 69–150 (1944).
[5] L. L. Beranek, J. Acoust. Soc. Am. **19**, 556–568 (1947).
[6] C. Zwikker and C. W. Kosten, *Sound Absorbing Materials* (Elsevier Publishing Company, New York, 1949).
[7] J. W. McGrath, J. Acoust. Soc. Am. **24**, 305–309 (1952).