# Design of the Wide-Range Ultra-Compact Regal Speaker System

ROBERT C. AVEDON,
WAYNE KOOY, and
JACK E. BURCHFIELD*

**Since small loudspeakers have become particularly popular for stereo installations, it is only natural that different concepts of design would be reached by different manufacturers; all, however, directed toward achieving a high degree of performance in a minimum of volume.**

In recent months the ultra-compact loudspeaker system has become popular where space limitations must be met. Most people are reluctant to sacrifice sound quality for extreme compactness. The ultra-compact loudspeaker must preserve much of the sound quality available in present day larger systems in order to be acceptable for high fidelity use. Unfortunately, many misconceptions exist concerning this type of speaker and include matters of efficiency, size, distortion, and a variety of constructional details.

The ultra-compact cabinet has one big advantage: small size. However, no diminutive speaker system can perform *because* of its size. On almost every point of performance the small cabinet speaker is at a disadvantage. These performance problems must be solved on a compromise basis.

It is the purpose of this article to dispel the present misconceptions and to arrive at the optimum design requirements for the ultra-compact loudspeaker.

The reader is invited to follow a series of experiments and to participate in the arrival at the optimum solutions that are found in the Electro-Voice Regal.

* *Engineers, Electro-Voice, Inc., Buchanan, Michigan.*

#### Bass Response and Efficiency

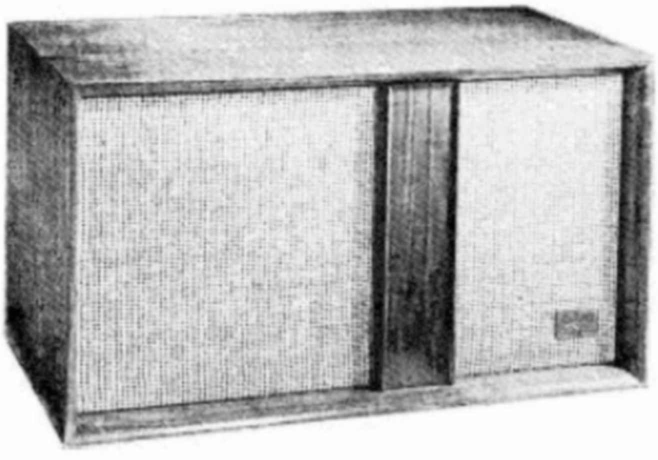
The well known problem encountered in preserving high fidelity sound quality in a very compact enclosure is achieving a flat bass range with reasonable efficiency. The objective treatment below will illustrate the bass-range problem.

A cabinet which was built for the experiments enclosed 2500 cubic inches or about 1.4 cubic feet, excluding the volume taken up by drivers and crossover networks. This volume is the generally accepted size for bookshelf type speakers.

All frequency-response curves were machine run with the speaker placed in a free field corner. The free field corner consisted of 8 foot high false walls built on a flat roof. These walls extended 10 feet in either direction from the corner.

A conventional dynamic 12-inch cone driver of high quality having a free air resonance of 39 cps was "infinitely" baffled in the 1.4 cubic foot cabinet. (A free air resonance of 35 to 45 cps is representative of a 12-inch driver of this type.) The primary or first resonance of the system occurred at 88 cps. When swept with an oscillator at constant voltage the acoustic output began to fall with decreasing frequency below 88 cps, becoming at 40 cps about 12 db down from the nominal output above 88 cps. Obviously, a system which reaches to only 88 cps before its output begins to diminish cannot be acceptable for high fidelity reproduction because of lack of musical balance.

How, then, is the bass response to be extended flat below 88 cps in this experiment? The most obvious attack (and, in fact, the very crux of the matter when dealing with sealed cabinet systems) is simply to lower the first resonance of the system. Unfortunately, simply lowering the system's first resonance is a matter requiring consideration of sacrifice in efficiency and transient response. A digression dealing with equivalent electrical circuits will show why efficiency and transient response must be sacrificed to accomplish lower bass range.



The Electro-Voice Regal III

#### Equivalent Circuits and the Bass Range

The electrical circuit analogies pertaining to the driver in free air and to the sealed cabinet system will now be examined.

The free-air resonance of a cone driver is determined by the mass of the entire moving assembly and its mechanical suspension compliance. The equivalent circuit of a cone unit, operating in free air, is shown at (A) in *Fig.* 1. $M_c$ represents the moving mechanical mass, $M_a$ represents the air mass load, and $C_s$ is the combined compliance of the suspension (spider and rim rolls). $R_r$ represents the radiation resistance component of the air load as seen in the mechanical circuit, and $R_m$ represents any mechanical resistances in the moving assembly. This circuit shows an impedance at resonance of just the radiation resistance plus the mechanical resistance. At this frequency the cone velocity will be maximum, limited only by this radiation and mechanical resistance.

When the driver unit is baffled in a small sealed volume the mechanical system has the equivalent circuit shown at (B) in *Fig.* 1. Because the rear of the driver piston is now enclosed, front to back cancellation is eliminated, and the radiation resistance, $R_r$, is increased and appears wholly on the front of the cone. The radiation to the rear into a small cavity at frequencies in the bass range is essentially non-existent. $R_m$, the mechanical resistance of the moving assembly, remains the same. $M_a$, the air mass load, is increased by virtue of the presence of the baffle. But the most significant change as far as this discussion is concerned is the addition of another series compliance component, $C_a$. This addition is due to the sealed cavity behind the cone. Recall that the primary resonance was higher for the driver in the sealed cabinet than for free air operation. This is caused by the addition in series of the air cavity compliance, $C_a$, which affected a *reduction* in the total compliance seen by the driver.

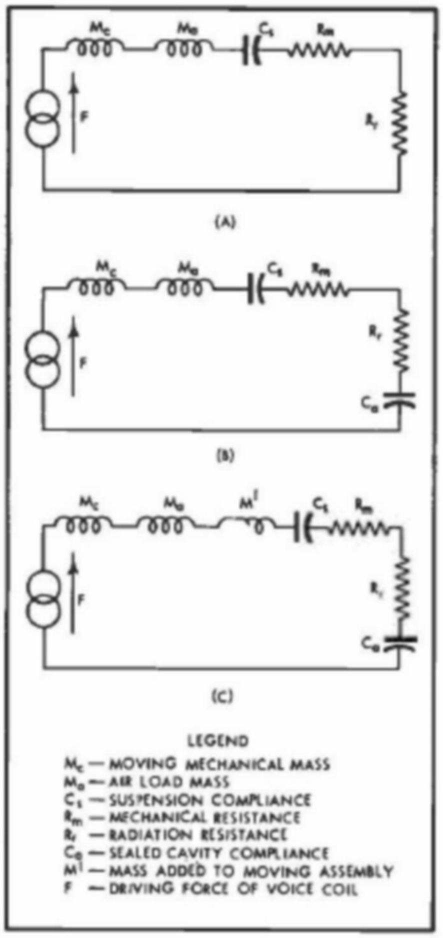



Fig. 1. Equivalent circuits: (A) Conventional cone driver in free air, and (B), "infinitely" baffled in small sealed cabinet; (C) Weighted cone driver "infinitely" baffled in small sealed cabinet.

Returning to the experimental system it will be seen that for a constant applied voltage on the driver voice coil the system will exhibit a flat energy output from resonance (88 cps) to approximately 1000 cps. Although the mechanical velocity of the piston is falling with increasing frequency above resonance at approximately 6 db per octave, the radiation resistance component, $R_r$, of the air load is rising 6 db per octave (taken as a power ratio). The radiation resistance is a function of frequency up to about 900 or 1000 cps for a nominal 12-inch cone, quadrupling each time the frequency is doubled. At this frequency it reaches an ultimate value and is constant for all higher frequencies provided the cone still operates as a rigid piston. But all 12-inch cones break up at approximately 1000 cps. Therefore, the considerations at present must be limited to frequencies below 1000 cps for the single degree of freedom circuits of *Fig.* 1 to be valid.

The falling cone velocity just complements the rising radiation resistance for frequencies between resonance and 1000 cps. Such a fortuitous combination of circumstances leads to flat acoustic energy output in this range. Below resonance, however, the cone velocity drops nearly 6 db per octave with decreasing frequency while the radiation resistance also drops 6 db per octave with decreasing frequency. Thus, in the frequency range below resonance a complementary situation between the radiation resistance and the cone velocity *does not exist* and the result is falling response below resonance at the rate of about 12 db per octave. The exact roll-off characteristic below resonance depends on the cone size and, hence, the relative value of the circuit parameters.

Now that it is clear exactly what problems exist and why the system's first resonance must be lowered, a discussion of resonance-lowering techniques can proceed.

#### Lowering the Resonance

It is contended by some manufacturers of ultra-compact speaker systems that when the conventional driver is baffled into a small sealed volume the total stiffness as seen by the driver piston is increased, resulting in greatly raised primary resonance for the system with bass response lacking below this resonance.

The previous discussion has shown this to be quite true. This school further contends: To lower the resonance and obtain a flat bass response the total stiffness can be reduced sufficiently by virtual elimination of the mechanical stiffness of the driver suspension, leaving the sealed cavity as the only stiffness. It is claimed also that elimination of the

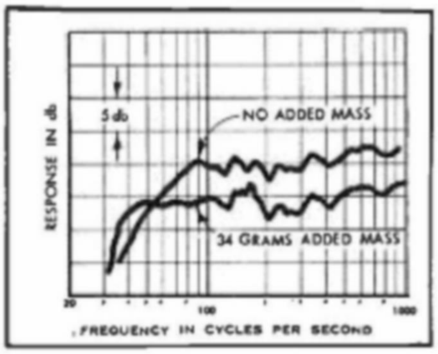



Fig. 2. Effect of added mass on cone driver "infinitely" baffled in small sealed cabinet.

mechanical suspension provides extreme linearity of cone movement due to the linear air spring of the enclosed cabinet volume as compared with the extremely non-linear mechanical cone suspension.

Are these presumptions correct? Further experiment will reveal that this is *not* correct.

Attention will now return to the experimental cabinet and the 12-inch driver used with it. A series of experiments were performed to determine how much the resonance could be lowered by *completely* removing the driver's mechanical suspension. The description of these experiments to follow will show how moving-mass, mechanical-compliance, and air-cavity-compliance data were collected and used to find the new resonance *without* a mechanical suspension.

The moving mass was found by the following method. Recall that the free air resonance of the driver was 39 cps. A metal ring serving as an added mass and weighing 28.2 grams was cemented firmly to the cone at its apex. The free air resonance was now observed to be 28.5 cps.

The equation shown describes resonance under these circumstances:

$$f = \frac{1}{2\pi\sqrt{MC}} \qquad (1)$$

where $f$ is the resonant frequency, $M$ is all the moving mass involved, and $C$ is the total compliance.

The new free air resonance obtained with the added mass in conjunction with the former free air resonance applied to Eq. (*1*) will allow computation of the mechanical suspension compliance and the total moving mass of the driver and air load.

Two independent equations will be necessary and are derived from Eq. (*1*).

$$f_1 = \frac{1}{2\pi\sqrt{MC}} \qquad (2)$$

$$f_2 = \frac{1}{2\pi\sqrt{(M+28.2)C}} \qquad (3)$$

where $f_1$ is the resonant frequency *without* added mass and $f_2$ is the resonant frequency *with* the added mass of 28.2 grams.

Simultaneous solution of Eq's. (*2*) and (*3*) for $M$ obtains the expression

$$M = \frac{(28.2) f_2^2}{f_1^2 - f_2^2} \qquad (4)$$

Substitution of the resonant frequencies yields for the total moving mass *including* air load: $M = 32.3$ grams.

It is noteworthy that the moving mechanical mass of the driver determined by weighing the moving assembly on an analytical balance is just 15.0 grams.

One concludes, then, that the air-load mass for free-air operation must be 17.3 grams. Rearranging Eq. (*2*) and substituting for the total moving mass a value of 32.3 grams along with the unweighted free-air resonant frequency of 39 cps, the mechanical compliance is found to be:

$$C = \frac{1}{4\pi^2 M f_1^2} = 0.514 (10^{-6}) \; \frac{cm.}{dyne} \qquad (5)$$

This means that the application of one dyne of force will move the cone about one-half millionth of a centimeter.

At this juncture there has been determined the total moving mass and the suspension compliance for the free-air case. The total moving mass and the total compliance (suspension plus air cavity) has also been determined at this time by the same method for the same driver sealed into the experimental cabinet. Recall that the primary resonance was 88 cps for this case (no added mass). With the same added mass of 28.2 grams the resonance dropped to 65 cps. By the same method of solution as for the free-air case the total moving mass was found to be 33.7 grams (an increase of 1.4 grams over the free-air case) and the *total* compliance was 0.097 $(10)^{-6}$ cm/dyne.

The compliance actually contributed by the cabinet air volume will now be found. From the equivalent circuit of (B) in *Fig.* 1 recall that the suspension compliance adds in *series* with the cabinet-air-volume compliance. The total compliance of two series compliances is *not* their arithmetic sum. But the total stiffness (reciprocal of compliance, $1/C$) *is* the arithmetic sum of two series stiffnesses. Then it can be said:

$$\frac{1}{C_{t(TOTAL)}} = \frac{1}{C_s} + \frac{1}{C_A} \qquad (6)$$

Now knowing the values of the suspension compliance, $C_s$, and of the total compliance, $C_t$, the compliance of the air volume is found to be 0.119 $(10)^{-6}$ cm/dyne by Eq. (*6*).

The reader can see that the total of two series compliances is always *less* than the smallest of either of the two compliances individually.

At this point a simple computation will be made to find the new resonant frequency obtainable, conceding that the mechanical compliance could be completely removed.

Return to Eq. (*1*) and calculate the resonant frequency of the driver with its unaltered mass of 33.7 grams sealed into the example cabinet volume of compliance 0.119. Notice that the mechanical compliance of the suspension has *not* been included.

Frighteningly enough, by completely eliminating the suspension compliance the resonance drops to 79.5 cps, a mere 8.5 cps decrease from the situation including the suspension compliance! Is this the improved result desired? Such a reduction in resonant frequency of only a little over 10 per cent is a piddling effort at best!

It is quite obvious that mere elimination of the suspension compliance is not enough to effect a large reduction in resonant frequency. Re-examination of the equivalent circuit of (B) in *Fig.* 1 and Eq. (*1*) will reveal the only other alternative to reducing the resonant frequency. Further lowering of resonance can be accomplished only by addition of mass. Here is where the supreme sacrifice in efficiency comes. Referring to (C) in *Fig.* 1 a mass is added in series in the form of additional weight of cone material (thicker cone) or say, a machined metal ring affixed to the voice coil form. It is seen plainly that this additional reactive element will reduce the velocity circulating through the radiation resistance, $R_r$, above resonance with consequent reduction in acoustic output for the same driving force. The driving force is produced by electrical current circulating through a voice-coil wire containing electrical resistance. Thus, with the same driving force the electrical losses remain the same but the acoustic output drops as mass is added to the moving system. Consequently, the over-all efficiency of the system drops.

It is worth noting that each time the moving mass is doubled the acoustic output above resonance drops 6 db for the same voice-coil current. If this is true, then the addition of exactly 33.7 grams to the moving assembly of the driver (in its original form "infinitely" baffled in the experimental cabinet) should reduce its output just 6 db. Recall that the total moving mass was previously determined to be 33.7 grams. The addition of a like mass of 33.7 grams reduced the output, on an average, about 6 db for frequencies above 100 cps in this case. The curves shown in *Fig.* 2 are reproduced from machine-run charts.

So it is seen that the decision the design engineer must make in the ultra-compact cabinet is to make an optimum compromise between loss of efficiency and lowered resonance. A resonance of 55 to 60 cps with an efficiency loss of 7 or 8 db would be an excellent optimum compromise and is the situation existing

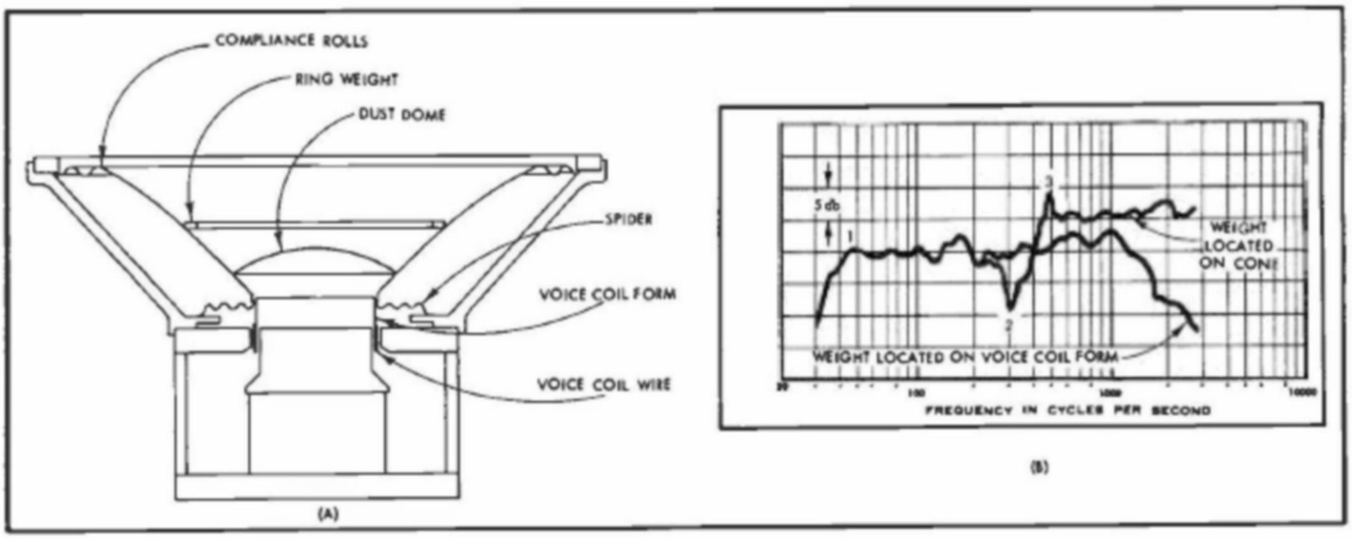



Fig. 3. Adding mass to cone driver moving assembly. (A) Cross section of driver showing ring weight added to cone. (B) Effects on frequency response of added weight location.

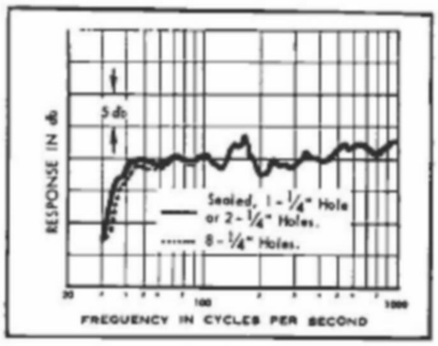



Fig. 4. Effects of cabinet leaks on frequency response.

in the Regal system. Because the resonance in this system shows up as a very broad effect in the acoustic output the level at 35 cps is down from the resonance level only 5 db measured in the free field corner.

How can the additional mass required to lower the resonance be added to the moving system? The problem in adding mass to the moving system is maintenance of a smooth frequency response which is uniform from speaker to speaker. A cone which is constructed so that it is heavier than a conventional cone can be used. Molded paper cones are made by placing forms in a slurry. It is difficult to maintain a uniform thickness when forming a heavy section by this method. A more precise way to add extra mass is just the way it was done in the experimental case discussed—in the form of a machined metal ring affixed to the cone and concentric with the voice coil. However, a problem arises in the placement of this extra mass.

#### Locating the Mass

In *Fig.* 3, (A) shows a ring of arbitrary diameter on the cone, and (B) shows the response of this driver in the experimental cabinet. Point 1 on the curve is the primary or first resonance. Point 2 is the frequency at which the ring weight goes into resonance with the effective compliance of the cone surrounding it. At this frequency, the voice coil sees an anti-resonant condition and a dip is produced in the response. Point 3 on the curve is the frequency at which the cone has "broken up" and the ring is nearly standing still while the voice coil and central cone area are in vigorous motion. This frequency is a new resonance for the moving system as seen from the voice coil. Here the effective compliance of the cone from ring to voice coil along with the spider compliance is in resonance with the distributed mass of the cone from ring to voice coil, the air mass load on this part of the cone, and the voice coil mass. As the ring is made of smaller diameter (but the mass held constant) and as it comes closer to the voice coil, points 2 and 3 move higher in frequency. When the ring is mounted directly to the voice coil form the system returns to a single degree of freedom and Points 2 and 3 on the curve disappear. Under this condition the system returns to its original operation up to 1000 cps with the exception that the level is lower due to the added mass.

Note the higher output above Point 3 in (B) of *Fig.* 3 with the weight placed somewhere out on the cone as compared with the output for the same frequencies with the weight attached to the voice-coil form. With voice-coil form mounting the weight never "decouples" and the system has always one degree of freedom up to the frequency where the cone would normally break up. With the weight mounted somewhere on the cone it goes through the process of decoupling between Points 2 and 3 in the curve. Above Point 3 the system operates with reduced mass with the ring and cone area outside the ring essentially inoperative. Hence, the higher output.

Another important consideration is that a cone has a lower effective mass above 1000 cps because of breakup. *All* of the cone does not move, and therefore the effective mass is less. This causes the difference in curve shape above 1000 cps between the original driver and the unit weighted at the voice coil form. When the cone breaks up a higher ratio of added mass to original moving mass exists. As a result the output above the breakup is reduced (by comparison to the original driver) in greater proportion than is the output below breakup. If the breakup frequency is adjusted by choice of cone geometry to coincide with the frequency of crossover into the next driver this phenomena can be put to good use by letting it aid the electrical networks in attenuating the output of the low-frequency driver above crossover.

#### Non-linearity and the Sealed Back Cavity

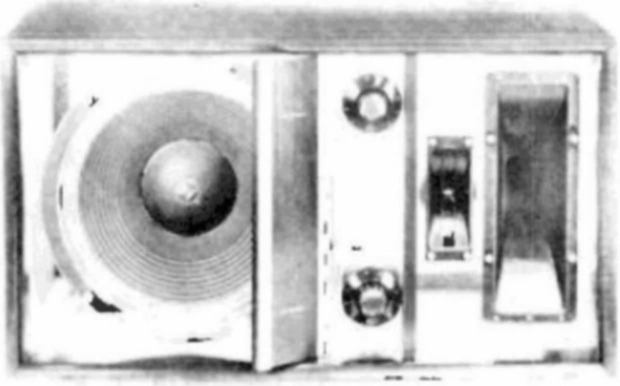
It has been contended that the mechanical suspension non-linearity is much greater than the extreme linearity of the air spring or sealed air volume of the cabinet. This is not true. In any good physics text one can find proof that sound is an adiabatic process. This means that when air is compressed or rarefied in the sealed air cavity behind the speaker there is no heat transfer to or away from the cabinet walls or interior components.

It is extremely difficult to make this process anything other than adiabatic. The physics text will also show that adiabatic compression is inherently non-linear. However, for small pressure variations relative to the ambient atmospheric pressure the non-linearity is vanishingly small. Consider a conventional 12-inch speaker in a 6-cubic-foot box. When undergoing the excursions required for satisfactory bass response the pressure variation relative to ambient atmospheric pressure is indeed small. However, when a 12-inch piston in an extremely small box undergoes these necessary excursions there are much larger pressure variations relative to ambient atmospheric pressure. These larger pressure variations must cause greater distortion for a given excursion of the cone due to the inherent non-linearity of the adiabatic process. This is an unavoidable consequence of the laws of physics. So when it is said air suspensions are inherently more linear than mechanical suspensions a misstatement has been made, for mechanical suspensions are often made that are more linear than these compact air springs.

#### Leaks in the Back Cavity

Need the cavity behind the driver be sealed absolutely air tight, resorting even to a stethoscope to detect minor air leaks? Even at the lowest usable frequencies the inductance of a small leak is high enough to prevent any loss in output. A test to prove this has been conducted in the following manner: A frequency response curve was run on a perfectly sealed cabinet containing a 12-inch loudspeaker. A small hole of a 1/4-in. diameter was drilled into the cabinet and another response curve was run. This process was repeated many times with more holes and more fre-

Rear view, Electro-Voice Regal III



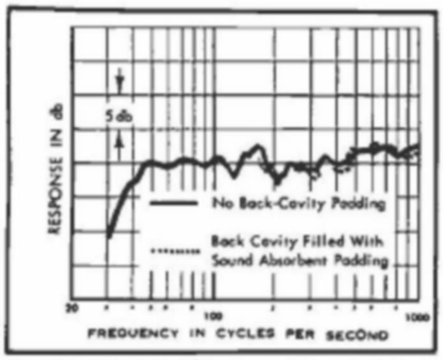



Fig. 5. Effect of filling small sealed-back cavity with sound absorbent material.

quency response curves. The results are shown in *Fig.* 4.

It is seen that nowhere was the output reduced by more than 1 db with a total of eight ¼-in. holes. This represents a substantial air leak. Therefore, a well crafted box with reasonable joinery is all that is necessary.

While on the subject of cabinet construction it might be noted that for a box of this size ¾" lumber stock is entirely adequate. No panel vibration of any consequence whatever will be experienced.

### Use of Absorbent Material in Small Cabinets

At this point mention should be made of filling the back cavity of ultra-compact cabinets with sound absorbent material. A test was conducted in the following manner: Curves were run on a sealed box system in the free field corner with and without the interior of the box filled with sound absorbent material. The two curves are shown in *Fig.* 5. With accurate recording equipment the results show a negligible difference between the two curves. Filling the interior of small cavities with sound absorbent material is unnecessary.

### Long-Throw Voice Coils vs. Efficiency

The throw of the voice coil is defined as the maximum excursion it can execute without distortion due to non-linearity of movement. At any particular frequency the excursion will be proportional to the voice-coil current as long as the same number of turns of the voice coil always remain in the dense flux field. In *Fig.* 6, (A) shows the voice-coil flux configuration for maximum efficiency. However, this arrangement will not be able to move without causing distortion because any excursion will remove turns from the gap; (B) shows a voice-coil arrangement where a constant number of turns will be maintained in the flux field with excursion. Twice the amount of overhang, *X*, will be the total excursion available without non-linearity. Overhang results in a loss of efficiency, and the greater the overhang the greater the loss of efficiency. Turns of the voice coil outside the dense flux field do not contribute to the driving force but do add d.c. resistance, an undesirable condition.

A compromise must be struck between linearity and efficiency. The maximum total excursion is dictated by the lowest usable frequency and the maximum practical listening level. The excursion is limited by the maximum tolerable distortion which is caused by the non-linearity of the air spring. This non-linearity is due to excessive pressure variation in the back cavity. For a 12-inch direct radiating speaker operating at 35 cps, ⅜-in. total excursion will provide room shaking level. The low-frequency driver in the Regal system employs a voice coil with a 3/16-in. overhang. This provides ⅜-in. total linear excursion. Any more linear excursion than this is unnecessary and results only in further loss of efficiency. The non-linearity of the air suspension overshadows any reduction in distortion derived from a throw longer than ⅜-in.

### Transient Response and Series Resistors

To increase the bass level it is sometimes recommended that a resistor in series with the system be used. This is not good, for under these conditions the benefits of the high damping factor in a quality amplifier are destroyed. Without sufficient damping the low-frequency driver at resonance is essentially "free wheeling" (undamped), resulting in a peaked output at primary resonance along with greatly increased transient distortion. In some ultra-compact systems the transient characteristics have already been degraded by the excessive addition of mass. Anything which further degrades the transient response is retrogressive.

Additionally, series resistance results in another loss of efficiency which cannot be afforded in the ultra-compact loudspeaker.

### The Mid and High Frequencies

The reader's attention is now directed to the mid and high frequencies. The following design features are considered desirable, and are especially effective for stereo reproduction: smooth response, good distribution, and low distortion of the mid-range and high frequencies. Excess efficiency in the treble and high-frequency drivers along with continuously variable controls allow the listener to compensate for room conditions and his own individual taste. Many compact systems provide a switched variation of only ± 2 db which is inadequate.

A horn-loaded radiator provides a good match of the diaphragm to the air load and more than enough level to balance the lower frequencies. If the horn has a small lateral dimension compared to the wavelength of the lowest frequency it is to reproduce, the dispersion will be excellent. The flat response curves frequently published do not tell the complete story unless dispersion or polar pattern is taken into account. It can be seen that a flat curve on the axis with progressive beaming at the higher frequencies would sound deficient in highs unless the listener were directly in front of the speaker. The dispersion of horns of the diffraction type is the best by far that is available and a smooth off-axis frequency response is easily obtained. A superior advantage of horn loading is that a small movement of the diaphragm will provide high acoustic output. This small movement allows completely linear operation of the diaphragm. If care is taken in loading the horn to the diaphragm, annoying dips and peaks found undesirable in many horn units are completely removed.

Some alternate efforts to achieve good high-frequency performance are inadequate. Attempts to promote smooth dispersed high frequencies by using a small dome radiating directly into the air have been tried. While these domes give good dispersion, they fail miserably in delivering efficiency, and low distortion. To illustrate, a unit was made from a 2-in. dome and a 1-lb. Alnico V magnet. The response was only fairly flat on axis and the distortion content averaged 20 times higher for the same acoustic output than that derived from a comparable horn-loaded unit. The efficiency was 15 db below the horn unit. This means that for the same output the diaphragm was moving many times further than in the horn unit. This extreme excursion made linear movement practically impossible

*(Continued on page 68)*

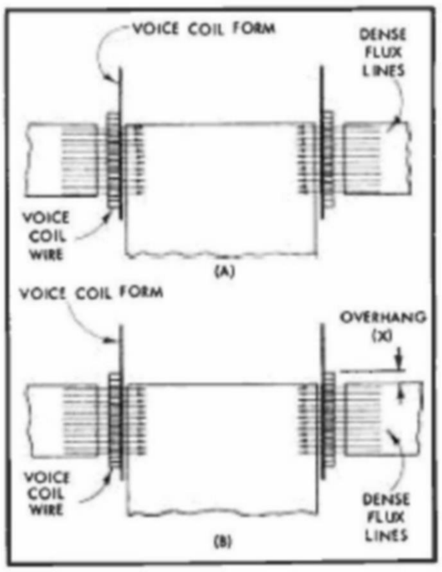



Fig. 6. Voice-coil geometry. (A) Voice coil no longer than dense flux field. (B) Voice coil overhanging the dense flux field.

# DESIGN OF WIDE RANGE SPEAKER SYSTEM

(from page 26)

at usual listening levels. It is retrogressive to tolerate unnecessary distortion in a speaker system when care and pains are taken to keep it out of the amplifier, cartridge, and records. To get enough output with direct radiating domes to match the low-frequency driver a great amount of magnet and iron structure must be used. All these problems can be overcome simply by the use of proper loading structures and horns. The high-frequency drivers in the Regal system employ the Sonophase throat design and exhaust into diffraction horns.

#### Power Requirements

Many ultra-compact systems have an EIA sensitivity rating of about 38 db. Conventional size enclosures using medium efficiency drivers have a sensitivity rating of about 50 db. This means that these compact systems must be 12 db less efficient than medium efficiency systems. To produce a given sound level these compact systems require 16 times the amplifier power. It is generally accepted that 20 watts is a minimum requirement for sufficient room levels and dynamic range with systems having EIA sensitivity ratings of 50 db. By this criterion a system which requires 16 times this power would need a 320-watt amplifier. This is a horrendous power requirement. Such an amplifier is not available for home use.

Most manufacturers of ultra-compact speakers recommend a 50 to 70 watt power amplifier. A 70-watt amplifier capacity is the highest that can be obtained within reasonable economy of size and cost.

Also the 12-inch ultra-compact loudspeaker cannot safely handle an amplifier of larger output than 70 watts because of heat dissipation requirements in the voice-coil area.

The Regal system is three times as efficient as most of the ultra-compact loudspeakers. Therefore, it requires one-third the amplifier power to produce the same acoustic output. For the Regal, then, a 20-watt amplifier rating will suffice to produce the same output as lower efficiency systems requiring 50 to 70 watts.

#### Conclusion

An effort has been made in this article to dispel misconceptions regarding ultra-compact speaker systems.

It has been shown that in order to obtain flat bass response the efficiency of ultra-compact speakers *must* be reduced. However, if the efficiency is reduced excessively, impossible power-amplifier requirements or excessively compressed dynamic range results. A happy compromise between loss of efficiency and flat bass response *must* be met.

It has also been shown that conventional 12-inch drivers in 1½ cubic foot cabinets will not produce flat bass response because of an inordinately high primary resonance. It has been proved that removal of the driver suspension stiffness *cannot* sufficiently reduce the primary resonance to effect a flat bass response. Some mass *must* be added to the moving system to produce the desired results. It is this mass addition which reduces the loudspeaker efficiency.

It has been found that the addition of mass to the moving system is not necessarily just a simple matter of utilizing a heavier thicker cone. A precise way of adding mass while maintaining flattest possible frequency response is through the use of a machined metal ring of exact weight located at or very near the voice coil.

Misconceptions concerning the inherent linearity and consequent freedom from distortion of air springs has been dissolved. Attention has been called to any elementary physics text to find formal proof that the adiabatic air compression and rarefaction is *inherently non-linear*. Distortion produced by this non-linearity *must* be greater in ultra-compact sealed cabinets than in larger systems. Too, there is a practical limit of 3/8-in. to the excursion for 12-inch drivers with 1½ cubic foot sealed back cavities no matter how linear the voice-coil structure and mechanical suspension is. Under these conditions a voice-coil overhang in excess of 3/16-in. is a flagrant waste of precious efficiency.

It has been proved that workmanlike cabinet joinery will sufficiently seal the back cavity. Minute air leaks detectable only with a stethoscope *cannot* have an effect on the performance.

High damping factor is an imperative in ultra-compact systems and the insertion of series resistances not only negates the high damping factor of a quality amplifier with resultant degradation of transient response, but wastes precious amplifier power.

The absolute necessity for good high-frequency dispersion, even more important for stereo than monophonic, has been made clear. Diffraction horn units provide optimum distribution of sound into the listening area. These diffraction horns maintain smooth frequency response and the low distortion that can be obtained only with horn-loaded high-frequency drivers.

The ultra-compact system which embodies the features described in this article as well as optimized bass range performance is the Electro-Voice Regal. The photographs are of the Regal III three-way system. A Regal IA two-way system is also manufactured.

In the Regal III both the treble and high-frequency drivers are equipped with Sonophase loading assemblies for flat response, and diffraction horns for dispersion suitable for stereo. Behind the hinged front panel are located the continuously variable level pads which control the balance of these treble and high frequency units. Front location of these controls eliminates the necessity for removal of the cabinet from a bookshelf location for access to the back panel where such controls are often located.

The Regal is fully finished on all four sides to accommodate vertical or horizontal placement. The exterior appearance of this cabinet is designed to blend well with all decors. It is available in beautiful walnut, mahogany or limed oak.

It would appear that the Regals are optimumly designed, with consideration given to *all* factors, to deliver in an ultra-compact enclosure the maximum in frequency response range, efficiency, and freedom from distortion. Those who participated in its development look back with satisfaction on what is called, in laboratory circles, a "happy" design.

### Measuring Equipment

The following laboratory equipment was used for the measurements covered in this article:

Hewlett Packard Distortion Analyzer, Model 330B
100-watt Electro-Voice power amplifier, Model 6006
Electro-Voice Logarithmic Translator, Model 6700
Ballantine a.c. Voltmeter
Electro-Voice Laboratory Standard Microphone and Preamp, Model 6100 (Calibrated 10 cps to 100,000 cps)
General Radio Beat-Frequency Oscillator, Type 1304-B
D'Arsonval-movement rectilinear recorder.

Æ