Revolutionary Loudspeaker and Enclosure

The author describes a fundamentally new loudspeaker system whose 12-inch woofer utilizes an enclosure volume of only 1.7 cubic feet, but whose bass performance is claimed to be superior to that of a true infinite baffle installation.

EDGAR M. VILLCHUR*

THREE OUTSTANDING PROBLEMS that still plague the field of loudspeaker design may be categorized as:

1. How to keep harmonic distortion low in the frequency region below 70 or 80 cps, especially at high power.

2. How to keep frequency response *uniform* and extended at all power levels.

3. How to solve the above two problems without requiring architectural installations, very large cabinets, and difficult final adjustments.

The loudspeaker system here described is the fruit of an investigation that was primarily directed towards solving the first of these problems, that is, towards creating an electro-acoustic transducer that made no compromise with low distortion bass down to 40 cps. The solution to the distortion problem turned out at the same time to be a solution to the problems of uniform bass frequency response and of cabinet size.

The greatest source of distortion in a typical high-quality reproducing system is the loudspeaker. Speaker harmonic distortion in the bass range is tolerated in amounts far greater than would ever be allowed in the amplifier or pickup—values between 5 and 10 per cent below 60 cps and at moderate power are common even in high-quality units. The greatest single source of distortion in the loudspeaker itself is the non-linearity of the voice-coil and rim suspensions which hold the cone and voice-coil to the speaker frame. The elastic stiffness of the suspending members, a property which they must possess in order to perform their functions properly, does not remain constant over the excursive path of the cone; the further the cone moves from its central position the greater is the resisting force constant of the suspensions.

The design of these suspensions and of the speaker's moving system has been refined but not changed radically over the last twenty years or so. The situation is comparable to that of the acoustic phonograph in the nineteen twenties—there *wasn't* much further to go in the direction of improved performance until designers retraced their steps, back to the basic problems associated with converting needle vibrations to sound, and applied a new approach, the electrical one. In the present case, instead of attempting to re-design an already refined mechanical suspension system for a linear force displacement relationship, the elastic stiffness of the mechanical suspension system was substantially eliminated, and a linear, *acoustic* elasticity used instead. Thus, the domination of voice-coil motion by the non-linear elastic mechanical suspensions was also substantially eliminated. The phrase "substantially eliminated" can mean many things; here it is used to denote reduction by a factor between 6 and 10.

Acoustic Elasticity

The acoustic elasticity is provided by the enclosure's sealed-in air, which must be compressed when the cone moves back, and rarefied or stretched when the cone moves forward. In other words the air of the enclosure is used as an elastic cushion, which supplies to the special speaker the restoring force that the moving system is by design deficient in, and that it needs.

Such use of the enclosure's air turns out to have most fortunate consequences, and it is possible to reap large extra dividends over and above the reduction of distortion. The amount of acoustic elastic stiffness available is determined by the cubic volume of the enclosure; the cubic volume which must be provided (not as a minimum but as an optimum value) is of the order of one-fifth the volume of a conventional totally enclosed cabinet for an equivalent speaker mechanism.

The function of an infinite baffle or totally enclosed cabinet is to provide acoustical separation between the waves radiated by the front and back surfaces of the speaker cone, waves which are out-of-phase and would cancel at lower frequencies. One may ask then, why it has not been possible to simply house a speaker in any small enclosed box, or even to close up the back of the speaker frame so that it is airtight, in order to achieve the necessary separation. The answer lies in this same acoustic elasticity referred to, which increases the elastic stiffness of the speaker's moving system and raises its main resonant frequency. The nature of modern loudspeakers is such that below the resonant frequency response falls off rapidly—at the rate of 12 db per octave, in terms of pressure, in an undamped unit.

Suppose, for example, we have a 12-in. loudspeaker mechanism whose main resonance occurs at 50 cps in free air. If we now mount the speaker in a wall the resonant frequency will drop due to the air load mass, perhaps to 45 cps, and if the speaker has been well designed we can expect good response to something below 40 cps, with about 6 db of attenuation at 32 cps.

If we now take this same loudspeaker mechanism and mount it instead in a conventional totally enclosed cabinet (a second choice dictated by the landlord) we will find that the resonant frequency is raised by the additional acoustic stiffness of the enclosed air. Probably the best that we can hope for is to keep the resonant frequency at about 50 cps, an achievement that will certainly require a cabinet volume of over 10 cu. ft. A cabinet of 5 cu. ft. will raise the resonant frequency into the 60 cps region, and the system will suffer a corresponding loss of bass response.

The problem, then, resolves itself into these terms: how provide complete acoustic separation between the front and back of the speaker cone, without raising the resonant frequency above what we want it to be, and without a wall installation or a monster cabinet? The answer dovetails with the solution for suspension distortion referred to previously. We select the values of mass and elasticity for our speaker system as for a conventional speaker, on the basis of the resonant frequency we decide upon. We then design the speaker mechanism with perhaps only 10 per cent of the elastic stiffness that it needs, so that the resonant frequency for the unmounted speaker mechanism is subsonic, of the order of 10 cps. For reasons that will be apparent a little later, this speaker mechanism is useless as a bass speaker in any conventional mounting—which was not designed for 10-cps resonance but for 45-cps resonance.

The final step in the construction of the complete speaker system follows logically. We enclose the back of the

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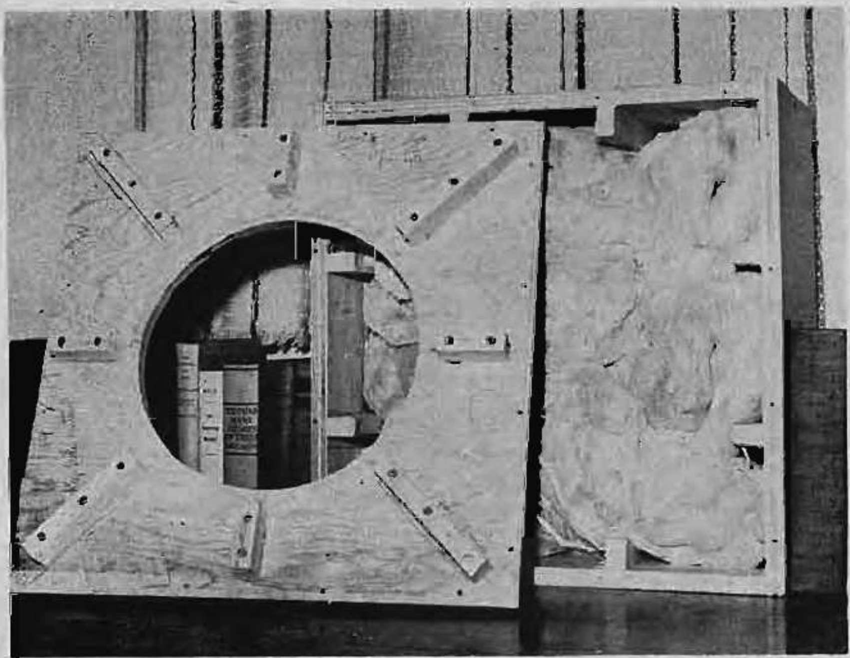

Fig. 2. Experimental enclosure, showing Fiberglas made up in cheesecloth-covered "pillows." The enclosed volume of air, rather than mechanical suspensions, supplies elastic restoring force to the special 12-inch speaker.

speaker with an acoustically sealed volume of air which will supply the remaining 90 per cent of the elastic stiffness to the moving system, and which will raise the resonant frequency to 45 cps. The interior volume of the experimental acoustic suspension speaker, using a 12-inch woofer and designed according to this principle, is 1.7 cubic feet. Increasing the cubic volume will not improve the performance of the system, but will degrade it.

We can now compare the characteristics of the infinite baffle with corresponding characteristics of the acoustic suspension system. This is done in Table I.

Speaker Restoring Force

Speaker suspensions serve two purposes, that of centering the voice coil in the magnetic gap so that it does not rub, and that of providing elastic restoring force to the moving system. The restoring force of a particular speaker cannot be decreased below an optimum value for that speaker. Too low an elastic stiffness will result in increased bass distortion, as the voice-coil will travel out of the path of linear magnetic flux on high-amplitude low-frequency signals, or will actually "bottom" against parts of the magnet structure.

The same principle may be explained in terms of the main resonant frequency of the speaker, which, as we have seen, is determined by the values of elasticity and mass, both mechanical and acoustical, of the suspended system. Other things being equal it is desirable to have speaker resonance as low in frequency as possible, but too low a resonant frequency results in cone excursions too great for the length of the magnetic path provided by the particular speaker. Voice-coil excursion in the bass, for constant radiated power, must be quadrupled for each lower octave, and the attenuation of response below resonance protects the speaker against over-large excursions.

Thus when a speaker is designed with the correct resonant frequency, voice-coil excursion is always kept within the limits of linear flux for all signals, regardless of frequency, up to rated power. Small speakers which can only allow short voice-coil travel relative to their power rating, and which provide relatively poor coupling to the air are properly assigned high resonant frequencies, while speakers which allow greater excursion, or can radiate the same power with less excursion due to some special means for matching them to the air, (such as a horn, for example) can be given lower resonant frequencies.

With the understanding, then, that the non-linearity of the speaker's elastic restoring force cannot be cured by removing or reducing the restoring force itself, the necessity for substituting an acoustic restoring force for the decimated mechanical one becomes apparent. Boyle's law tells us that the restoring force will be symmetrical—that it will be the same coming and going. Acoustic pressure is a function of volume, and it makes no difference that the variations in pressure occur above and below normal atmospheric pressure as a reference level. When the cone moves back the enclosure pressure on the back of the cone is greater than the atmospheric pressure on the front surface; when the cone moves forward the atmospheric pressure on the front of the cone is greater than the pressure of the rarefied enclosed air on the back surface.

For twenty-five years the air in speaker enclosures has been considered an unavoidable evil. It has been unavoidable because of the necessity for providing acoustical separation between the front and back of the cone, and it has been an evil because of the effect of the added acoustic stiffness raising the resonant frequency of the speaker above its optimum point and cutting off bass response. Thus the enclosed air can be rendered innocuous by providing a very large volume whose acoustic stiffness is negligible; this means, ideally, an infinite baffle wall installation or a *very* large, well braced cabinet, both of which are impractical in most homes. Folded horns solve the problem, but again at the expense of large size—a horn that delivers clean, non-boomy bass requires a long flared path and an extremely large mouth diameter. Different methods of "tuning out" the stiffness of the enclosed air have also been used, some using Helmholtz resonance, as illustrated by the various and popular bass-reflex type enclosures, and some by air-column resonance. Critical adjustments are usually required for optimum results.

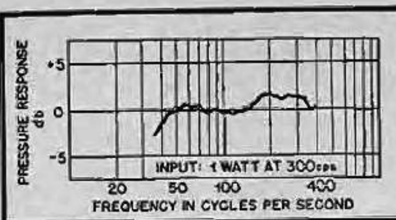

Fig. 1. Typical bass frequency response, on axis, of the acoustic suspension speaker under open field conditions.

The enclosed air in the present system is not a necessary evil but an integral and indispensable part of the loudspeaker, without which the speaker could not operate properly. Since we cannot conquer the acoustic stiffness readily we join it and make it work for us. The enclosure volume is so regulated that in conjunction with the mechanical moving system of the speaker the final resonant frequency is precisely what has been intended—about 45 cps.

When the first experimental model of the acoustical suspension speaker was planned it was reasoned that the bass performance, at worst, would be equal to that of an equivalent conventional speaker in an infinite baffle. It was known that the experimental speaker would provide complete separation between front and back waves, that the cabinet used no acoustical resonators, and contributed no *unwanted* stiffness to the moving system, and that the resistive loading on the back of the cone in an infinite baffle would be more equalled by Fiberglas filling in the experimental cabinet. Accordingly a control twelve-inch speaker, identical except for the suspension system, was mounted in a stairwell.

The difference bewteen the experimental model and the infinite baffle installation, however, was immediately apparent. The experimental unit, because it did not flatten the bass peaks on large cone excursions, had a fuller and cleaner bass, especially in the 40 to 60 cps region. In the beginning it seemed a little unreal to hear the fundamentals of organ pedal notes, which could be felt as well as heard, issuing from this little box.

Later measurements of frequency response and harmonic distortion indicated the reasons for the bass sounding as it did. *Figure* 1 shows the bass frequency response of the experimental model, taken under open field conditions. Bass response uniform within ±1½ db, as indicated in *Fig.* 1, would be ordinary for an amplifier, but is quite unusual for a loudspeaker system. This uniformity of response partly results from the fact that the restoring force is applied smoothly to the whole of the cone surfaces, rather than to the apex and rim of the cone by mechanical suspensions, and partly from the optimum damping of the resonant peak.

The practical result of such uniform response is the absence of boominess. Speech program material, which normally contains no energy below 100 cps, gives no hint of the fact that the woofer reaches down into the low bass. Organ pedal notes, bowed or plucked double basses, etc., are reproduced true in pitch and without ringing.

It must be emphasized that the reproduced response curve is for a complete system rather than for a loudspeaker mechanism alone, mounted as the testing laboratory sees fit. As an illustration of the necessity for care in interpreting response curves for loudspeakers alone, it has been demonstrated that variations in mounting the same speaker in different commercial cabinets can change the effective bass cut-off frequency by an octave, and the amplitude of the bass resonant peak by more than 10 db.

It must also be emphasized that the resonant frequency of 45 cps is for the complete system rather than for an unmounted speaker mechanism, or for a speaker mechanism mounted by the testing laboratory in an infinite baffle. The value of 45 cps was chosen to give full response down to slightly lower than 40 cps; this low-frequency limit was determined to be as low as practically required. Although the above determination was made on the basis of direct experiment with various types of program material, it is supported by authorities in the field, such as Olson.[1]

The harmonic distortion of the experimental model speaker was reduced, from that of the control model in the infinite baffle, by a factor of about three. The harmonic distortion of a later model was measured by an outside testing laboratory and found to reach 1.4 per cent at 46 cps, 10 watts input.[2] It will therefore be seen that this speaker system has not been designed as a compromise "small unit," and it was not intended that a handicap weighting of its performance be allotted to it because of the small size of its enclosure. It is the author's opinion that the bass performance of a speaker with given magnetic and electrical design will be optimum, at the present state of the art, when a moving system with the acoustical suspension is utilized; the small enclosure not only entails no penalties but contributes a tremendous advantage from the point of view of performance quality. It is anticipated that the acoustical suspension principle will become increasingly universal in the industry, and will have general application to speaker systems of all sizes. One obvious application is in electronic organs, where pedal note fundamentals of low frequency can be produced cleanly and at high power from a speaker system installed right in the console.

Damping

The amount of Fiberglas damping material in the enclosure (see *Fig.* 2) is fairly critical. The Fiberglas, in the amount used, completely damps out standing waves at higher frequencies (a task made easier by the small cabinet dimensions, since the standing waves that tend to form are shorter wave lengths, and such sound waves are more easily absorbed) and reduces the Q of the moving system so that the main

(Continued on page 100)

TABLE I

Comparative characteristics of an infinite baffle, a 12-cu. ft. totally enclosed conventional cabinet, and the acoustic suspension system, all using a 12-in. speaker mechanism

	Distortion due to suspensions	Raising of resonant frequency above desired value	Acoustic separation between front and back	Resistive damping on cone	Introduction of acoustic resonances
Infinite baffle and installation	Amount normal to speakers of current design	Very Slight	Complete	Good; air resistance loads both sides of cone	Possible resonance of space into which back of cone faces
12-cu. ft. totally enclosed conventional cabinet	Slightly less than above	Slight	Complete	Fair; air resistance does not load back of cone at bass frequencies	Possible standing waves in cabinet unless properly damped out
Acoustic suspension system	Practically non-existent	None	Complete	Optimum; air resistance loads front of cone and controlled acoustic viscosity applied to back	None

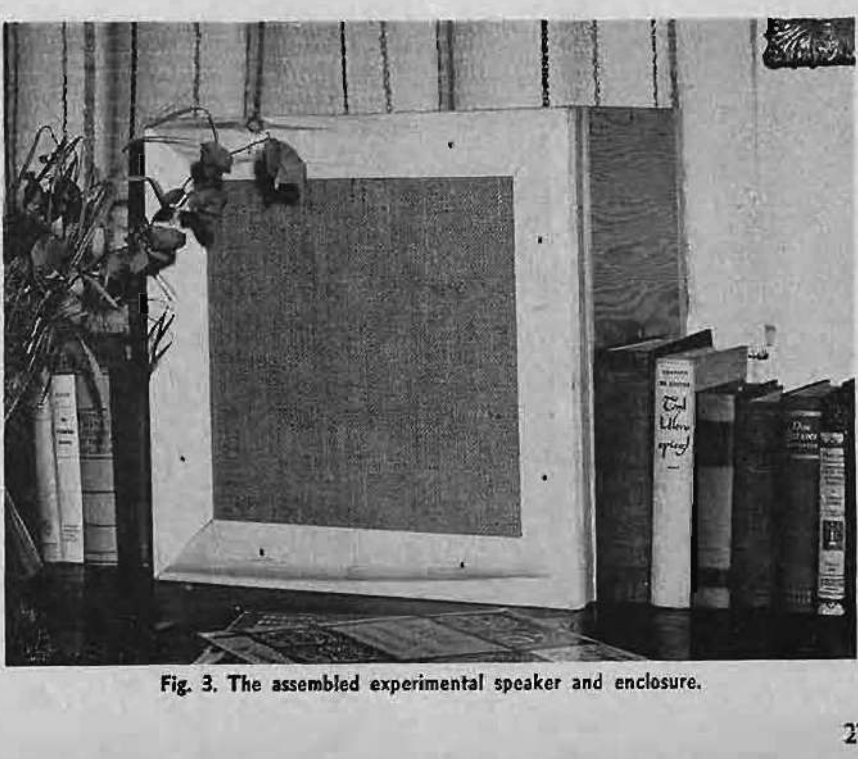

Fig. 3. The assembled experimental speaker and enclosure.

[1] Harry F. Olson, *"Elements of Acoustical Engineering,"* D. Van Nostrand Co., 2nd ed., 1949, p. 477. Dr. Olson lists 40 cps as the low-frequency limit required for the reproduction of orchestral music with perfect fidelity.

[2] These figures were taken with the electrical input as reference level, and are therefore favored by the low efficiency of the system.

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resonant impedance peak is broadened. The Q can be controlled at will; too little Fiberglas yields an output peak in the bass, while too much overdamps the system and attenuates the bass response.

Since the enclosure is actually an integral part of the speaker it cannot be built like conventional cabinets. An acoustical seal must be provided, and the unrelieved pressures that are built up are so great that an unusually large number of ribs and braces are required, together with ¾-inch walls. *Figure* 3 is a photograph of the experimental model of the loudspeaker, its outside dimensions measuring 19 × 19 × 11 in. It uses a twelve-inch woofer and an outside tweeter, the latter not shown. The production model of the acoustical suspension loudspeaker that will be exhibited at The Audio Fair will not be square, for greater convenience in use on shelves or bookcases, and will utilize a twelve-inch woofer with 52-oz. Alnico 5 magnet, plus a high-frequency section mounted within the enclosure. Either the speaker mechanism or the enclosure is useless by itself. The speaker mechanism alone is only half a speaker, and mounting it in any conventional cabinet would be no more feasible than mounting a conventional twelve-inch speaker in the totally enclosed cabinet of approximately 1.7 cubic ft. interior volume.

Figure 4 illustrates by electrical analogy the substitution of acoustical for mechanical stiffness in the moving system. The total compliance, that is,

$$\frac{C_{speaker}\quad C_{air}}{C_{speaker} \times C_{air}},$$

is not changed by the new design from its optimum value in terms of the moving mass and length of linear magnetic path.

An examination of the negative side of the ledger will reveal the fact that the acoustic suspension speaker system is relatively inefficient. The efficiency rating does not, however, fall outside the range of values for conventional direct radiators and a 10-watt amplifier is sufficient to provide ample volume for a room of 3,000 cubic feet. Some of the reasons for the relative inefficiency are:

1. There are *no* acoustical resonators employed in coupling the cone to the air.
2. The moving system has purposely been given a low Q, to damp the bass resonant peak, and has a relatively high mass reactance in the commercial model.
3. The voice-coil gap cannot be made very narrow due to the nature of the centering spider, although the gap width does not fall outside the range of values for conventional units. The acoustical suspension speaker system can, of course, be used as the driver unit for a horn where high efficiency is required.

A patent application for the speaker system described in this article has been filed by the author.

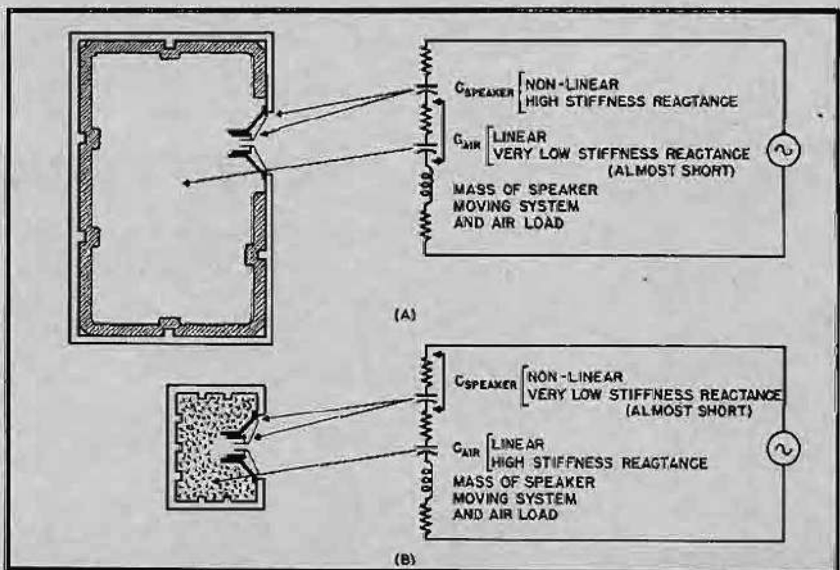

Fig. 4. (a) Electrical-mechanical analogy of the moving system of a conventional speaker in a totally enclosed cabinet. Since C_{air} is made as large as possible, the non-linear $C_{speaker}$ determines the elastic stiffness of the system.

(b) Electrical-mechanical analogy of the moving system of the acoustic suspension speaker. Since $C_{speaker}$ is made very large, the linear C_{air} determines the elastic stiffness of the system. The resultant total elastic stiffness, however, is not changed from its optimum value.

Commercial Acoustic Suspension Speaker

EDGAR M. VILLCHUR*

Performance data on the new loudspeaker system in which cone suspension is a carefully controlled volume of air.

In the October 1954 issue of Audio the writer described an experimental speaker system[1] in which the bulk of elastic restoring force was supplied by the pneumatic spring of the enclosuers' air rather than by the cone's mechanical suspensions. The speaker mechanism itself had a subsonic resonant frequency, but when mounted in its acoustically sealed, Fiberglas-filled enclosure the final resonant frequency of the system was raised to a predetermined value of about 45 cps.

The present article is a report on a commercial unit built to the above design. It is considered that there is general interest in performance measurements of a device constructed on a new principle; in addition quantitative data on loudspeaker performance under carefully defined test conditions is relatively rare, and it is hoped that such a report may stimulate the publication of similar reports on other speakers.

The commercial unit is made as a two-way system, using a 12-inch acoustic suspension woofer in combination with a conventional cone-type high-frequency speaker, and is also made as a woofer system for use with other high-frequency units. The system is illustrated in *Fig. 1*. It was decided to report on the model with the woofer alone for the following reasons: (1) The point of the article is to make a quantitative report on the capabilities of the acoustic suspension system, and (2) test procedures are simplified and made more reliable, thus better subject to accurate duplication by others. Variables such as radiation angle, microphone calibration at higher frequencies, and interference effects between the two speakers which may make microphone positioning critical, are largely eliminated. The performance of the high-frequency section will be only briefly summarized.

The two basic criteria of measurement techniques are validity and reliability. Validity refers to the degree to which the tests measure what they are supposed to measure, and are uninfluenced by other factors. Reliability refers to the accuracy of the measurements: it is an index of the extent to which the measurements can be duplicated at other times and places.

Reliability

A valuable paper on loudspeaker frequency response measurements[2] has pointed out the dangers involved in interpreting speaker frequency response curves when the exact test conditions are not known. Three such curves, made by three different acoustic laboratories using the identical speaker, were compared with each other and shown to differ by as much as 10 db at different portions of the frequency spectrum.

In the present case it was decided that unless test conditions could be established that would make it possible for results to be readily duplicated by anyone with the necessary facilities and skill, the measured data, while it might have a limited usefulness, would be unsuitable for publication. The frequency response and distortion measurements published here have been duplicated without significant variation. It is believed that any tests conducted under the same controlled conditions will achieve results which are within 1 db of the frequency response curve and will add or subtract less than one per cent to the values of the distortion curve.

Conditions Of Test

The enclosure was placed in a hole in the middle of a 2-acre field, its face flush with the surface of the ground (see *Fig. 2*). The speaker was fed from an amplifier with a controllable source impedance, and a microphone was suspended at a distance of 5 feet above the enclosure, on axis with the cone. This means that the speaker was radiating into a controlled 180-deg. solid angle, into essentially free space, driven by an amplifier with a controllable damping factor.

The reliability gained by such test conditions involves a certain sacrifice in validity if what we are measuring is musical fidelity. One does not listen to a loudspeaker when one is suspended from a boom in the middle of a field, and in addition the effect of refraction from the cabinet edges, which would be present under any conditions except a bookshelf installation, is not taken into account.

The alternatives, however, are worse. If we test the speaker in a normally live listening room we will get a different frequency response curve for each microphone position and for each room we use, and interpretation becomes difficult. Room resonances may accentuate or suppress various harmonic distortion products. Similarly, diffraction from the cabinet edges will create interference effects that will change the response curve with even small changes of microphone position. Thus RETMA Standard SE-103, in describing methods of measuring speaker pressure-frequency response, states that if there are no manufacturer specifications on mounting for a direct-radiator speaker:

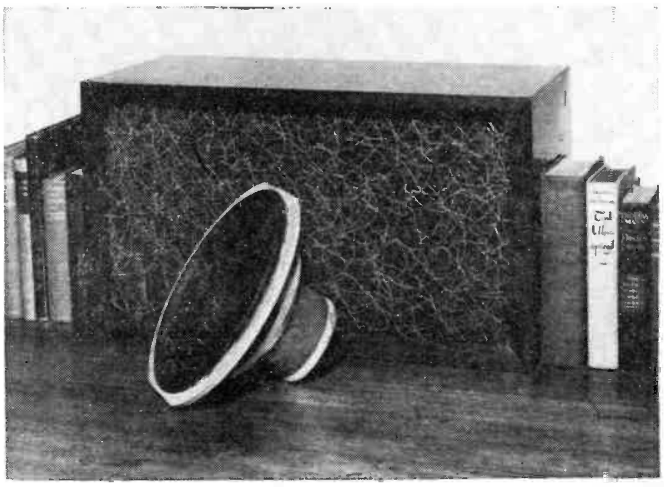

Fig. 1. Acoustic Research AR-1W low-frequency speaker system. The two-way system uses the same cabinet.

[2] Jensen Mfg. Co., "Loud Speaker Frequency-Response Measurements," Technical Monograph No. 1, 1944.

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[1] E. M. Villchur, "Revolutionary loudspeaker and enclosure," Audio, Oct. 1954.

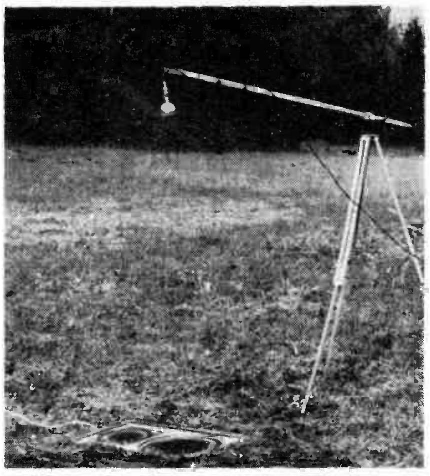

Fig. 2. Field measurement of the AR speaker, using General Radio Sound Level Meter (the microphone is detachable).

". . . it shall be mounted on a rigid, nonabsorbing, effectively infinite baffle. (*This implies a radiation solid angle of 180 deg.*)

"Substantially free-space conditions shall exist in the acoustical environment of measurement.

"The microphone shall be placed on the axis of the speaker as specified by the manufacturer and at a distance at least three times the maximum transverse dimension of the radiating area. (*About 30 inches minimum for a 12-inch speaker.*)

"The values of E_G (*signal voltage*) and R_{SG} (*driving source impedance*) used in the measurement, and the value of R_{SR} (*rated speaker impedance*) shall be specified."

(Italicized portions have been inserted by the writer.)

The conditions of measurement also comply with the recommendations of Standard C16.4-1942, "American Recommended Practice for Loudspeaker Testing," published by the American Standards Association, for both pressure-frequency response and distortion measurements. The ASA specifies 5 feet on-axis for microphone position.

Under the above conditions the measured data was found to be reliable, and results could be repeated at will. The equipment used included the following:

Krohn-Hite audio oscillator, model 430-AB.

Altec 21-BR-150 capacitor microphone, with cathode follower output.

Freed a.c. vacuum-tube voltmeter model 1040.

(Low-frequency pressure response was re-checked in repeat tests using General Radio Sound-Level Meter Type 1551-A.)

Hewlett-Packard Distortion Analyzer 330B.

Dumont Oscilloscope Type 208.

Validity

In a narrow sense the validity of the tests is assured by the use of established and recognized techniques. The readings did measure r.m.s harmonic distortion, pressure-frequency response, and bass transient response.

In a more general sense, however, it is difficult to state that "flat" frequency response under free-space conditions represents musical fidelity under normal listening conditions. We know, for example, that the perception of relative bass content in reproduced program material varies with the volume level of the sound, a phenomenon called the Fletcher-Munson effect. Room acoustics, speaker placement, and other listening conditions can strongly influence the actual perception of sound in the final listening.

Validation must then be made by correlating objective data with subjective judgments, and by deductions that cannot, by their nature, be as rigorously tested as can the data itself. It is universally accepted, however, that the avoidance of dips and peaks in the response curve (ignoring slope) is a good thing. The writer then suggests that, considering the sensitivity of control that can be exerted over electronic as opposed to mechanical devices at the present state of the art, any equalization be assigned to amplifier circuitry.[3]

This leaves us with a response curve for the 180 deg. solid angle which, if flat,

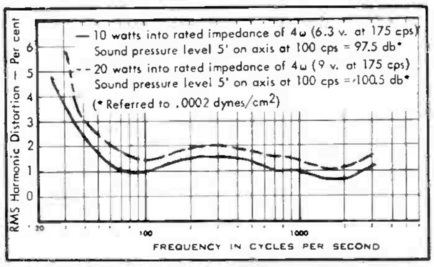

Fig. 3. Harmonic distortion-frequency characteristic of the AR-1W, under conditions noted.

will be transformed into a response that includes boost of the low bass when the speaker is mounted in a corner or at the junction of the floor and the wall. In the case of the unit tested the effective amplifier source impedance can be changed externally by switching from the 8-ohm to the 4-ohm connection, providing flexibility of bass response for different mounting positions. A test run was made with the speaker placed in the corner of a normal listening room, the driving source impedance being low—that is, using an amplifier with a relatively high damping factor. The results of this test are indicative of general performance under such conditions but not as rigorous as the free field tests. Optimum use of the acoustic suspension system, as of any other speaker, is considered to be with an amplifier that has a variable damping factor, which is adjusted to optimum (not maximum) bass response under the conditions of operation. It is agreed among acoustics authorities that there is an optium source impedance from which to drive a given speaker mounted in a given way for the most uniform and extended bass, but this

[3] This applies to the woofer; the problem is complicated in the case of the high-frequency speaker by the differences between on-axis and off-axis response.

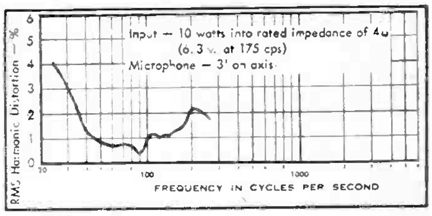

Fig. 4. Harmonic distortion of the AR-1W in the corner of a room.

fact is only recently acquiring general recognition.

General Speaker Characteristics

A brief listing of characteristics of the unit tested appears below:

Model No.	AR-1W
Magnetic circuit	3.3 lbs. Alnico 5 6 lbs. Armco iron
Frame	Cast aluminum
Nominal diameter of speaker	12 inches
Resonant frequency of unmounted speaker	Subsonic
Final resonant frequency of system	43 cps ± 1 cps
Cabinet	Ribbed 3/4 in. stock, dimensions 25 × 11 3/8 × 14 in., acoustically sealed and filled with Fiberglas.

It should be noted that the main factors which determine the final resonant frequency are the mass of the moving system, the cubic volume of the cabinet, and the amount of Fiberglas filling, all of which are readily subject to accurate control. The elastic stiffness of the suspensions, a factor which is not so easily controlled with accuracy, contributes only about 10 per cent of the total elastic stiffness, and the resonant frequencies of different production units can therefore be kept within small tolerances.

Harmonic Distortion

Distortion data is listed before frequency response data in order to validate the former. We are not interested in the total amount of sound put out by the speaker when stimulated at particular frequencies, including spurious harmonics, noise, etc., but in the output of reasonably undistorted sound at different input frequencies. A frequency response rating that extends down to 32 cps has little significance if the harmonic distortion at this frequency is 40 per

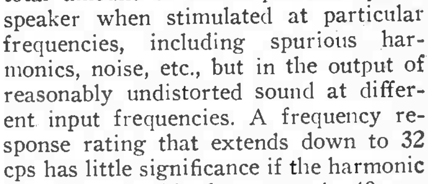

Fig. 5. Oscillograms of acoustic output of AR-1W (conditions listed in Fig. 4) at 10 watts to rated impedance, at 32, 60, and 100 cps, respectively, from left to right. The extreme uniformity of output is accidental, as can be seen from the frequency-response graph of Fig. 7.

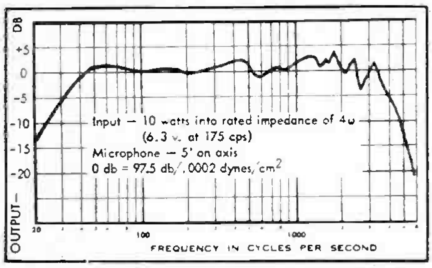

Fig. 6. Frequency response of the AR-1W under conditions noted.

cent at moderate power, other than to indicate that the designer has attempted to extend the bass response further than he should have for that particular speaker. Attenuated output is by far preferable to distorted output. Similarly, a low distortion reading at 32 cps has little positive significance if speaker response is down 20 db at that point. The distortion graph of *Fig.* 3 should be read in conjunction with the frequency-response graph of *Fig.* 6, so that the distortion curve refers to the reproduced frequency range, and the frequency response curve refers to the range of low-distortion reproduction. The necessity for validation of the frequency-response curve by distortion data is clearly indicated in ASA Standard C16.4-1942, which in listing essential practices for plotting pressure-frequency response characteristics states:

"Unless otherwise stated, the values of pressure plotted shall be those corresponding to the *fundamental* frequencies." (Italics supplied.)

The graphs of *Fig.* 3 plot the r.m.s distortion-frequency characteristic of the AR-1W at 10 and 20 watts input to the rated impedance (4 ohms). Amplifier gain was adjusted to 6.3 and 9 volts output across the speaker at 175 cps, and left that way for the run. The radiation angle was 180 deg. *Figure* 4 plots the distortion-frequency characteristic of the AR-1W in the corner of a normally live living room, and *Fig.* 5 shows oscillograms of the acoustic output of the speaker at 32, 60, and 100 cps, with 10 watts into the rated impedance under the latter conditions.

A word of caution about the interpretation of the distortion figures must be inserted at this point. The reference power levels are electrical, and the inefficiency of the speaker (referring to the electrical power required for a given acoustical power) therefore favors the readings. The sound level is also quoted, but will be meaningless to many readers. The only correct way to compare the data with corresponding data from another speaker would be at the same sound pressure level at a given frequency, not at the same electrical power input. The range of difference in commercial speaker efficiencies is probably at least 25 to 1.

The efficiency of the AR-1W, on the other hand, is close in value to that of several other commercial units for which the writer has a high regard. It should also be noted that the efficiency of the AR-1W remains essentially constant down to very low frequencies (see *Fig.* 5), and the absolute efficiency in the 30–60-cps octave may be greater than that of another speaker with a much higher over-all efficiency rating. For example, if pressure response at 40 cps is down 9 db (which still allows for an excellent low-frequency reproducer) the efficiency at that frequency is reduced by a factor of 8, and it will require 8 times the amplifier power to create the same sound pressure level as at the reference level. The AR-1W used as an organ pedal tone generator could not be considered an inefficient speaker.

When the 8-ohm connection is used for 180 deg. radiation conditions (mid-wall shelf mounting) the efficiency is halved. The conditions of the 8-ohm connection can be achieved without such loss of speaker efficiency by using an amplifier with a damping factor of 1.

Frequency Response

The graph of *Fig.* 6 plots the frequency response of the AR-1W under

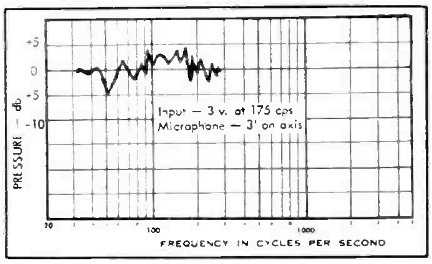

Fig. 7. Frequency response of low range of AR-1W in the corner of a normally live room.

the open-field conditions described above. The ratio of horizontal-to-vertical scale follows RETMA Standard SE-103, which states:

"The length of a 10 to 1 frequency interval shall be the length of 30 db on the ordinate scale."

The graph of *Fig.* 7 plots the frequency response of the speaker in the corner of a normally live living room, fed by an amplifier with a damping factor of 4.

Transient Response

Good transient response is associated with a uniform steady-state frequency-response curve. The transient response of the AR-1W can be predicted from the frequency-pressure curve of *Fig.* 6.

The bass transient response was also checked visually with square waves and an oscilloscope. *Figure* 8 illustrates the response of various speakers to a square wave of subsonic fundamental frequency. In (A) the cone of an ideal speaker moves forward to the top of the square wave and remains completely motionless over the horizontal portion, while air pressure at the microphone decays smoothly. There is no hangover whatsoever. A poorly damped system is represented by the acoustic output in (B), which exhibits definite ringing after the initial stimulus. (C) is the acoustic output of the AR-1W speaker system as recorded with microphone and oscilloscope, showing slight overshoot.

Efficiency

Efficiency has no direct relation to quality, but it does have an indirect one in that the power demanded from the amplifier by an inefficient speaker may exceed the amplifier rating. If the available voltage driving the amplifier is great enough the amplifier may then overload and distort. It is also true that in A-B tests the louder system tends to sound better automatically, and an efficient speaker has the edge in audio salesrooms if the electrical levels are not adjusted for equal volume from each speaker.

Interpreted in general terms the sound pressure levels indicated in *Fig.* 6 mean that a good 10-watt amplifier is adequate for the AR-1W or AR-1 speaker for moderate listening levels in typical living rooms. For larger rooms and for those who like very high levels of reproduced sound, at least 30 clean watts are required. The RETMA efficiency rating of the AR-1W at 100 cps is 21.5 db.

In constructing a figure of merit for a loudspeaker system designed for home reproduction the question would arise as to what place, if any, efficiency would receive. If manufacturers were canvassed as to the significant factors in such a

(Continued on page 33)

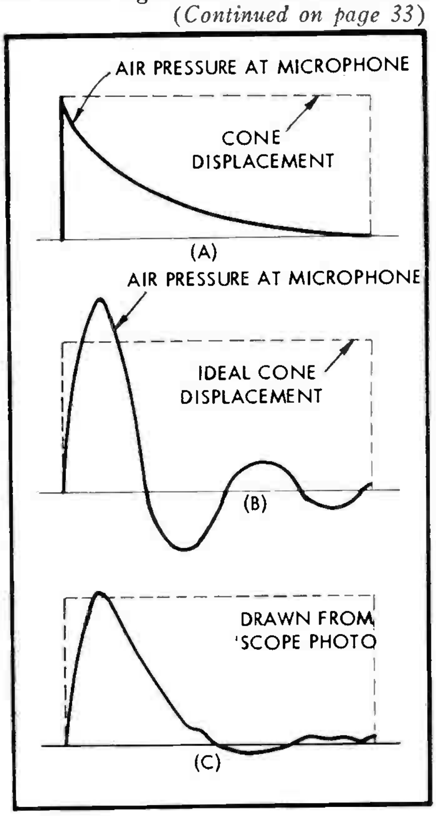

Fig. 8. (A) Acoustic output of a speaker with unlimited and perfectly uniform frequency response, when stimulated by first half-cycle of square wave. (B) Acoustic output of poorly damped speaker system, same stimulus. (C) Oscilloscope photograph of acoustic output of AR-1W to first half-cycle of a square wave of 8 cps fundamental frequency. The rounding of the initial impulse is due to the lack of woofer high-frequency response.

Ravel: Bolero, La Valse, Valses Nobles et Sentimentales, Alborada del Gracioso, Pavane pour une Infante Defunte. Orch. du Th. Champs-Elysées, P. de Freitas Branco.
Westminster WL 5297.

Ravel: Bolero, La Valse.
Honegger: Pacific 231.
Dukas: Sorcerer's Apprentice. The Paris Conservatory Orch., Ansermet.
London LL 1156.

These two should have plenty of hi-fi appeal, for one thing. Our scout finds both brilliantly recorded, the Westminster "spectacularly so" (no doubt with Westminster's ultra-brilliant close-up technique of microphoning), the London "with better balance and fuller body." Both discs are well chosen collections of related hi-fi-style material, and those who know the respective Westminster and London styles of recording will hardly be dissappointed in either case.

The two feature pieces on both, Bolero and La Valse, Scout #1 finds musically not too satisfactory: both feature slow tempi that, he feels, detract from the vitality and drama of these two *tours de force* of dramatic climax.

Debussy: Nocturnes for Orchestra (Nuages, Fêtes, Sirènes).
Ravel: Rapsodie Espagnol, Menuet Antique. Paris Conservatory Orch., Fournet.
Epic LC 3048.

Scout #1 is full of enthusiasm for this disc and feels that these are "definitive" performances. He particularly likes the Ravel Rapsodie which he says has no equal on LP by a wide margin and makes a remarkable listening experience.

I might add that Fournet's work in other records I've heard recently would confirm this judgment; he is evolving into one of the best conductors of French music we have on records. The Scout speaks of the "distinctively Gallic sound of the horns" in the third movement of the Rapsodie—this, of course, would be due to the peculiar French-type instruments, quite unlike the ordinary "French" horns of other nations and played with a tell-tale vibrato, like a saxophone, where all other horns play without vibrato. The French-style horn playing is absolutely dreadful in Wagner or Brahms, but in French-composed music it is, needless to say, entirely natural and proper.

An adequate recording, not as fancy in sound as some competitive versions (Mercury, Angel)—this release dates from awhile back (last summer) before Epic had hit the hi-fi stride it now maintains. Get this one for the music, not the fi.

Franck: Symphonic Variations.
Fauré: Ballade.
Saint-Saëns: Piano Concerto #5. Jean Doyen; Magda Tagliaferro (Saint-Saëns); Lamoureux Orch., Fournet. **Epic LC-3057.**

Another Fournet-conducted disc and our scout again approves heartily. The Saint-Saëns, with pianist Tagliaferro, is well ahead of competition on M-G-M and Vox, he feels; the Franck and Fauré with Doyen at the piano benefit from the "fine idiomatic support" of the orchestra and Fournet. A first class musical offering in all respects, it would seem. (Indeed, I trust this account so well that I'm going to have to put the disc aside and play it anyhow—ghost or no ghost. I expect it to be a pleasure. There's nothing like French music well played.)

Recording, Scout says, is "good middle-period Epic." Readers will remember that this department called attention last year to Epic's initial difficulties with the new Phillips imported recordings and prophesied that things would soon improve. They have and Epic is out in front soundwise this year. The present disc dates from the end of last year when things were beginning to go well technically—hence, "good middle-period."

Offenbachiana (adapted and orchestrated Rosenthal). RIAS Symphony, Rosenthal.
Remington R-199-183.

Offenbach-Rosenthal: Gaité Parisienne.
Chopin: Les Sylphides. Philadelphia Orch., Ormandy. **Columbia ML 4895.**

Three ballet scores adapted from 19th century works not originally ballet-intended, but two of them, at least, are so well known now as to be more famous in the ballet form than in the original. Ballet suites of this sort have had wide popular success as musical numbers (minus the ballets) and so it's not surprising that Manual Rosenthal has tried to follow up the phenomenal success of his "Gaité Parisienne" with another from the same composer.

"Lightning seldom strikes twice" is Scout #1's succinct comment on the Remington "Offenbachiana," above, that is the result. He thinks it of little interest, though the playing is "live and bouncy" with nice orchestral coloring. In all truth, Offenbach wrote some pretty thin stuff to pad out the relatively few bits of really fine music in his enormous output of opera. It takes a very careful search to come up with music as consistently good as "Gaité Parisienne."

As to "Gaité" itself, on Columbia, Scout reports it is a fast, virtuoso concert performance, over-brilliant. This, I'd say, is what usually happens when a top virtuoso orchestra takes up a popular light item and makes a big thing of it.

"Sylphides," he says, has a fine string sound (it's predominantly a string piece, arranged from the piano originals) not too heavy and beautifully recorded.

SUSPENSION SPEAKER

(from page 20)

figure of merit a strange correlation could undoubtedly be made between the features of a particular manufacturer's speaker and the qualities emphasized as most important to the figure of merit. Manufacturers of low-efficiency speakers would tend to deny the relevancy of efficiency, while manufacturers of high-efficiency speakers would probably take an opposite stand.

At the risk of the writer's seeming to have an ax behind his back in need of grinding, it is submitted that efficiency should not appear in the main term of a figure of merit for loudspeakers, but in a second term connected by a plus sign. The second term would include other factors such as price and size. Low efficiency simply means that for given performance results more money, weight, and space must be invested in amplifying equipment. The relative cost, in terms of these three factors, of the added electronic capacity can be calculated, but should not be reflected in the index of quality.

In the case of the Acoustic Research woofer, efficiency has been deliberately traded for extended and uniform bass response and low distortion. It is obvious that the magnetic circuit used in the AR woofer is sufficient for a motor of very high efficiency. The sacrifice of efficiency is justified, in the mind of the writer, by the performance data reported in this article.

High-Frequency Speaker

The high-frequency speaker used in the model AR-1 system is an 8-inch direct radiator. Its performance characteristics, as used in the system with a 12 db/octave bass-droop network and pad, and as measured by Acoustic Research, are: frequency response 800—13,000 cps ± 5 db, and distortion over above range with 10 watts input, 1 per cent maximum.

Loudspeaker Damping

The author presents an interesting and thorough discussion of the effects of mechanical, electrical, and acoustic damping on frequency response, resonances, and transient performance.

EDGAR M. VILLCHUR*

One of the less widely understood subjects in audio, it would appear, is damping and transient response, particularly in relation to loudspeaker systems. The following is a general outline of the problem, and an attempt to clear up some of the more prevalent misconceptions.

Some of the material here presented, by virtue of the fact that it is contrary to many popularly accepted (and even published) ideas, may appear to be radical in approach. It is, however, entirely conservative. The subject has been well investigated in the literature; the main concepts in this article, for example, appear in a much more complete and mathematically rigorous form in Beranek's *Acoustics*,[1] and a motor engineer should easily recognize the lack of novelty of the basic ideas relative to electro-magnetic damping.

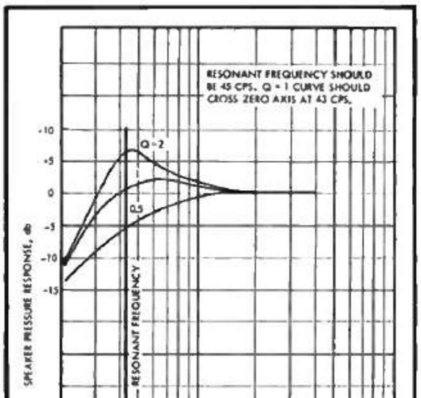

Fig. 1. Response of a direct-radiator speaker system, in the region of resonance, at different values of Q of its mechanical system, (After Beranek)

Terminology

Before proceeding further we must be clear about the meaning of our terms. Damping refers exclusively to the introduction of a *resistive* element into a vibratory oscillatory system. This resistive element may be electrical, mechanical, or acoustical.

If we introduce alternating energy into an electrical or mechanical system—we could apply a.c. to an electrical circuit, or vibratory force to a mechanical device—the system will respond, oscillating in the grip of the applied stimulus. The extent to which the system will oscillate, fondly referred to as its "response" by audiofans, depends on its impedance. Impedance may be thought of as mechanical, acoustical, or electrical intransigence—the unwillingness to be moved or to pass current under the particular conditions involved.

The *reactive* part of the impedance, associated with such characteristics as mass, elasticity, inductance, etc., allows the load to accept energy for storage only, not for absorption. A frictionless system of a weight on a spring would go on bobbing forever once it was started. The *real* or resistive part of the impedance, associated with electrical resistance, friction, and viscosity, permits the load to accept energy permanently, i.e., to absorb it (or, as in the case of radiation resistance, to accept energy for one-way transmission).

When we concern ourselves with how the system acts, not during the time when it is working steadily, but at the very start, on the "attack," and also at the end, after the stimulating and presumably controlling force has been removed, (the "decay"), we are dealing with *transient* rather than *steady-state* response.

Attack and Decay

It would be useful to consider concrete examples of the transient response of mechanical systems. Let us consider two such examples: the response of a kettle drum to the impact of the drumstick, and the response of a loudspeaker to a signal representing the drum's recorded sound.

When the drumstick falls it produces a deformation of the stretched skin. The velocity of the initial, complex movement of the membrane over the distance travelled will not be in step with the natural frequency of the drum's mechanical-acoustical system. The strike sound, instead of having the same pitch as that to which the drum is tuned, will exhibit fundamental components of much higher frequency.

The amplitude of the steady-state sound that will ultimately appear due to the blow will depend on the impedance of the drum's primary moving system, relative to the applied force. The amplitude and duration of the initial, higher-pitched attack sound will depend on the impedance of the drum to higher frequency stimuli, and the Q at these higher frequencies. The more amenable the drumhead is to moving at velocities and amplitudes corresponding to higher frequencies than its fundamental, the crisper will be the attack sound. The nature of the transient acoustical attack is therefore a function of the frequency response of the drum—the relative amount of sound it puts out when stimulated at different frequencies.

Once the drumstick bounces off, the drum is on its own. It can operate only on the energy that was supplied in the single stroke, as it will receive no more energy until it is hit again. We know, of course that the sound continues. If the drum were totally undamped it would continue to vibrate forever, but mechanical and acoustical resistance provide light damping to absorb the vibratory energy gradually, and the sound takes a relatively long time to die away unless it is checked by the player. This is the decay; the length of decay time depends on the amount of damping.

In the case of a loudspeaker reproducing the kettle drum's sound a similar

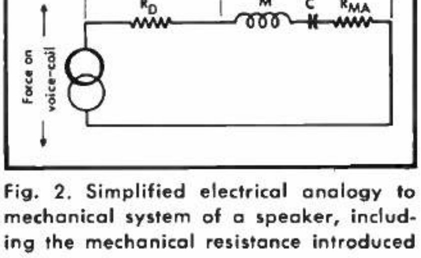

Fig. 2. Simplified electrical analogy to mechanical system of a speaker, including the mechanical resistance introduced by magnetic damping

* *Acoustic Research, Inc., 24 Thorndike St., Cambridge 41, Mass.*

[1] Leo L. Beranek, *Acoustics*, McGraw-Hill Book Co., 1954.

analysis applies. The initial stimulus is provided, not by an external blow, but by a surge of signal current from the amplifier, and the attendant magnetic field built up around the voice coil. And since the speaker should vibrate only when the signal so dictates, it must have a highly damped mechanical system. If the speaker cone, like the drumhead itself, continued to vibrate after the controlling stimulus had stopped there would be a hopeless confusion of sound.

The quality of the reproduced attack sound, as in the case of the drum, is a function of the speaker's response to higher-frequency sound components. Thus the attack sounds reproduced by multispeaker systems are controlled, not by the low-frequency performance of the woofer, but by the performance of whatever unit is assigned to reproducing the mid and higher frequencies, and may involve the woofer itself little or not at all, depending on how low the crossover frequency is. By definition, a woofer which covers only the low-frequency range cannot and is not intended to respond to most transient attack sounds. Its contribution to a crisp drum beat is to move, however lumberingly, in accurate reproduction of the fundamental and lower harmonic frequencies only; the sharper attack components are reproduced and contributed by other speakers.

So much for the general background of the problem. We may now turn our attention to the more specific question of loudspeaker damping.

Magnetic Damping in Speakers

Speakers are damped, in their main resonance region, in three ways: mechanically, through friction in the suspensions, acoustically, through various methods of applying acoustic resistance and through the air load resistance, and magnetically. In bass-reflex and horn systems acoustic damping normally predominates; in direct-radiator systems most of the burden of damping falls on the electro-magnetic system. Damping of cone break-up modes of vibration, at higher frequencies, also takes place in the cone material and in its edge termination, but this is not the subject of the present article.

Magnetic damping results in an additional mechanical resistance being applied to the moving system. This mechanical resistance can be investigated directly in a very simple manner—if one shorts out the terminals of a loudspeaker containing a fairly heavy magnet, and then tries to work the cone back and forth manually, it will feel as though the voice-coil has been immersed in a viscous fluid. The apparent viscosity disappears as soon as the terminal short is removed. When the speaker is connected to an amplifier with a low source resistance the amplifier source resistance replaces our experimental short. If the source resistance is raised in value (lower damping factor) the mechanical damping resistance is correspondingly decreased.

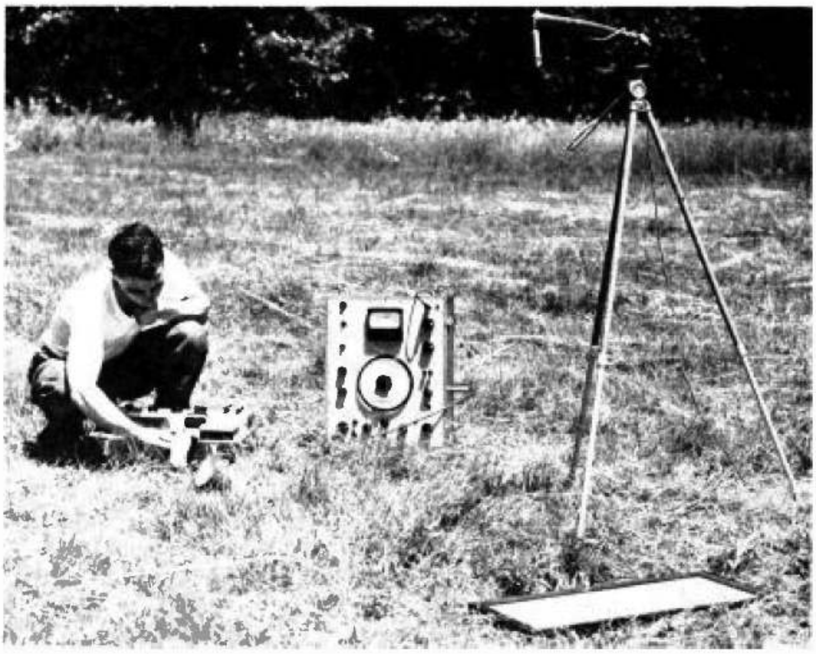

Fig. 3. Test set-up for measuring speaker performance. The speaker sees a controlled solid angle of 180 deg.

The effects of speaker magnetic damping are twofold:

1. It prevents cone vibration from continuing after the signal has stopped (hangover).

2. It controls bass response in the frequency region of resonance, perhaps an octave on each side.

The first of these effects is generally known and widely commented upon, while the second is not so well known.

The mechanical resistance introduced by magnetic damping may become the major element in the speaker's mechanical impedance in the region of resonance, where mass and compliance reactances cancel each other out. Actually, the influence begins at some frequency above resonance, when the mass reactance becomes equal to the damping mechanical resistance.

The extent of the influence of damping is a function of the value of moving mass in relation to the damping resistance, more precisely, on the mechanical Q of the system. Where the resistance is small relative to the mass reactance at resonance (a high-Q system) the effect of damping on bass response is small; where the value of resistance is large in relation to this mass reactance the effect on bass response is great. This is a simplified way of saying that which is described exactly by the well-known family of curves representing the frequency response of resonant systems for different values of Q. *Figure* 1 reproduces a set of such curves, specifically applied to the acoustic output of speakers.[2] All dynamic loudspeakers, of course, are mass-elasticity resonant systems.

The crux of the matter is that for that value of Q which will bring the resonant peak down to a flat curve, the damping will also be such as to prevent any hangover. For lower values of Q the hang-

[2] Ibid., p. 226. Also see D. E. L. Shorter, "Loudspeaker cabinet design," p. 382, *Wireless World*, Vol. 56, No. 11, Nov., 1950.

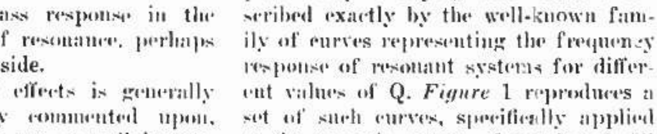

Fig. 4. Recorded speaker response curves for different values of amplifier damping factor, open field conditions. The calibration curve for the recording equipment, not including microphone, appears at the top. (See Fig. 5 for corrections).

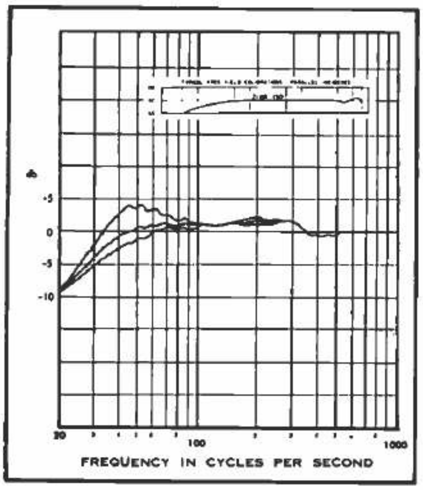

Fig. 5. Response curves of Fig 4, corrected for the calibrated errors of recorder and microphone (calibration curve for the latter appears in inset).

over will continue to be damped out in neither better nor worse fashion, but other things being equal, there will be an attenuation of bass response as indicated in *Fig.* 1.

The mechanical Q of a system is controlled by the relative values of mass and resistive elements at the resonant frequency, according to the relationship:

$$Q = \frac{\omega_R M}{R}$$

where M = mass

R = mechanical resistance

ω_R = 2π times the resonant frequency of the system. (Note that $\omega_R M$ is the mass reactance at resonance)

In the case of a loudspeaker this expression may be elaborated into:

$$Q = \frac{\omega_R M}{R_A + R_M + R_D}$$

where

M = mass of voice-coil and cone, plus acoustical mass reflected into the system

R_A = acoustical resistance

R_M = mechanical resistance of suspensions

R_D = equivalent mechanical resistance associated with magnetic damping. This is equal to $\frac{B^2l^2}{R_{vc} + R_{int} + R_x}$, where B is the air gap flux density, l is the length of wire in the gap, R_{vc} is the voice coil resistance, R_{int} is the amplifier source resistance, and R_x is any other d.c. resistance in the line.

The dynamical analogy to the mechanical system of a loudspeaker, including the mechanical resistance due to magnetic damping, is shown in *Fig.* 2. Since the amplifier source resistance determines the value of this magnetic damping mechanical resistance, a variable damping factor control can be used, particularly with a direct-radiator speaker, to control the Q over a fairly large range of values.

It can be seen in *Fig.* 2 that at frequencies well above resonance the equivalent electrical circuit is inductance controlled, that is, the net reactance is inductive, representing mass control in the speaker's mechanical system. As the frequency is lowered in the direction of resonance the net inductive reactance decreases, and current flow (velocity in the mechanical system) correspondingly increases. This is as it should be; the cone velocity of a direct-radiator speaker, for constant acoustical power, must double with each lower bass octave in order to offset the progressive decrease in air-load resistance.

At some frequency, depending on the speaker used—perhaps an octave above resonance—the net inductive reactance will become equal to the total resistance. R will thenceforth, as the frequency is lowered, act to reduce current progressively, compared to the rising value that would exist in a pure LC circuit. That value of R which produces a Q of about 1 gives an approximately flat curve, with neither resonant peak nor bass attenuation.

If R_D is swamped by large values of other resistive elements due to the nature of the speaker system, its effect will obviously be minor.

Below resonance the net capacitive reactance of the circuit begins to mount, until it is greater in value than the total R. An octave or so below resonance, then, R again loses its influence.

It should be clear at this point that the absolute value of the mass of the speaker's moving system has no relation whatever to damping or hangover. It is the mass-resistance *ratio* that influences the Q. The only exception to the former statement is provided by the new electrostatic units, where the mass of the very light diaphragm may be kept so low that the controlling resistive element is the actual air load resistance.[3] In such a case all system constants become tied to a fixed reference of resistance.

Nor does the absolute value of the mass influence attack performance. What is needed for the proper reproduction of attack sounds is: (a) the same level of system response at the attack frequencies as at the fundamental, however this is achieved, and (b) uniform response in the region of attack frequencies (corresponding to proper damping in this range), so that the attack frequencies themselves don't ring.

So much has been said and written contrary to some of the above conclusions that it was felt that a set of actual field measurements, illustrating the main points of discussion, would prove both interesting and informative. Accordingly a direct-radiator speaker system of known characteristics was fed by an amplifier with controllable damping factor, and facilities for measuring the speaker frequency response and decay characteristics were provided, as illustrated in *Fig.* 3. The test set-up in which the speaker is sunk into the ground in the middle of an open field, its face flush with the surface, have been described by the writer.[4] The speaker sees a controlled solid angle of 180 deg., and test conditions conform to ASA and RETMA specifications. Validation of the frequency-response curves of the speaker used as representing essentially fundamental output was also described in the article referred to.

The equipment used included the following:

- AR-1W Acoustic Research speaker system (woofer only)
- Fairchild 275 power amplifier, with variable damping factor
- Bruel and Kjaer beat frequency oscillator BL-1014, mechanically coupled to:
- Bruel and Kjaer level recorder (automatic) BL-2304
- Electro-Pulse pulse generator 1310A
- Bruel and Kjaer microphone amplifier BL-2601
- Altec 21-BR-150 capacitor microphone

The acoustic output of the speaker over its frequency range was measured, using the automatic frequency-level recorder, at an input power of 20 watts. These curves were re-run on the same graph paper[5] with all conditions the same, except for a change of setting of the damping factor control (thereby changing the damping resistance and the speaker's mechanical Q). The results are reproduced in *Fig.* 4. It will be seen that the curves conform closely to the

[3] Arthur A. Janszen, "An electrostatic loudspeaker development," p. 89, *JAES*, Vol. 3, No. 2, April, 1955.

[4] Edgar M. Villchur, "Commercial acoustic suspension speaker," p. 18, AUDIO, July, 1955.

[5] Unfortunately, 30 db per decade (American standard) graph paper was not available, and 20 db per decade paper had to be used.

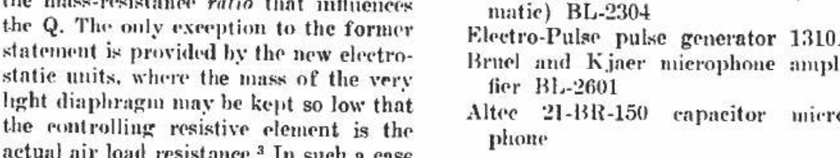

Fig. 6. Acoustic output of speaker, as monitored by microphone and oscilloscope, in response to step-front of low-frequency square wave: A (left), With amplifier damping factor of 6; B (center), with damping factor of 1; C (right), With damping factor of 0.1.

theoretically plotted curves of *Fig.* 1, especially when they have been corrected for the calibrated errors in recorder and microphone (*Fig.* 5).

The effect of the increased amplifier source resistance on the upper frequency range can also be seen: this is due to the rising *electrical* inductive reactance of the voice-coil.

Figure 6 is a series of three oscilloscope photos of the wave forms of the acoustical output of the speaker, in response to the step front of a low-frequency square wave. They represent magnetic damping conditions associated with each of the three curves of *Fig.* 5. Note that there is no significant difference between the recorded hangover associated with a properly damped and an over-damped system. Ringing at the speaker's resonant frequency is clearly seen, however, in the under-damped condition, as is a large increase in the slight initial secondary ringing, at a higher frequency, which shows up as a disturbance halfway down the first decay slope.

It should be possible at this point to see the error in that misconception about damping which denies ability to the amplifier source resistance to damp the speaker mechanically because of the latter's low conversion efficiency. Magnetic damping is, of course, powerless to control effects which take place in the course of energy transfer between the mechanical system of the speaker and the surrounding air, but it is of paramount importance in controlling the mechanical system itself. It should also be noted that the magnetic damping of the speaker is a function of the magnetic field strength and of the amount of copper in the gap. Since these two factors do not uniquely determine electro-acoustic efficiency (the mass of the moving system, and the method of coupling the diaphragm to the air are at least as important), there is no direct relationship between electro-acoustic efficiency and damping. The AR-1W used in these tests, for example, a speaker with very low over-all efficiency, has unusually high magnetic damping due to its heavy magnet and to the large amount of copper in the gap. It is, as a matter of fact, in danger of being over-damped when improperly used, as under conditions similar to those of the lower curve of *Fig.* 3 (180 deg. solid angle of radiation, high damping factor), in which bass attenuation can be seen.

Figure 7a is a recorded graph of the frequency response of the speaker in a living room, with the two extremes of damping factor used in *Fig.* 4. Note that the over-all shape of the curve is affected by changing the damping factor, in the same way as it was in *Fig.* 4, but that the irregularities due to the acoustical environment of the room are completely uninfluenced. This illustrates the independence of room ringing—associated with peaks and dips in the steady-state frequency-response curve—from damping in the speaker system itself. The only damping that can have any effect here is that connected with the room surfaces; neither magnetic, mechanical, nor acoustical damping of the speaker's moving system can affect a cure. The latter point is further illustrated in *Fig.* 7b, a frequency response record of the same speaker in a different part of the same room.

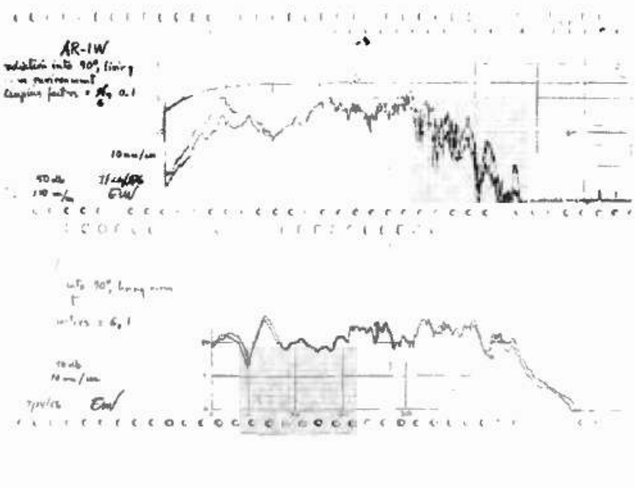

Fig. 7. A (top), Response curve of speaker mounted in two-sided corner of room. Lower curve is for damping factor of 6, upper curve is for damping factor of 0.1. B (bottom), Response curve of the same speaker in a different position in the same room at damping factors of 6 and 1.

Effect of Solid Angle Seen by Speaker

It may have been noted that the condition of high damping factor produced bass attenuation when the speaker radiated into 180 deg. in the open field, but that the same high damping factor is associated with essentially uniform response down to 30 cps in the indoor measurement (ignoring room-derived irregularities, and correcting for microphone and recorder). In the room the lower damping factor produces a somewhat exaggerated bass. The primary reason for this lies in the fact that the speaker in the room was mounted so that it faced into a reduced solid angle (90 deg.)—in a corner, off the floor.

Figure 8, also borrowed from Beranek's *Acoustics*, shows the change in bass response produced by restricting the solid angle seen by a speaker. Higher-frequency components are concentrated in the area ahead of the cone, and if the environmental solid angle seen by the speaker does not similarly restrict the non-directional bass, it will be thinned out relative to the treble. As might be expected, below the frequency at which the speaker's signal becomes essentially non-directive each successive halving of the solid angle doubles the bass power, or raises the response curve by 3 db.

It would seem to be a good idea for someone to design an equalizer network to produce variable bass boost to compensate for this effect on performance due to change in solid angle. In the meantime the closest approximation to such a circuit is a variable damping control, which gives the user additional flexibility in tailoring the low bass response of his system to the conditions of speaker mounting. Lowering the damping factor may also affect the mid and high frequencies, and a circuit which only varied the damping factor (from a high value down to one-half or so) over the bass frequency range would be useful.[6]

Other Misconceptions

I would like to add some further comments to this article in an attempt to lay to rest some of the old wives' tales about speaker damping. The insertion of a few numbers into the general rela-

(*Continued on page 84*)

[6] The writer has, since completing the draft of this article, learned of such an amplifier design available commercially—in the McIntosh MC-30*A* and MC-60*A*. Tests on a sample MC-60*A* showed it to perform precisely according to expectation.

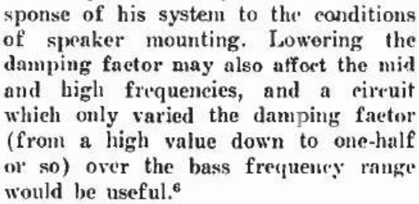

Fig. 8. Effect of restricting the solid angle seen by a loudspeaker. The top curve (A) is for a solid angle of 45 deg.; each succeeding lower curve represents an increase of the solid angle by a factor of 2. (After Beranek)

DAMPING

(from page 27)

tionships that we have discussed should help in this connection.

1) Let us study the case of a speaker whose nominal impedance is 8 ohms. The d.c. resistance of its voice-coil will be of the order of 6 ohms. Thus the total d.c. resistance that the speaker sees, looking back at the amplifier, is equal to the sum of the amplifier source resistance, the d.c. resistance of any series choke from a crossover network, and its own d.c. resistance. (The representation of internal resistance by an external resistor of equivalent value is standard practice for generator diagrams.)

The d.c. resistance of the series choke is likely to be about 0.5 ohm. The source resistance of the amplifier, with a damping factor as low as 4, will be 2 ohms. The total resistance seen by the speaker is then 8.5 ohms.

Eliminating the choke (a component which is sometimes severely frowned upon) gives us a reduction from 8.5 ohms to 8 ohms. Doubling the damping factor (halving the source resistance) gives us another sweeping reduction, to 7 ohms. In brief we must remember that, even with the speaker terminals shorted out by heavy copper wire of .001 ohm resistance, the smallest braking resistance we can ever achieve is 6.001 ohms. There is thus little to be gained by worrying about small resistive components in the speaker line, or by increasing the damping factor to astronomical values.

2) If we connect a second, identical speaker in series, the total internal d.c. resistance is increased to 12 ohms. But the ratio between resistance and reactance remains the same, as we now have a 16-ohm system, and the damping is unchanged. (Each 8-ohm voice-coil may be thought of as one-half of a 16-ohm voice-coil.) The series connection is perfectly good practice.

3) Another well quoted misconception relates to the fact that the coupling, at bass frequencies, between an infinitely baffled cone and the air into which it radiates decreases as the frequency is lowered, and that this decrease is compensated by progressively increasing speaker cone velocity, as discussed previously.

The belief has somehow gained ground that the loss in acoustical coupling referred to has to do with the low bass regions only, below one or two hundred cps, and that the compensating increase in cone velocity is related to speaker resonance: that is, that the resonant peak is used to "fill in" the acoustical losses.

Actually the air-load resistance presented to the cone decreases with frequency at an orderly rate (a factor of 4 per octave), below a frequency which is a function of the cone diameter—for a 12-inch speaker about 800 cps. No change in this progressive loss occurs in the extreme low bass. The theoretically ideal compensation for the decrease in air load resistance would be provided by a purely mass-controlled mechanical system, without resonance, which would dictate a doubling of cone velocity for each lower octave. (The electrical analogy is a purely inductive circuit—for the same applied voltage, current through the choke will double with each lower octave, due to the progressively decreasing inductive reactance.) Such a system is a non-existent entity, but if the speaker's resonant peak is properly damped the mechanical system acts as if it were purely mass-controlled at frequencies above resonance, and the proper compensation is provided.

4) As the frequency of the input signal to a loudspeaker is lowered in the direction of resonance, the electrical impedance of the speaker rises far above its nominal value, perhaps by 5 or 6 times. With a high-damping-factor amplifier the voltage across the speaker remains essentially constant, involving a severe drop in the electrical power drawn from the amplifier; with a lower damping factor the drop in electrical bass power is less severe; and with an even lower damping factor electrical power may remain constant, or may increase towards resonance. That value of damping factor which achieves the most uniform *acoustical* output and optimum performance is not tied to a condition of uniform electrical power, but is a function both of the particular speaker used, and of its conditions of mounting. While high damping factors are generally most suitable for horn or resonant-type systems, the same is not necessarily true for direct-radiators. No special virtue can be attached to that value of damping factor which produces constant voltage, constant power, or some intermediate type of relationship betwen amplifier output and frequency, if speaker system performance is unknown.

I would like to express my appreciation to Dr. J. Anton Hofmann for his patient reading of the draft of this article and for his valuable suggestions. Æ

Another Look at Acoustic Suspension

EDGAR M. VILLCHUR*

In order to clarify some apparent misunderstandings about the functioning of the acoustic suspension principle, its originator reiterates some of the basic philosophy and adds some supporting information.

During the past year an increasing number of articles about the type of speaker system known as "acoustic suspension" or "air suspension" have appeared, whether or not these terms were actually used.

An article[1] in the March, 1959, issue of this magazine attacks almost every point which the writer has used as a theoretical basis for explaining the acoustic suspension design. At the same time the authors describe another high-compliance-speaker/sealed-enclosure combination whose design they apparently justify on other grounds. Without taking up each point in turn, the writer felt that this would be a good opportunity to review some of the basic principles of the acoustic suspension system, and perhaps prevent some misconceptions from getting started.

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[1] Robert C. Avedon, Wayne Kooy, and Jack E. Burchfield, "Design of the wide-range ultra-compact Regal speaker system," Audio, March, 1959, pg. 22.

One statement in particular is made in the above-mentioned article which, if true, destroys as invalid the entire basis of the acoustic suspension speaker system. This is the statement that the air in the cabinet of an acoustic suspension system is significantly non-linear.

The first aim of the acoustic suspension design, over and above uniformity of frequency response, compactness, and extension of response into the low-bass range, is to reduce significantly the level of bass distortion that had previously been tolerated in loudspeakers. This is accomplished by substituting an air-spring for a mechanical one. If, as claimed by Messrs. Avedon, Kooy, and Burchfield, the sealed cushion of air in the small cabinet is in actuality less linear than a good mechanical suspension, the writer has been barking up the wrong tree. Replacing a non-linear element—the elastic restoring force of the mechanical suspensions of a speaker—with another element even more non-linear certainly does not put us ahead. Here is the first published theoretical objection to the acoustic suspension system, that the writer is aware of, which is really germaine to the subject and which, if correct, invalidates the whole idea.

Linearity of the Acoustic Suspension System

There are three basic types of speaker mounting for bass reproduction—the horn, the resonant enclosure (bass reflex, acoustical labyrinth, and so on) and the direct-radiator baffle.

Once a baffled direct-radiator system is chosen, it should be clearly understood that bass performance depends exclusively upon cone excursion, assuming a rigid cone. Knowledge of the distance, damping characteristics, and linearity of the motion of a given speaker cone—in short, knowledge of the position of the cone at every instant of time—will enable us to describe bass performance without knowing the size of the cabinet. This is like saying that knowledge of the diameter and r.p.m. of the wheels of a

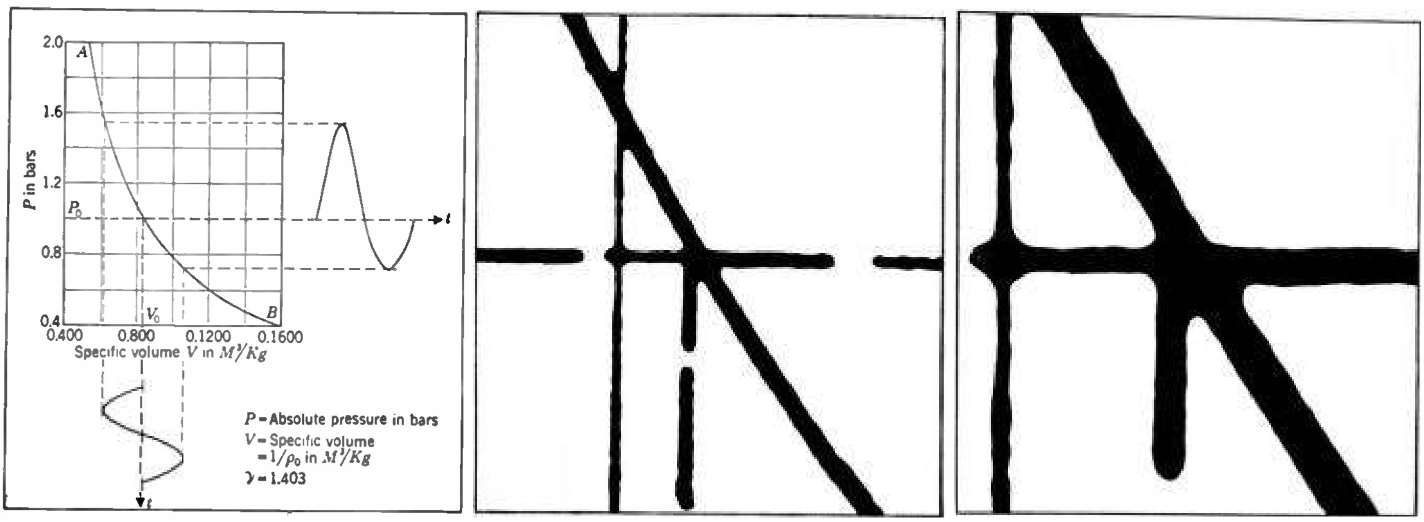

Fig. 1. (A), left. Graph showing non-linearity of air when volume is compressed and expanded by a large amount—in this case approximately ± 25 per cent. (P_o = normal atmospheric pressure.) The curvature of the transfer characteristic produces the wave distortion shown. (B), center. Enlarged center portion of the same graph. Note that the curvature of the transfer characteristic is all but undetectable. (C), right. Further enlargement of the center portion. Although this section of the transfer characteristic (the heavy diagonal line) still represents a volume change of ± 5 per cent, the curvature for such a small change is not apparent. The actual maximum air volume change in an AR-1 speaker cabinet is ± 0.75 per cent, one-sixth of the section shown.

(By permission from "Acoustics," by Beranek. Copyright 1954, McGraw-Hill Book Co. Inc.)

vehicle will enable us to predict road speed, without knowing the horsepower of the engine, the number of cylinders, or other side matters.

Cone amplitude is dependent (assuming that the speaker is capable of sufficient excursion) on the bass resonant frequency of the woofer as mounted; response drops 12 db per octave below the resonant frequency. Linearity, or absence of distortion, depends primarily on the nature of the elastic restraint seen by the cone, and on the homogeneity of magnetic flux over the voice-coil path, the latter normally taken care of by voice-coil overhang.

The current commercial speakers employing the acoustic suspension system—in which the overwhelming part of cone restoring force is due to the air-spring—have resonant frequencies representative of the lowest in the field (the AR-1, for example, has a mounted resonant frequency of 43 cps). There are no published technical arguments, to the writer's knowledge, which could lead anyone to expect any less excursion capability from a speaker in a 2-cubic-foot enclosure, with a given resonant frequency as mounted, than from a speaker in a 15-cubic-foot cabinet, where the two systems exhibit the same final resonant frequency. The speaker cone is obviously not able to sense the difference between mechanical suspension stiffness and air-pillow stiffness, ignoring questions of linearity, when it is deciding how far to move in response to a given applied force.

Until the appearance of the above-mentioned article, there has also been nothing in the literature which would lead one to expect less linearity from a speaker utilizing an air-spring. On the contrary, both theory and practice clearly point to a significant increase in

Fig. 2. Fiberglass, in weighed-out amounts, being used to fill an AR speaker cabinet.

linearity and reduction in bass distortion. Now, however, we have on record an argument that does predict higher rather than lower bass distortion. The argument is stated as follows:

> "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.
>
> ". . . It is extremely difficult to make this process [*air compression and rarefaction at an audio rate*] anything other than adiabatic. The physics text will also show that adiabatic compression is inherently non-linear . . . 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.
>
> ". . . the non-linearity of the air suspension overshadows any reduction in distortion derived from a throw longer than ⅜-in."

Linearity of the Air Spring in the Speaker Enclosure

The interior volume of the enclosure of an AR-1 or AR-3 speaker system, making approximate allowance for the space taken up by the speaker itself and by the nine reinforcing braces, is 1.5 ft.[3]. The effective cone area for a 12-inch speaker may be calculated on the basis of a 10-inch diameter flat piston, as 78.5 sq. in.[2]. Therefore, when the cone is undergoing peak-to-peak excursions of ½ inch, the enclosure volume is alternately decreased and increased by 19.6 cu. in., the volume taken up by a center-to-peak excursion of the cone. This is readily calculable, in relation to the 2592 cu. in. of the enclosure, as representing a volume change of 0.75 per cent. Such a volume change, in turn, can be converted to terms of linearity, either by the well known gas equation[2] or by a chart[3] relating pressure and volume of air above and below atmospheric pressure.

The gas equation tells us that the pressure of our enclosed body of air will be inversely proportional to its volume raised to the 1.4 power. If we halve the volume, the pressure will not be merely doubled, but will be increased by $2^{1.4}$, or 2.64. The non-linearity indicated is significant. On the other hand, when the volume is changed by a very small amount (in this particular case 0.75 per cent) the variation of pressure change from the inverse of volume change will be insignificantly small.

When the speaker cone moves back half an inch and decreases the air volume to 0.9925 of its former value, the air pressure would, in the perfectly linear case, increase by a factor of 1.00755. Instead, in the non-linear case, it increases by a factor of $(1.00755)^{1.4}$, or 1.01. Raising the former number to the 1.4 power hardly changes it. The non-linear aberration involved—the difference between 1.00755 and 1.01—is of the order of one-fourth of one per cent, a totally insignificant figure in the field of loudspeakers.

One may achieve a better intuitive understanding of the mathematical principle illustrated above by an exercise in which different numbers are raised to a given power. For example, 10^2 equals 100; the ratio between the base number and its square is ten to one. 2^2 equals 4, and the ratio has decreased to only two to one. As the number being squared approaches one the ratio between it and its square also decreases, until, when the number is one, it is *equal to* its own square. This is why the number 1.00755, when raised to the 1.4 power, is increased by only 0.24 per cent.

The preceding analysis can also be represented graphically, as in (A), (B), and (C) of *Fig.* 1.

Adiabatic and Isothermal Pressure Changes

The non-linearity described is characteristic of a volume of air subjected to pressure changes when there is no chance for the heat generated to flow out of the system. When a volume of air is compressed and heated, the accompanying rise in heat and hence in molecular activity increases the number of molecular collisions, and the air is effectively stiffened. Such a pressure change, accompanied by a change of temperature, is called *adiabatic*. Pressure changes associated with sound in free atmosphere are adiabatic, and sound pressure changes in an unlined speaker cabinet are also adiabatic, because the walls of the cabinet are unable to conduct the heat generated by these pressure changes quickly enough to the outside atmosphere.

If the speaker cabinet is filled with the proper kind and amount of material such as fiberglass, the air of the cabinet is exposed to a very large area of material, provided by the interstices and convolutions of the fiberglass. The generated heat of the compressed air can flow very quickly—this means within the period of the audio frequency pressure change—into the fiberglass, and back again. Thus the temperature of the air itself, and the corresponding molecular activity, remains constant. Such pressure changes are called *isothermal*. The isothermal behavior of air in a cavity filled with the proper absorbent material has been known for years, and is described in the literature.[4]

(*Continued on page 75*)

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[2] $P = \frac{K \text{ (a constant)}}{V^{1.4}}$ (For certain gases, including air).

[3] Leo L. Beranek, "*Acoustics*," pg. 274, McGraw-Hill Book Co., 1954.

[4] Ibid., pg. 4, 220.

ACOUSTIC SUSPENSION

(*from page 25*)

Thus even the tiny amount of distortion associated with air non-linearity is not present in an acoustic suspension system designed according to the writer's patent.[5] It is true that the primary purpose of the fiberglass is not to eliminate distortion due to air non-linearity (which is so small in amount as to be without significance to begin with), but it is interesting to note that even this small amount of distortion does not remain. The air-cushion of the acoustic suspension speaker enclosure has been described, and I believe with accuracy, as a near-perfect spring.

Further Effects of Fiberglass in the Cabinet

The article referred to also contains the following statements:

> "Curves were run on a sealed box system . . . with and without the interior of the box filled with sound absorbent material. . . . 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."

This, too, is in direct contradiction to theory and practice described by the writer.

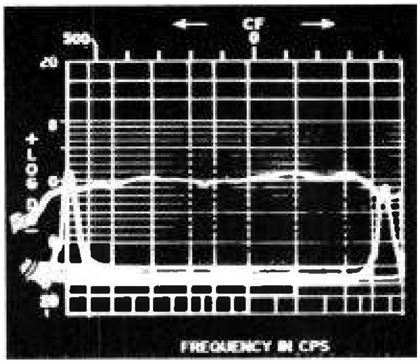
The change from adiabatic to isothermal conditions created by the fiberglass, as we have seen, decreases the stiffness of the enclosed air by a factor of 1.4. This is the equivalent of saying that the effective cubic volume of the cabinet is increased 1.4 times. The result is a reduction of the resonant frequency of the speaker as mounted (assuming that at least three-quarters of the elastic restoring force is due to the air cushion) by about 16 per cent.

Such a difference would have to show up clearly in the response curve. Since no difference in bass response was noted by Messrs. Avedon, Kooy, and Burchfield, it must be assumed that their (unspecified) sound absorbent material was not of the type that created isothermal conditions in the amounts used.

At the Acoustic Research plant the amount of fiberglass that is used in the cabinet (see *Fig.* 2) is determined by measurements of the bass resonant frequency of the system with and without the fiberglass; that amount of fiberglass which reduces the resonant frequency from approximately 51 cps to 43 cps is the correct amount for the AR-1 or AR-3.

In addition to the function just described, the fiberglass damps out the familiar standing-wave resonances that

[5] E. M. Villchur, "Sound Translating Devices," U. S. Patent No. 2,775,309, Dec., 1956.



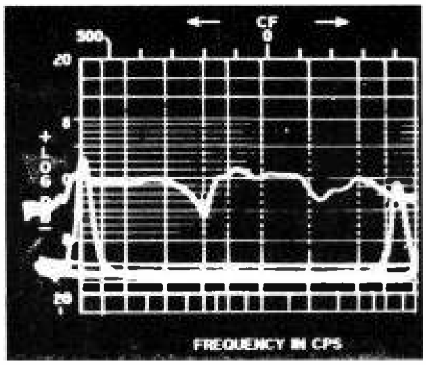


Fig. 3. (A), left. Automatic frequency-response trace, on a linear 1000-cps sweep, of a standard AR-2. Marker pips are at 350 and 1350 cps. (AR's anechoic chambers are not suitable for measurements at low frequencies.) Horizontal divisions are in 1-db steps. (B), right. Response trace of the same speaker system over the same frequency range, with the fiberglass removed from the cabinet. All other measurements are the same as in (A). Display is on a Panoramic Sonic Analyzer, model LP1a.

would form in a rectangular wooden enclosure, resonances which would create easily measurable peaks and dips in the frequency response of an otherwise smooth loudspeaker. In *Fig.* 3, (A) shows the acoustic frequency response trace, recorded automatically, of a standard AR-2 woofer in the range between 350 and 1350 cps. (AR's indoor measuring facilities are not adequate at lower frequencies) (B) in *Fig.* 3 is the response trace of the same speaker system with the fiberglass removed from the cabinet, measured in the same anechoic chamber with all conditions held constant. The response irregularities of the latter are obvious.

Enclosure Leaks

The three authors are quoted again:

"Need the cavity behind the driver be sealed absolutely air tight, resorting even to a stethoscope to determine air leaks? . . . It is seen that nowhere was the output reduced by more than 1 db with a total of 8 ¼-in. holes. . . . Therefore, a well crafted box with reasonable joinery is all that is necessary."

Fig. 4. Stethoscope check for air leaks. The speaker is being driven by a 20-cps signal.

Here the writer must plead half-guilty to the implied charges of over-design. The extreme care used in conserving a good acoustic seal at the AR plant, care which involves gaskets between the speaker flange and the cabinet, and stethoscope checks (see *Fig.* 4), is a "touch-up" rather than a basic operation. In return for a relatively inexpensive additional procedure, we receive freedom from the danger of slightly decreased low-bass output, and more important, freedom from a sort of hissing noise that would accompany low-frequency program material of high power when leaks are present.

Conclusion

Messrs. A., K., and B. state:

"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 should be clear by now that the authors do not have to worry about nonlinearity of the air in the cabinet, and I can find no other disadvantages to the box described in their article that have to do directly with the fidelity of sound reproduction rather than with efficiency.

There is a group of old wives' tales in the audio field which are not subject to the attack of reason, but only give way to time. For example, there is the old saying, almost forgotten by now: "Triodes are always sweeter-sounding than pentodes." Another oft-repeated maxim states that bass reproduction from a small speaker enclosure is inevitably inferior to that from a large one. In my opinion, the support for this principle derives merely from its own repetition. Æ