Feb. 17, 1931. E. L. NORTON **1,792,655**

SOUND REPRODUCER

Filed May 31, 1929 2 Sheets-Sheet 1

FIG. 1

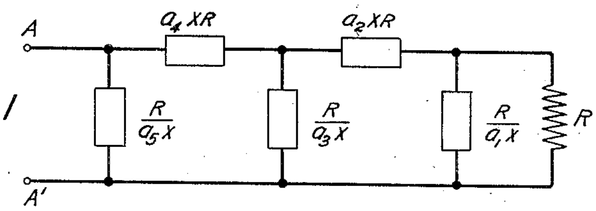


FIG. 2

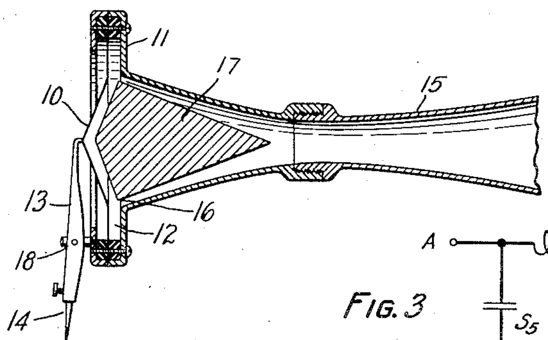


FIG. 3

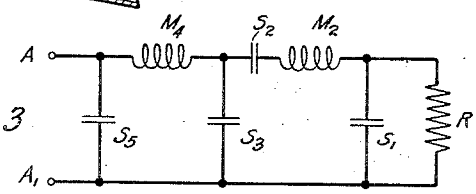


FIG. 4

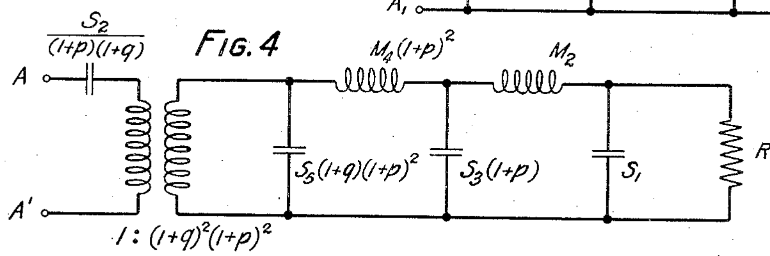


FIG. 7

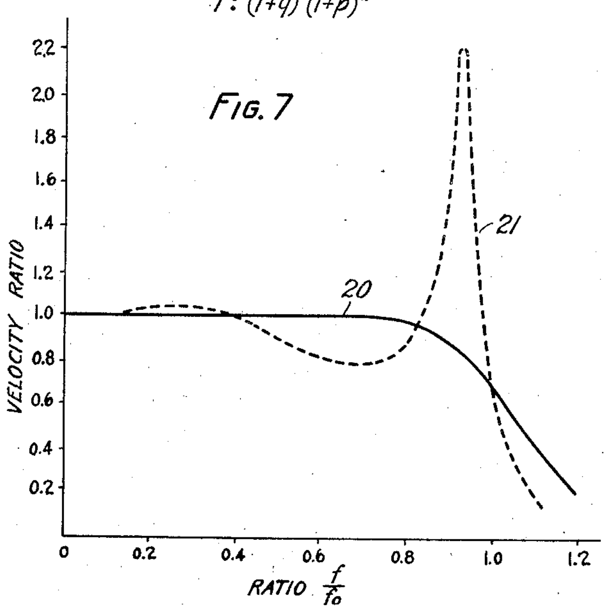


FIG. 5

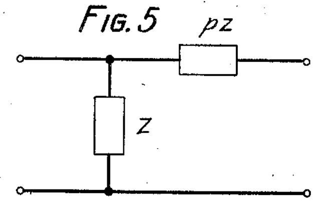
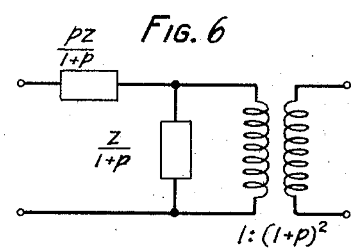


FIG. 6



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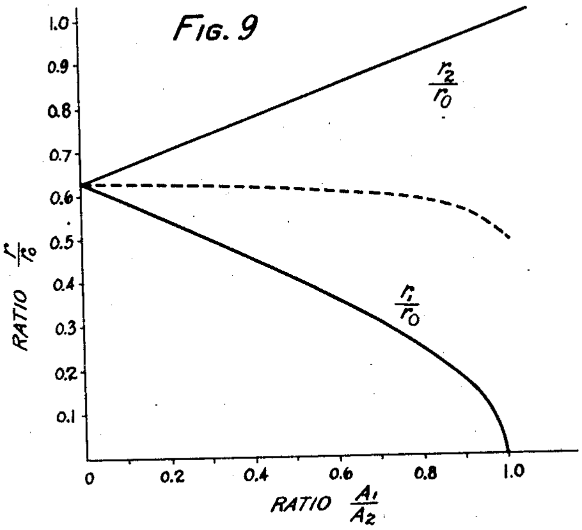

FIG. 9

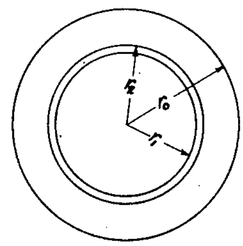

FIG. 8

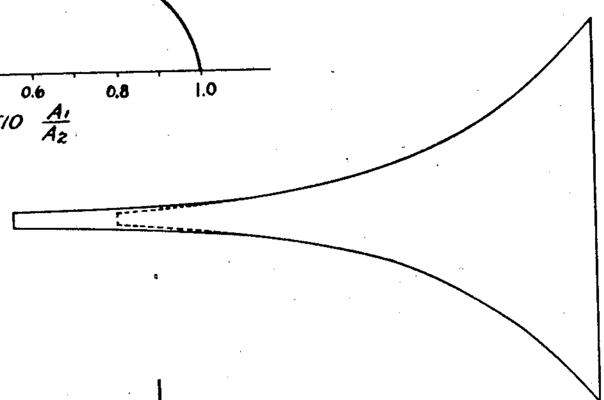

FIG. 10

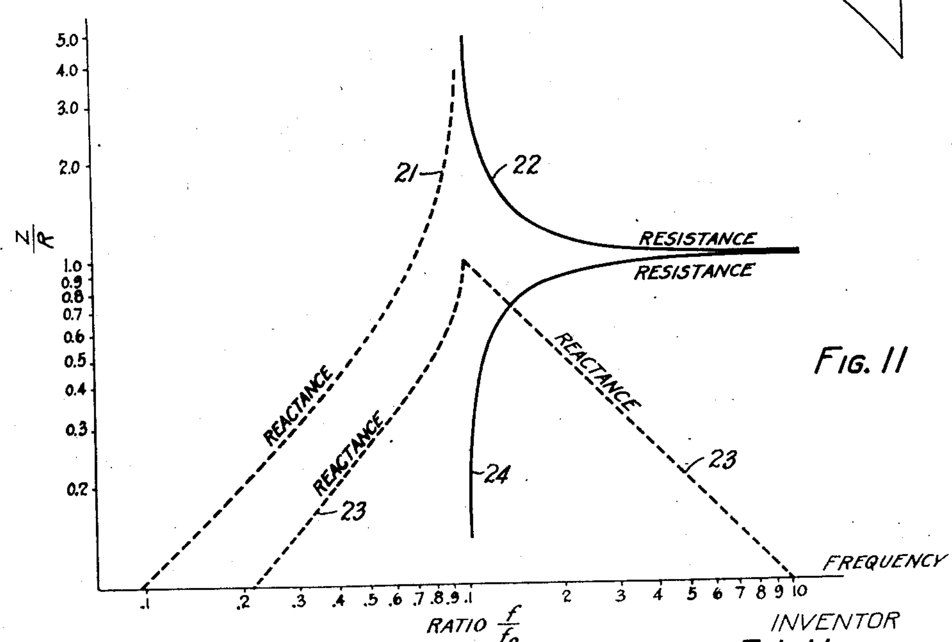

FIG. 11

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UNITED STATES PATENT OFFICE

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SOUND REPRODUCER

Application filed May 31, 1929. Serial No. 367,487.

This invention relates to sound reproducers, and more particularly to reproducers of the type in which a horn driven by a diaphragm acts as the sound radiator. It has for its principal object the improvement of the response characteristics of devices of this type by the elimination of variations due to the resonances of the vibrating system.

Another object is the elimination of mechanical reactance from the air column of the horn which constitutes the acoustic load upon the diaphragm, such reactance ordinarily entering into the resonances of the system and contributing to the unevenness of the response characteristic.

It has been recognized that the vibratory system of a sound reproducer, including the horn, constitutes a serially coupled chain of mass and elastic elements and may be regarded as a mechanical transmission line through which the vibrations are propagated to a terminal load represented by the radiation resistance of the horn mouth. The horn itself constitutes a true wave transmission line in which the mass and elastic properties are distributed in a continuous manner along its length. The diaphragm and its driving mechanism, together with the air chamber customarily interposed between the diaphragm and the horn, constitute a system in which the masses and the elasticities are substantially segregated, or lumped, in separate elements and in which, therefore, a true space wave does not exist. However, for a limited frequency range this lumped system can act in a manner analogous to a true transmission line and, by the proper proportioning of the elements, it can be made to transmit freely all vibrations in the range of frequencies needed for the accurate reproduction of speech or music.

The proportions heretofore used for the vibrating elements to secure free transmission in a single broad band covering the essential speech range of frequencies, are such that the vibrating system is, in effect, a chain of uniform mass elements coupled serially by springs of uniform elasticity, the values of the masses and the elasticities being determined by the range of frequencies to be transmitted and the impedance of the acoustic load. In such a system the resonances of the vibratory elements are substantially damped by the acoustic load but are not so completely damped that the accentuation of the tones corresponding to the resonance frequencies is eliminated. The principles underlying the design and operation of systems of this type are clearly explained in U. S. Patent 1,730,425, issued October 8, 1929 to H. C. Harrison.

In accordance with the present invention the masses of the elements of the coupled system are given progressively increasing values and the elasticities are given progressively decreasing values from one section to another, the mass increments and the diminishing values of the elasticities being substantially in accordance with a definite law, whereby the effects of resonance are completely suppressed. For the carrying out of the invention in its best manner it is desirable that the vibratory elements should closely approximate simple masses and simple elasticities at all frequencies in the speech range and that the horn should have a resistive impedance as free as possible from reactance. A feature of the invention directed to this end is a modified form of the air chamber in front of the diaphragm. The opening into the horn throat is in the form of an annulus the diameter of which bears a particular ratio to the diaphragm diameter whereby the mass effect of the air in the chamber is annuled and the chamber is made to act as a pure elasticity. Another feature of the invention is a horn of novel form having the property that its impedance is completely free from reactance at all frequencies in its useful range. The cross sectional area of the sound passage varies in accordance with two exponential functions of the length, one of the functions having increasing values and the other diminishing. In general it resembles the logarithmic horn, but differs therefrom in that the throat portion is longer and less tapered.

In the usual constructions the diaphragm does not correspond to a pure mass, but rather to a simple mass in combination with an elastic restraint due to the edge support. This elastic restraint is also an undesired reactance which tends to produce irregularities in the response characteristic and a further feature of the invention relates to the elimination of these irregularities by the adoption of special proportions in the elements of the vibrating system.

In the accompanying drawings:

Fig. 1 is a schematic diagram illustrating the basic impedance relationships amongst the elements of the vibrating system of a sound reproducer of the invention;

Fig. 2 shows a sound reproducer of the stylus driven type in connection with which the application of the invention will be described;

Figs. 3 and 4 are schematic impedance diagrams corresponding to Fig. 2;

Figs. 5 and 6 are schematic diagrams illustrating a theorem used in explaining the invention;

Fig. 7 shows the type of response characteristic of the systems of the invention;

Fig. 8 is a schematic figure relating to the feature of the air chamber design;

Fig. 9 is a set of curves used in the determination of the air chamber opening dimensions;

Fig. 10 shows the form of the acoustical horn of the invention; and

Fig. 11 shows by means of curves the impedance characteristic of the horn of Fig. 9.

The schematic diagram of Fig. 1, representing the impedance relations of the sound reproducers of the invention is drawn in accordance with electrical conventions, but by the well known analogy between mechanical and electrical systems, it equally well represents the impedance relations of a mechanical system. The system illustrated is in the nature of a wave transmission line comprising in combination with an energy dissipative load resistance R, two "line" elements of impedances A_4XR and A_2XR, and three "coupling" elements

$$\frac{R}{A_5X}, \frac{R}{A_3X}, \text{ and } \frac{R}{A_1X}.$$

The terms "line" and "coupling" correspond respectively to the electrical terms "series" and "shunt" and serve to distinguish the functions of the different elements. An oscillatory motion is modified in velocity as it traverses a coupling element and in force as it traverses a line element. A coupling element transmits the force impressed on it undiminished to the next line element and a line element transmits the velocity of the motion without change. The transmission line to which the vibrating system of a sound reproducer corresponds is generally of a very simple type, comprising line elements in the form of simple masses, and coupling elements in the form of springs. The energy dissipative load is provided by the air in front of the sound radiating device, or, if the sound radiator is a horn, by the air column in the horn in combination with the radiation from the mouth.

The analysis of the motion in a system of this sort is greatly facilitated by the concept of mechanical impedance. This quantity provides a direct measure of the forced oscillatory velocity in a body or system in response to an oscillatory force applied thereto. It is defined as the ratio of the applied force, assumed to have a steady effective value, to the velocity of the body or of the system at the point of application. It is a vector quantity defining both the magnitude of the velocity and its phase difference from the applied force. The impedance of a simple mass element is directly proportional to frequency, the velocity lagging 90 degrees behind the applied force. Expressed mathematically the impedance is equal to $j\omega M$, where ω is 2π times the frequency, M the mass, and j the cusmary operator $\sqrt{-1}$. The impedance of a massless spring measures the "extension" velocity, that is, the velocity of the one end relatively to the other, in response to an applied force. Its value is $-j\frac{S}{\omega}$, where S is the elasticity of the spring, and where the negative operator indicates that the velocity leads the applied force by 90 degrees. Following the established practice in electrical theory the real component of the vector quantity denoting the impedance is termed "resistance" and the imaginary component is termed "reactance". The impedances of complex coupled systems can be expressed in terms of the impedances of elements by the same rules as apply to electrical systems. For example, a mass element supported by a spring from a rigid abutment is a series combination, the impedance of which is obtained by adding the component impedances, while a free mass to which the force is applied through a spring is a parallel combination.

In the system of Fig. 1 the line and the coupling impedances are pure mechanical reactances, i. e. combinations of masses and elasticities in which there is no energy dissipation. The impedance values are made up of three factors, a common factor X which is an imaginary numerical quantity and which expresses the frequency variation of the impedance, a common factor R equal to the resistive impedance of the dissipative load, and numerical coefficients a_1, a_2, etc. giving the relative magnitudes of the impedances. It is to be noted that the impedances of the line and the coupling elements are inversely related with respect to their frequency variations; thus the line elements may be simple masses having impedances proportional to frequency and the coupling elements may be simple elasticities the impedances of which are inversely proportional to frequency.

The response characteristic of the system, by which is meant the frequency variation of the magnitude of the velocity in the load impedance R for a constant oscillatory input at the terminals AA′, may be controlled by varying the impedances of the elements. The energy absorbed by the load resistance R is proportional to the square of the velocity therein and, therefore, to the square of the response at any particular frequency. In accordance with the invention the impedances are given such relative values that the response of the system for a constant oscillatory input is proportional to the quantity

$$\frac{1}{\sqrt{1+|X|^{2n}}} \quad (1)$$

where n is the total number of the line and coupling impedances. This result is obtained by giving the coefficients a_1, a_2 etc. the values

$$a_1=\sin\frac{\pi}{2n},$$

$$a_2=\frac{\sin\frac{3\pi}{2n}}{\cos^2\frac{\pi}{2n}}, \qquad (2)$$

$$a_r=\frac{\sin\frac{2r-1}{2n}\pi\,\sin\frac{2r-3}{2n}\pi}{a_{r-1}\cos^2\frac{r-1}{2n}\pi},$$

$$a_n=n\,\sin\frac{\pi}{2n}.$$

When n is equal to 5, as in Fig. 1, the numerical values are

$a_1=.309$,
$a_2=.896$,
$a_3=1.380$,
$a_4=1.695$,
$a_5=1.545$.

In the type of system shown in Fig. 1, which is characterized by a shunt, or coupling, element at the input end the expression for the response characteristic gives the ratio of the output velocity to the input velocity, that is it shows the variation of the output velocity with frequency for a constant input velocity.

The mathematical steps in the determination of the coefficient values are discussed at length in my copending application Serial No. 355,607, filed April 16, 1929, and need not be reproduced here.

The application of the foregoing principle to a sound reproducer will be described in connection with the device illustrated in Fig. 2. This figure shows in section a stylus driven phonograph reproducer the elements of the vibrating system of which comprise a diaphragm 10, clamped at its edge within a casing 11, an air chamber 12, formed between the diaphragm and the front face of the casing, a stylus arm 13 pivoted at 18 on the casing, and a stylus 14. When used for reproduction from a phonograph record, the stylus velocity is determined only by the speed of the record and the amplitude of the groove and the accuracy of reproduction depends upon the uniformity with which this velocity is transferred to the air in the horn. The response characteristic therefore corresponds to the frequency variation of the ratio of the output to the input velocity. The air chamber 12 opens into a horn 15 which constitutes the terminal acoustic load. In the particular device illustrated the air chamber communicates with the horn by an annular opening 16 around the base of a solid conical block 17 which is mounted in an expanded portion of the horn throat. The features of the horn design and of the throat opening by which a truly resistive and constant load impedance is obtained will be discussed later.

The schematic arrangement of the vibrating system is shown in Fig. 3 in which the conventions of Fig. 1 are employed to distinguish the line and the coupling elements and in which also the electrical symbols for inductance and capacity are used to indicate the corresponding quantities of mass and elasticity respectively. It should be remembered, however, that although the impedances of a capacity and of an elasticity vary in the same way with frequency, their magnitudes are inversely related, the correspondence being between elasticity and the reciprocal of capacity.

The resistance R represents the terminal load constituted by the impedance of the horn. The impedance of the air chamber is represented by a simple elasticity S_1 in shunt to the load. Ordinarily the air chamber, due to the effect of the air mass, does not act like a simple elasticity except at low frequencies, but it has been found that, by using an annular opening of proportions to be described later, the mass effect can be eliminated and the air made to act as a pure elasticity. The diaphragm is represented by a mass M_2 in series with an elasticity S_2, the elasticity being due to the edge support of the diaphragm. By forming the diaphragm of thin sheet metal and embossing it in a suitable manner to make the center portion rigid, it may be made to approximate closely to a simple mass, the edge elasticity being so small that its effect, to a first approximation, may be neglected. The stylus arm is represented by a series mass M_4 and a shunt elasticity S_3. The mass M_4 is the effective mass of the arm at the center of the diaphragm due to its moment of inertia about the pivot support 18, and the elasticity S_3 is the effective value of the bending elasticity of the arm transferred to the same point. The elasticity S_5 is that of the stylus likewise transferred to the axis of the diaphragm, the value being transformed in accordance with a transformation ratio introduced by the lever action of the stylus arm. The stylus and stylus arm elasticities may be taken as the ratio of force to linear deflection at the stylus point and at the end of the stylus arm respectively when the arm is rigidly clamped at the pivot.

If the edge elasticity S_2 of the diaphragm be ignored, as it may be in the case of an ideal piston diaphragm, the system then includes only simple masses as line elements and simple elasticities as coupling elements, the former having impedances proportional to frequency and the latter having impedances inversely proportional to frequency.

The values of the masses and the elasticities required to make the system conform to the general type of Fig. 1 are determined as follows: The factor X is evidently proportional to frequency and, since it is necessarily a numerical quantity, may be written as

$$X=j\frac{f}{f_0}$$

where f_0 is the frequency at which X is numerically equal to unity. The response is then proportional to the quantity.

$$\frac{1}{\sqrt{1+\left(\frac{f}{f_0}\right)^{10}}} \quad (3)$$

the index 10 appearing since there are in all five impedance elements. With a constant resistive horn impedance, the sound energy delivered to and radiated by the horn is proportional to the square of the above quantity when the input velocity is constant. At frequencies greater than f_0 the response becomes very small while for lower frequencies it is substantially equal to unity. The frequency f_0 is therefore an important design parameter since it marks the limit of uniform response. Comparing the impedances of the corresponding elements of the systems of Figs. 1 and 3, the values of the masses and the elasticities required to give the desired response characteristics are readily obtained. These values are as follows:

$$S_1=2\pi\frac{R}{a_1}f_0,$$

$$M_2=\frac{a_2R}{2\pi f_0},$$

$$S_3=2\pi\frac{R}{a_3}f_0, \quad (4)$$

$$M_4=\frac{a_4R}{2\pi f_0},$$

and

$$S_5=2\pi\frac{R}{a_5}f_0.$$

The values of the masses and the elasticities are thus fully determined when the horn impedance is known and the limiting frequency of uniform response has been assigned.

The form of the response characteristic is illustrated in Fig. 7 in which the magnitude of the ratio of the input to the output velocity is plotted against the ratio

$$\frac{f}{f_0}.$$

The full line 20 shows the response of a five element system in accordance with the invention. For comparison purposes the dotted line 21 shows the response of a similar system having uniform line masses and uniform coupling elasticities. Both curves are drawn on the assumption that the elements are quite free from energy dissipation, the effect of which is to add a slight downward slope with frequency to the full line curve and to diminish considerably the high peak in the dotted curve. The characteristic difference between the two types of response is not changed.

The foregoing formulæ are developed on the assumption that the edge elasticity of the diaphragm is negligibly small, but in most practical cases it is desirable that this quantity should be taken into account in proportioning the system. By means of a theorem of equivalence which is illustrated in Figs. 5 and 6 it can be shown that the system of Fig. 3, including the elasticity S_2, is equivalent to a simpler related system involving only simple line masses and simple coupling elasticities to which is connected in series at the input end an elasticity proportional to S_2 but of a modified value. The presence of this series elasticity does not modify the input velocity, although it may increase the input impedance of the system and, especially at low frequencies, cause the stylus point to react more strongly against the sides of the record groove. The simplified system may be proportioned in the manner indicated above and the necessary modifications may be made in the actual system in accordance with the relationships establishing the equivalence.

Fig. 5 shows a combination of two similar impedances of values Z and pZ, one in shunt and the other in series, the series impedance being to the right of the shunt impedance. Regarding this as a portion of a transmission line, the combination may be replaced by that shown in Fig. 6 which comprises a series impedance

$$\frac{pZ}{1+p}$$

to the right of which is a shunt impedance

$$\frac{Z}{1+p}$$

and an ideal transformer of impedance ratio

$$1:(1+p)^2.$$

By this equivalence the relative positions of the series and the shunt impedances are changed and their values modified by constant numerical factors. The theorem may be applied to the system of Fig. 3, first to the combination S_2, S_3 and then again to the combination of S_5 and the elasticity appearing in series with M_4 as the result of the first application. The ideal transformers may of course be moved to any point in the system without affecting the equivalence provided appropriate changes in the impedance values are made.

As the result of the steps outlined above the system of Fig. 4 is obtained which is the precise equivalent of that of Fig. 3 in respect of its transmission properties. The first step has changed the elasticity S_3 to the value $S_3(1+p)$, or S_3+S_2. The second step has introduced a new transformation ratio of $1:(1+q)^2$, where

$$q=\frac{S_2}{(1+p)S_5}, \quad (5)$$

that is, q is equal to the ratio of the modified elasticity appearing in series with M_4 after the first step to the shunt elasticity S_5. The two transformations are represented in the figure by a single transformer having a ratio equal to their product.

The portion of Fig. 4 to the right of the transformer may be computed rigidly in accordance with the method described in the earlier paragraphs to obtain the response characteristic defined by Formulæ 1 and 3 and the impedance coefficients of the actual physical system readily follow.

The design of an actual sound reproducer of the type shown in Fig. 2 will now be described. The most convenient starting point is the assignment of the frequency range of uniform response and the selection of a diaphragm of practical dimensions from a manufacturing standpoint. Let it be assumed that the limiting frequency, f_0, is 4000 c. p. s. and that the diaphragm has a diameter of 4.5 cms. and a mass of 0.20 grams. Such a diaphragm may be constructed from aluminum or aluminum alloy sheet about .0035 cm. thick, the central portion being embossed in conical or other form to give rigidity. The edge stiffness S_2 may be taken as 5×10^6 c. g. s. for a diaphragm of these dimensions.

The effective impedance that the horn should have is found from the relationship

$$M_2=\frac{a_2R}{2\pi f_0}, \quad (6)$$

the coefficient a_2 having the value 0.896 corresponding to a five element system. Substituting the values assumed for M_2 and f_0 in Equation (6) the value of R is found to be 5600 c. g. s. The required air chamber elasticity is given by the formula

$$S_1=2\pi\frac{R}{a_1}f_0, \quad (7)$$

which gives a numerical value of 455×10^6 c. g. s. For the stylus arm elasticity we have

$$S_3(1+p)=S_3+S_2=\frac{2\pi Rf_0}{a_3} \quad (8)$$

from which $S_3=97\times10^6$ c. g. s. The stylus arm effective mass is given by

$$M_4(1+p)^2=M_2\frac{a_4}{a_2}=.378 \text{ grams.} \quad (9)$$

The relative values of S_3 and S_2 give the value of $(1+p)^2$ as 1.105, hence $M_4=3.42$ grams. Finally, the effective elasticity of the stylus is given by

$$S_5(1+q)=S_5+\frac{S_2}{1+p}=\frac{S_1}{(1+p)^2}\cdot\frac{a_1}{a_5}=82.4\times10^6 \quad (10)$$

and $S_5=77.7\times10^6$ c. g. s. The dynamical proportions of the elements having been found the geometrical forms and dimensions can be arrived at by known methods. The required horn impedance has been found to be 5600 c. g. s. This is the value effective at the diaphragm and is different from the value measured at the throat aperture due to the impedance transforming effect of the air chamber. The value of the impedance measured at the throat of a horn in general is not constant with frequency and is not a pure resistance, but with the type of horn to be described later the impedance is purely resistive within the working range and is substantially constant. This constant value is equal to $41A_1$ c. g. s., where A_1 is the throat area. Taking account of the transformation ratio due to the air chamber, as described in the hereinbefore mentioned Patent 1,730,425 to H. C. Harrison, the effective impedance at the diaphragm is given by

$$R=41A_1\cdot\frac{A_2^2}{A_1^2} \quad (11)$$

where A_2 is the area of the diaphragm. From this equation, knowing R and A_2 the required value of A_1 is found to be 1.85 square centimeters. The air chamber elasticity S_1 is determined by its volume and its diameter in accordance with the formula

$$S_1=\frac{1.41\times10^6A_2^2}{V}$$

where V is the volume. The volume necessary to make S_1 have the required value of 455×10^6 c. g. s. is found from this equation to be 0.785 c. c., which with a diameter of 4.5 cm. corresponds to a depth of .05 cm.

The proportioning of the stylus arm to have the required effective mass and elasticity is a matter of mechanical design and may be carried out by well known processes. The effective value of the stylus of about 78×10^6 c. g. s. is closely equal to that of an ordinary "full-tone" needle. This makes it unnecessary to provide any impedance transformation by the lever action of the stylus arm and accordingly the stylus arm should be designed so that the distance from the needle point to the pivot is the same as that from the pivot to the center of the diaphragm. If a more flexible needle is used the length from the needle point to the pivot should be diminished to provide the proper transformation.

In the foregoing, it is assumed that the air in the air chamber acts as a simple elasticity, but with the ordinary construction in which the horn is coupled to a central aperture this will not be the case. Due to the uniform distribution of mass and elasticity throughout the air volume the motion in the air chamber is a true wave motion in which the wave is propagated parallel to the surface of the diaphragm towards the aperture. At low frequencies the wave is so long in comparison with the longest path in the chamber that the space variation of the motion is unimportant, but at high frequencies the wave length becomes short enough so that a considerable fraction of the wave may be encompassed within the dimensions of the chamber and a nodal point may exist at the horn aperture. Under such conditions it is possible to have the response fall to zero at some frequency well within the range necessary for good reproduction of speech or music. The "distributed constant" or "mass" effect of the air, to which this action is due, can be substantially eliminated by the use of an annular opening of a properly chosen mean diameter. If the mean diameter of the annulus were equal to the diaphragm diameter, that is, if the sound were taken off from the outer edge of the diaphragm, the conditions would be just as bad as in the case of a center opening. At some intermediate value it would be expected that the conditions would be most advantageous and the frequency at which the response fails would be moved to the highest value. It has been found and can be shown mathematically that the optimum value of the mean diameter of the annulus is equal to 0.63 times the diameter of the diaphragm. With this proportion the mass effect of the air is so completely eliminated that it is negligible at all frequencies required for the highest quality of reproduction.

The determination of the proper radii for the annulus opening is illustrated by Figs. 8 and 9. In Fig. 8 r_0 represents the diaphragm radius, r_1 the inner radius of the opening, and r_2 the outer radius. In Fig. 9 the values of the ratios

$$\frac{r_1}{r_0} \text{ and } \frac{r_2}{r_0}$$

are plotted as ordinates against the values of the ratio

$$\frac{A_1}{A_2},$$

A_1 being the aperture area and A_2 the diaphragm area as before. The ratio

$$\frac{A_1}{A_2}$$

measures, as already pointed out, the impedance transformation in the air chamber and determines the effective value of the load impedance. These curves represent the solution of the equation

$$\frac{r_1}{r_0}J_1\left(3.832\frac{r_1}{r_0}\right)=\frac{r_2}{r_0}J_1\left(3.832\frac{r_2}{r_0}\right) \quad (12)$$

where J_1 indicates a Bessel function of the first order. The principles underlying the analysis from which the above equation is found are as follows: The annular opening from the air chamber is used for the purpose of insuring that the wave transmission paths from all points within the chamber to the opening shall be kept small so that even at the highest frequencies involved the maximum length of any path shall be much less than the corresponding sound wave length. As a first approximation, if an annular opening is placed at one-half the radius of the chamber the maximum distance to any point of the chamber is one-half the radius. Any other location would result in a larger maximum distance. More complete analysis shows that maintaining the wave paths at minimum length is not sufficient for the best results. For example, in the case of a uniform air chamber with the opening at half the radius the volume of air displaced in the region outside the annulus would be three times as great as that displaced in the region inside and although the wave paths may be of equal length the disparity of the air volumes would give rise to interference effects in the neighborhood of the annulus. It is thus to be expected that the optimum radius would be somewhat greater than half the radius of the chamber.

To obtain the best value of the annulus diameter the differential equation for the sound waves in the chamber may be expressed in polar coordinates to suit the circular configuration. The solution is then obtained in terms of Bessel functions and after putting in the boundary conditions of the problem it is found that the solution indicates a series of resonances at various frequencies at which substantially no power is delivered to the horn. Since it is desired to transmit a band of frequencies from some very low value up to as high a frequency as possible it is most important to suppress those resonances which occur at the lower frequencies. Each of the resonant terms in the mathematical solution has a coefficient the magnitude of which may be changed by changing the radius of the annular opening. By placing the opening at the particular point defined by Equation (12) the coefficient of the lowest frequency resonance is made equal to zero, or in other words, the lowest frequency resonance no longer occurs. For chambers of the ordinary size used in loud speakers or phonographs the second resonance occurs at a very high frequency outside of the ordinary speech range and the result is that by the elimination of the first resonance substantially uniform efficiency is maintained over the whole speech range. The dotted curve of Fig. 9 represents the mean diameter of the annulus; this has the nearly constant value of 0.63 times the diaphragm diameter.

The achievement of a load impedance which is a constant resistance and free from reactance is accomplished by the use of a horn of the type illustrated in Fig. 10. Assuming a straight axis and a circular cross section the outline of this horn is given by the equation

$$r_x = r_t \cosh bx \quad (13)$$

where r_x denotes the radius at a distance X along the axis from the throat, r_t denotes the radius at the throat, and b is a coefficient defining the rate of expansion. The above equation may also be written in the form

$$r_x = r_t \frac{e^{bx} + e^{-bx}}{2} \quad (14)$$

which shows that the radius varies in accordance with two exponential functions, one of which increases with x and the other of which diminishes. Both are equally important at the small end of the horn but at the large end the increasing term is most important. In Fig. 10 the dotted outline represents a simple exponential horn of the same expansion coefficient, only the throat portion being indicated, since the two outlines are indistinguishable at the large end. The figure indicates that the horn of the invention is somewhat longer than the corresponding exponential horn and tapers more gradually at the throat. The impedance characteristic of the horn is shown by the curves of Fig. 11 in which the ratio of the impedance Z to the final steady value R is plotted against frequency. Curves 21 and 22, plotted to logarithmic scales, correspond to the horn of the invention and curves 23 and 24 to the simple exponential horn. The solid lines represent resistance values and the dotted lines reactances. Logarithmic scales are used for both frequency and impedance, the frequency values being given in terms of the ratio

$$\frac{f}{f_c}$$

where f_c is the cut-off frequency of the horn. At this frequency, which is given by

$$f_c = \frac{bc}{2\pi},$$

c being the velocity of sound, the impedance changes from resistance to reactance. At higher frequencies the impedance is purely resistive and at lower frequencies is purely reactive. The resistance value is nearly constant until it approaches the cut-off frequency when it begins to increase. The simple exponential horn having the same expansion coefficient has the same cut-off frequency but at the higher frequencies, that is in its working range, it is not free from reactance. In systems of the type described the presence of reactance in the load impedance is prejudicial to the realization of the desired response characteristic, but by the use of the special horn contour this is eliminated and the improved response is obtained.

The horn need not be constructed with a straight axis and with a circular section as shown in the figure, but may be formed in accordance with the common practice of "folding" to diminish the space required and the cross section may be of any shape so long as the area varies in the manner defined by Equations 13 and 14.

While the invention has been described with particular reference to a stylus driven phonograph reproducer its application is not limited thereto, but may include acoustic recording devices and also electrical telephone receivers and transmitters, for example of the type shown in my earlier U. S. Patent No. 1,681,554, issued August 21, 1928. In the latter the vibrating system is driven by a force applied to a magnetic armature, which constitutes a line mass, and is transmitted to the diaphragm through one or more intermediate elements. If the elements are proportioned in the manner described above the system will have the property that the output velocity in response to a constant force input varies with frequency in the manner that is characteristic of the invention.

What is claimed is:

1. A sound reproducer comprising a diaphragm, a horn, an air chamber between said diaphragm and said horn, and means for vibrating said diaphragm in accordance with sound vibrations, said diaphragm, air chamber, and vibrating means constituting a serially coupled system of masses and elasticities in which the effective values of the masses increase progressively in the direction away from the horn and the effective values of the elasticities diminish substantially as described whereby the sound energy delivered

to the horn for a constant input is substantially proportional to

$$\frac{1}{1+\left(\frac{f}{f_o}\right)^{2n}}$$

where f is the vibration frequency, f_0 a preassigned frequency defining the upper limit of uniform response, and n is the number of elements in the coupled system.

2. A sound reproducer in accordance with claim 1 in which the effective values of the masses and of the reciprocals of the elasticities vary progressively in the direction away from the horn in proportion to the the coefficients

$$a_1 = \sin\frac{\pi}{2n}$$

$$a_2 = \frac{\sin\frac{3\pi}{2n}}{\cos^2 \pi/2n}$$

$$a_r = \frac{\sin\frac{2r-1}{2n}\pi \sin\frac{2r-3}{2n}\pi}{a_{r-1}\cos^2\frac{r-1}{2n}\pi}$$

$$a_n = n \sin\frac{\pi}{2n}$$

where the subscripts 1, 2, r, n, denote the order of the elements of the vibrating system counting from the horn.

3. A sound reproducer in accordance with claim 1 in which the air chamber is coupled to the horn by an annular passage the mean diameter of the annulus being substantially 0.63 of the free diameter of the diaphragm whereby the mass effect of the air in the air chamber is substantially eliminated.

4. In a sound reproducer a diaphragm, an air chamber enclosing said diaphragm and a horn the small end of which opens into said air chamber, the opening from said horn into said air chamber being in the form of an annulus having a mean diameter substantially equal to 0.63 of the free diameter of said diaphragm.

5. A sound reproducer in accordance with claim 1 characterized in this that the cross sectional area of the horn increases progressively with the distance along the axis from the small end in accordance with the formula

$$A = A_0 \cosh{}^2 x$$

where A_0 is the throat area, and A is the area at the distance x along the axis from the throat, whereby the acoustic load provided by the horn is free from reactance throughout the useful transmission range.

6. An acoustic horn the cross sectional area of which varies with the distance along the axis from the small end in accordance with the formula

$$A = A_0 \cosh{}^2 x$$

where A is the area at the distance x from the small end and A_0 is the area of the small end opening.

In witness whereof, I hereunto subscribe by name this 27th day of May, 1929.

EDWARD L. NORTON.