The Dynamic Vibration Absorber Principle Applied to a High-Quality Phonograph Pickup

ALLEN R. GROH

Shure Brothers Incorporated, Evanston, Ill.
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Recent developments in phonograph disc recording have placed increased demands on the playback transducer. Both dynamic range and bandwidth have been extended, particularly on discrete four-channel records. The dynamic vibration absorber principle is studied as a means of extending transducer bandwidth without compromising other parameters. An application of the vibration absorber is developed for use in a top quality magnetic piono pickup that provides high performance through almost five decades of frequency.

INTRODUCTION: It is a well-known principle in engineering that for a given system, whether electrical or mechanical, improvements in performance usually are achieved by compromising other factors. Amplifier bandwidth, for example, may be increased by trading off gain; and similarly, in a phonograph transducer, bandwidth may be increased by either limiting the dynamic range or increasing the complexity of the mechanical structure. In many cases the tradeoffs required to obtain an improvement in performance can outweigh the gains.

The most straightforward theoretical approach for increasing the bandwidth or dynamic range in a phonograph pickup is to reduce the effective mass of the moving system. A practical limit exists, however, as the stylus becomes too fragile or the output signal becomes too low to be useful. Other alternatives such as decreasing the compliance of the bearing element or stiffening the stylus can lead to poor compromises in terms of wideband performance. Through the many development efforts in phonograph reproduction in recent years, we have learned that the playback process must be observed as a complete system that includes the tone arm, the playback transducer, and the record itself. This means that signals other than the audio frequencies must be considered, including record warps that extend approximately five octaves below the audio band and may include the discrete four-channel carrier frequencies up to 45 kHz.

For the purpose of this study, a more complex playback transducer assembly is considered that is optimized to accommodate the wide range of signals required to effectively reproduce discrete four-channel recordings without compromising stereo performance. The new assembly employs a variation of a damped dynamic vibration absorber, which is coupled to the moving system to provide controlled and selective damping. Through proper design, the dynamic absorber can be optimized to be a useful control element for a magnetic phonograph pickup.

THEORY OF OPERATION

In its simplest form, the dynamic vibration absorber or inertial damper consists of an auxiliary mass coupled to the main vibrating mass by a spring and damping element [1, pp.112-131] (Fig. 1). The system is described by the differential equations

\[ M_1 \ddot{x}_1 + R(\dot{x}_1 - \dot{x}_2) + K_1 x_1 + K_2 (x_1 - x_2) = P_0 \sin \omega t \]  
\[ M_2 \ddot{x}_2 + R(\dot{x}_2 - \dot{x}_1) + K_2 (x_2 - x_1) = 0 \]

where

\[ M \] mass of vibrating system
\[ M_2 \] mass of dynamic absorber
\[ R \] damping in absorber system
\[ K_1 \] stiffness of vibrating system
\[ K_2 \] stiffness of absorber system
\[ X_1 \] displacement of main mass
\[ X_2 \] displacement of absorber mass.

At frequencies well below the resonance of the auxiliary damper mass and spring, the masses move in phase and no damping occurs. Above its resonance frequency, the damper mass provides an apparent fixed point in space [2] which results in a force on the damping element that is transmitted to the main system. When the inertial damper is optimally tuned to resonate just below the resonance frequency of the main system, the damper mass will move out of phase with the main mass at resonance, thereby amplifying the damping force. Fig. 2 shows the amplitude response [from Eqs. (1) and (2) with \( P_0 = \) constant] for various amounts of damping. The absorber tuning is defined by the expression [1, p. 126]

\[ f_a = \frac{1}{1 + \mu} f_s \] (3)

and

\[ \mu = \frac{M_2}{M_1} \] (4)

where

\[ f_a \] resonance frequency of absorber, \( - (1/2\pi) (K_2/M_2)^{1/2} \)

\[ f_s \] resonance frequency of vibrating system, \( = (1/2\pi) (K_1/M_1)^{1/2} \).

This tuning provides for equal amplitudes at two frequencies.

The plot shows that for \( R = \infty \) the auxiliary mass \( M_2 \) is locked to the main system, resulting in a resonance \( \omega_r \) of infinite \( Q \) below the natural resonance of the main system \( \omega_0 \). The resultant resonance frequency \( \omega_r \) may be described as:

\[ \omega_r = \left( \frac{K_1}{M_1 + M_2} \right)^{1/2} \] (5)

When no damping is present \( (R = 0) \), two peaks (with an infinite \( Q \)) appear above and below \( \omega_r \). Vibration cancellation occurs only at the resonance frequency of \( M_2 \) and \( K_2 \) where the relative displacements of the two masses (Fig. 1) are [1, p. 114]

\[ X_1 = 0 \] (6)

and

\[ X_2 = - \frac{P_0 \sin \omega t}{K_2} \] (7)

When a dynamic vibration absorber without a damping element is used to control system resonance, proper tuning for \( M_2 \) and \( K_2 \) would be \( \omega_r = \omega_0 \) so that maximum cancellation occurs at resonance. This special case of the inertial damper would normally be used only on a system that operates at a fixed frequency since the original resonance is merely replaced by two other resonances.

Between the zero damping and infinite damping conditions, a second special case exists where damping is optimized to provide the flattest frequency response shown by the curve \( R = R_{opt} \) in Fig. 2.

While the second special case offers maximum cancellation of vibration at all frequencies, another useful application of the inertial damper can be shown where tuning is other than optimum as described. When the mass, compliance, and damping elements are varied, the inertial damper may be used to control the shape of the frequency response and optimize the mechanical impedance of the system near resonance. Fig. 3 shows the frequency response and mechanical impedance for the simple system in Fig. 1, both with and without the inertial damper. Note that usable bandwidth can be effectively increased and mechanical impedance controlled through a desired passband. This characteristic of the inertial damper may be used to good advantage in a phonograph pickup where it is desirable to control both mechanical impedance and frequency response.
OPTIMIZING THE INERTIAL DAMPER FOR A PHONOGRAPH PICKUP

The objective in designing a phonograph cartridge is to optimize the important parameters so that in conjunction with a properly designed tone arm all the wanted and unwanted signals found on recordings may be properly handled. For this analysis it is useful to divide the applicable frequency spectrum into three parts.

1) Subsonic or Warp Frequencies (0.5 Hz to 20 Hz)

In this region the ability of the stylus to remain in contact with the record groove is largely controlled by arm resonance [3]. The parameters affecting this resonance are the effective mass of the tone arm—cartridge combination and the compliance (plus some damping) of the styli-bearing element. The effective mass of available tone arms is generally beyond the control of the phonograph pickup manufacturer, and mass values in the range of 13 to 30 grams (including cartridge) are not uncommon.

A stylus compliance figure should be chosen such that the tone arm—cartridge resonance will occur between 7 and 15 Hz [3, p. 15].

2) Audio Frequencies (20 Hz to 20 kHz)

Performance requirements for the audio band are well known [4]. Playback frequency response should be flat, and tracking capability must be high enough to accommodate recorded signals. For analysis of the mechanical requirements of the pickup, the audio spectrum may be divided into three parts: the amplitude-limited region (20 Hz to approximately 1000 Hz), the velocity-limited region (approximately 1000 to 4000 Hz), and the acceleration-limited region (above 4000 Hz). In the amplitude-limited region, stiffness of the bearing element limits trackability; therefore the bearing should be highly compliant. In the velocity-limited region, damping should be low; and in the acceleration-limited region (near resonance), a large amount of damping is required to reduce the mechanical impedance. Additional bearing stiffness may also be desirable to raise the stylus resonance above 20 kHz.

3) The Discrete Four-Channel Carrier Band (20 kHz to 45 kHz)

Frequency response in the carrier band (although not of primary importance) should be free from extreme peaks and dips to ensure minimal amplitude modulation of the carrier signal. Other important factors include reasonably linear phase response and enough output to provide a good signal-to-noise ratio for the decoder. Trackability must be sufficient to recover the 5-cm/s carrier in the presence of high-velocity baseband material.

A better picture of the interaction of these parameters may be shown by a simple electrical analog of the phonograph stylus [5] (Fig. 4). Each of the components in the analog circuit relates to a physical parameter of a moving-magnet phonograph stylus as shown in Table I. Record-tip compliance $C_1$ is considered a constant and was determined experimentally. The effective mass and compliance values of the stylus ($L_1$, $L_2$, and $C_3$) are limited by the materials available and represent an extremely low-mass state-of-the-art shank, magnet, and diamond tip. $R$ and $C_3$ are the damping and compliance characteristics of the bearing. The bearing material is a high-damping elastomer, selected for its low-frequency compliance and its ability to control mechanical impedance at resonance. For the purpose of the analog, both $C_3$ and $R$ are defined as a function of frequency representing the dynamic modulus of the elastomer[6]:

$$C_3 = F_d = (K^* f^b)^{-1}$$

and

$$R = G_d = R^* f^b$$

where

$K^*$ stiffnes of elastomer at 10 Hz
$R^*$ Damping of elastomer at 10 Hz.

Both $\alpha$ and $\beta$ were determined empirically.

Dynamic compliance of the bearing ($C_3$) is $18 \times 10^{-6}$ cm/dyn at 10 Hz, which corresponds to a favorable value in terms of compatibility with a wide range of tone arms. The 10-Hz resistance value ($R = 300 \Omega$) is that of a high-damping elastomer.

Solving the analog circuit with a digital computer yields the frequency-response and trackability curves pictured in Fig. 5 (response and trackability of an actual stylus are also shown). The resultant frequency response shows a sharp peak and dip in the carrier band and is less than ideal for best performance. The trackability curve also exhibits a corresponding fluctuation. The response may be contoured for better audio-band performance by shaping the electrical resistance of the coil and pole-piece structure, but only at the expense of output in the CD-4 carrier band. Similarly, the mechanical impedance in the CD-4 carrier band may be improved by sacrificing low-frequency trackability.

It is clear that, with the limitations on the stylus mechanical system imposed by the use of only one damping element (the bearing), a second means of obtaining high-frequency damping to control mechanical impedance would be advantageous. One device available to provide this impedance control is the inertial damper (or damped dynamic vibration absorber) which, in addition, can be used to shape the frequency response in the discrete four-channel carrier band.

In selecting damper parameters, care must be taken to avoid adding an amount of mass that significantly reduces the tracking ability at frequencies below the resonance of the absorber. It is not necessary for the damper mass to be as great as that of the stylus [$\mu = 1$; see Eq. (4)] to be effective. The value selected for the mass of the damper in the stylus electrical analog shown in Fig. 6 represents one
half the effective mass of the magnet end of the stylus (that is, $\mu = 1/2$). This value was found to provide adequate impedance and response control without adversely affecting lower frequency trackability. The frequency response and trackability curves are shown in Fig. 7. Both curves exhibit much more control near resonance than is shown by the stylus analog without an inertial damper (Fig. 5). Trackability below 15 kHz is affected only slightly by the added small damper mass; but near the stylus resonance (15–30 kHz) there is a significant reduction in mechanical impedance and therefore an increase in trackability. Frequency response could be further flattened by increasing the damper mass, but only at the expense of trackability below 15 kHz.

At this point it should be noted that while the electrical analog provides useful generalizations on stylus performance, the fact that it is a lumped-parameter analog implies some limitation on accuracy. Some of the concepts, however, may be extracted from the simple situation without involving a highly complex distributed-parameter analog. Large values of damping are easily achieved in the analog circuit, but physical realization of high damping becomes more difficult. The damping factor $(CR^2/M)^{1/2}$ is proportional to the square of resistance, and inversely proportional to the square root of the mass, indicating that the largest damping factor can be obtained when the mass of the damping material itself is used to provide the resonating mass. Thus in actual practice a distributed system should provide a higher damping factor than a lumped design.

CONSTRUCTION OF STYLUS WITH AN INERTIAL DAMPER

The first example of a practical design will approximate the lumped-parameter analog. For simplicity sake the damper mass is attached to the rear of the magnet of a moving-magnet transducer element (Fig. 8). The spring and dash-pot parameters are provided by a block of high damping elastomer with the proper spring constant. The

<table>
<thead>
<tr>
<th>Circuit Symbol</th>
<th>Dimension</th>
<th>Description</th>
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</thead>
<tbody>
<tr>
<td>$C_1$</td>
<td>Microfarads</td>
<td>Millimeters$^2$ per dyne</td>
</tr>
<tr>
<td>$C_2$</td>
<td>Microfarads</td>
<td>Millimeters$^2$ per dyne</td>
</tr>
<tr>
<td>$C_3$</td>
<td>Microfarads</td>
<td>Millimeters$^2$ per dyne</td>
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<td>Milligram</td>
</tr>
<tr>
<td>$L_2$</td>
<td>Millihenrys</td>
<td>Dyne $\cdot$ seconds per centimeter</td>
</tr>
<tr>
<td>$R$</td>
<td>Ohms</td>
<td>Centimeters/second</td>
</tr>
<tr>
<td>$I$</td>
<td>Amperes</td>
<td>Centimeters/second</td>
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</tr>
<tr>
<td>$V_{in}$</td>
<td>Volts</td>
<td>Tracking force</td>
</tr>
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Table I. Circuit elements from Figure 4.

![Fig. 5. Comparison of analog (solid line) and actual stylus (dashed line).](image1)

![Fig. 6. Stylus analog with dynamic absorber.](image2)

![Fig. 7. Performance of stylus analog with dynamic absorber.](image3)

![Fig. 8. Lumped-parameter stylus construction.](image4)
parameters of the other elements of the moving system closely approximate the values selected for the computer analog. The resultant frequency response and trackability curves, pictured in Fig. 9, show that proper tuning has been achieved; but there is insufficient damping to provide a smooth curve in the resonance region.

Increased damping may be achieved either by adding material to the bearing element or by somehow increasing the damping in the dynamic absorber coupling. From the trackability curve it is apparent that more damping in the bearing element is not desirable since it reduces trackability in the velocity-limited region. The alternative is to construct a distributed-parameter inertial damper to attempt to increase the ratio of the damping per unit mass. This is accomplished by using a rod of special elastomer to provide the necessary mass, compliance, and damping attached to the rear of the magnet (Fig. 10). The resonance frequency of the elastomer rod may be expressed by the equation for a cantilever beam [1, p. 459]

$$\omega_a = \frac{b_n}{\rho l^2} \left( \frac{EI}{\rho l^4} \right)^{1/2}$$

(11)

where

- $\omega_a$: natural frequency of the absorber
- $b_n$: numerical constant for the first mode of vibration
- $EI$: dynamic bending stiffness of the cross section
- $\rho$: mass per unit length of the elastomer
- $l$: overall length of the rod.

The distributed-parameter inertial damper results in the response and trackability curves shown in Fig. 11. Here it can be seen that even more overall damping has been achieved than was predicted by the lumped parameter analog. In addition to smoother frequency response the trackability curve shows a significant increase in the discrete four-channel carrier band. This is partly because compliance and damping both change with frequency [Eq. (6)], thereby skewing the mechanical resonance. The result is a moving system that provides high performance over a wide frequency range.

**CONCLUSION**

Many of the developments in phonograph transducers are, at least in part, a result of improvements in materials or processes. New high-energy magnetic materials and the development of extremely thin walled tubing are two examples of advances that have resulted in better pickup performance. There are many cases, however, where new materials or processes alone are not enough to provide a sufficient improvement in performance. In the preceding study the objective was to optimize a phonograph pickup to accommodate an extended range of frequencies. Here the degree of improvement desired necessitated a more complex assembly in the form of the dynamic vibration absorber. The absorber offers the unique advantage of providing selective damping only where it is needed. This is a useful property for a phonograph pickup considering the wide range of frequencies that must be accommodated. Since each division of the frequency spectrum optimally requires a different amount of damping, mass, and compliance, the ability to selectively control any one parameter offers a distinct advantage. In the structure described, careful optimization of all important parameters produced a pickup that indeed meets the objective.

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**REFERENCES**


THE AUTHOR

Allen R. Groh was born in Spokane, Washington, in 1947. He joined Shure Electronics of Arizona in 1970 after serving three years in the United States Army. After receiving his Bachelor of Science Degree in electronics from Arizona State University in 1974, Mr. Groh joined the engineering division of Shure Brothers Incorporated in Illinois. He is presently a senior development engineer in the electromechanical development department and is responsible for the design of state-of-the-art phonograph transducers. He is also presently active in graduate course work towards an MBA at the University of Chicago. An author of technical papers, Mr. Groh is a member of the AES and an officer of the Chicago Acoustical and Audio Group.