The Rational Design of Phonograph Pickups *†

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Design problems associated with signal transduction and with proper groove tracking can usefully be considered separately. This paper deals with the latter and presents a rational basis for specifying all the important mechanical parameters of a pickup and its tone arm. The only available driving force is that derived from the bearing weight and this is arbitrarily allocated equally among the three independent and uncorrelated driving-force demands imposed by tone-arm acceleration, stylus acceleration, and low-frequency tracking. These demands can be assessed by reference to standard record dimensions and the limiting characteristics of typical recorded material. Design inequalities can then be deduced which specify limiting values for the equivalent tone-arm mass, the stylus compliances, and the equivalent stylus mass in terms of the bearing weight. In turn, these parameters locate the frequencies of tone-arm resonance and of free resonance of the stylus system. The location of the stylus-groove resonances and of the cutoff frequencies for translation loss and scanning loss involve the foregoing parameters and also the stylus radius, the elastic modulus of the record material, and the groove velocity. The effect on pickup performance of different placements of these critical frequencies is discussed, and attention is directed toward the possibility of achieving superior performance by arranging these frequencies in a certain pattern. The discussion concludes with a critical comparison of these design criteria with the specifications of commercially available pickups.

INTRODUCTION

THIS is the first part of a serial report dealing with the design and construction of ultra-lightweight stereo phonograph pickups. The initial objective was to reduce the bearing weight required for satisfactory tracking by at least an order of magnitude. It soon became apparent that such a major change of scale could not be achieved merely as a simple modification or extrapolation of current designs. In order to establish guide lines for a new approach, therefore, the pickup problem has been stripped down to its bare essentials and the basic rationale of design has been re-examined critically.

An adequate review of the fundamentals of design must necessarily embrace both the pickup and the tone arm. Although the present study was primarily oriented toward the achievement of low bearing weight, it has also yielded a coherent set of design criteria by which the functional adequacy of any disk reproducer system can be assessed.

"Optical tracking" is almost always suggested whenever the ultimate in low bearing weight is under discussion, so perhaps this possibility ought to be dealt with at the outset. With modern techniques of servo control it would probably be technically feasible to devise an optical system, using light reflected from the groove walls, which could lock on and track the conventional spiral record groove. The error signal in such a control loop might then be extracted as a transduction of the groove modulation. The necessary degree of vibratory noise suppression in such a tracking system, however, would seriously strain the present state of the servo art, and it is almost certain that a playback system of this type would be neither simple nor compact. For these reasons, optical tracking was ruled out-of-bounds in the present investigation.

Mounting an orderly and systematic attack on the pickup design problem calls for recognizing at the outset that there is a distinct functional separation between the roles of *tracking* and of signal *transduction*. Perfect tracking is a necessary, but not a sufficient, criterion of the adequacy of a pickup, whereas imperfect tracking, even with ideal transduction, is a certain guarantee of inadequacy. In pursuance of this separation of function, the following exposition deals with the mechanical features of design that control proper tracking and with the rational basis for selection of a self-consistent set of mechanical pickup parameters. The role of transduction will be discussed in a subsequent paper.

It seems to be a common practice of pickup designers to select first the mode of transduction, then to choose the kind and size of associated vibratory structure on the basis of convenience or necessity, and finally to determine by experiment what bearing weight is required for satisfactory tracking. The present study suggests that the converse procedure is to be preferred and, therefore, that design should start with the specification of the desired bearing weight. This specification, in conjunction with a prudent interpretation of the limiting characteristics of the recorded material to be reproduced, is sufficient to determine the required values of all the other mechanical parameters of the pickup. The discussion of these mechanical parameters concludes with a speculative consideration of the possibility of achieving su-

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perior performance through a particular arrangement of the upper critical frequencies.

THE FRAMEWORK OF ANALYSIS

The pickup bearing weight M_B (grams) designates an equivalent unbalanced mass, located at the stylus end of the tone arm, on which the action of gravity would produce a downward vertical "stylus force" of $M_B g$ dynes. This stylus force is transmitted through an elastic stylus support to the stylus tip where it is just balanced, in both static and dynamic equilibrium, by the vertical components of the normal force reactions at each of the two groove walls. The transmission of this force through the vertical compliance of the stylus support serves to preload, or bias, this elastic element. The restoring force thereby made available must then be sufficient to impart to the stylus any acceleration which may be needed in order to keep it always in contact with both groove walls. From another point of view it can be said that all motions of the reproducer system, including the uniform traverse of the tone arm across the record as well as the high-frequency vibrations of the stylus, are to be regarded as driven by the groove-wall reaction forces. In this case the elastic element which is preloaded is the groove wall, and the normal reaction at the wall merely describes the restoring force aroused by the elastic (or plastic!) deformation of the groove wall under the stylus.

The net reaction force at each wall is the difference between its mean or static value and the inertial reaction of the equivalent mass of the stylus (at high frequencies) or of the tone arm (at very low frequencies). The inertial reaction components of these forces must have a zero time average as a consequence of the fact that neither the stylus nor the tone arm can have a time-independent residual or net acceleration. Moreover, the total reaction force can never reverse algebraic sign at either groove wall since there can be no adhesion between the stylus and the groove and since neither wall deformation can disappear entirely without violating the basic requirement that the stylus remain in continuous contact with both walls. It follows, therefore, that the peak value of the net dynamic force available to drive either the stylus or the tone arm is just that deriving from the bearing weight in static equilibrium.

It is useful to identify for subsequent discussion five different regions of the frequency spectrum, in each of which a different mechanical parameter plays the dominant role in controlling performance. These are:

A. *The very-low-frequency region below the frequencies of tone-arm resonance.*—The tracking behavior of the tone arm is of primary interest in this frequency range, and the compliance and mass of the stylus system are of no concern. The available component of driving force must satisfy the requirements imposed by the equivalent inertial mass and the steady-state damping resistance of the tone arm as it follows the mean groove trajectory.

B. *The region of the horizontal and vertical tone-arm resonances.*—These resonances occur at the frequencies determined by the corresponding stylus compliances and equivalent tone-arm masses. The enhancement of response at these critical frequencies is controlled by the effective tone-arm damping associated with each degree of freedom.

C. *The region extending from the tone-arm resonance frequencies to the frequency of free resonance of the stylus suspension system.*—This region constitutes the lower portion of what is usually referred to as "the frequency range of (playback) interest." Throughout this region, the stylus system behaves as a simple vibrator under stiffness control and the variational component of the stylus reaction force is in phase with the groove displacement. As the stylus follows large-amplitude low-frequency signals, its downward excursions will, in effect, relieve part of the static preloading of the vertical compliance. The portion of the static preload so removed by downward groove modulation is not available, therefore, to meet other concurrent demands on the stylus driving force.

D. *The region extending from the frequency of free resonance to the upper limit of the "frequency range of interest."*—Throughout this region the controlling pickup parameter is the effective inertial mass (referred to the stylus tip) of the stylus and all its mechanical appendages. A firm upper limit for the allowable value of this equivalent mass can be established by considering the maximum acceleration demand imposed by the recorded material.

E. *The region containing the upper critical frequencies.*—These are the frequencies which characterize the translation loss, the scanning loss, and the natural modes of vibration of the composite system comprising the stylus suspension and groove-wall compliance. For reasons that will be made obvious by the following discussion, each of these critical frequencies should be placed well above the "frequency range of interest," and it may be particularly advantageous to arrange them in a certain pattern.

The foregoing will also serve to identify three requirements which must be met by the available driving force: one imposed by the acceleration and viscous-drag demands of the tone arm in the tracking of normal or warped and eccentric grooves (Frequency Region A), one imposed by the acceleration demand of the vibratory mass of the stylus (Region D), and one which is characterized by the partial removal of the preloading of the stylus compliance during large downward excursions (Region C). Since each of these demands reflects an independent characteristic of the recorded material, the pickup designer is forced to presume that they must all be satisfied at the same time. It follows, therefore, that some arbitrary allocation of the available driving force derived from the bearing weight must be made among these simultaneous demands. It is hard to forego an inclination to regard the role of tone-arm drive as secondary to the primary function of the pickup as a transducer of the signal modulation in the record groove. Nevertheless, in the absence of a logical argument to support this inclination, it seems reasonable to assign equal weights—or rather equal fractions of the total bearing weight—to the three roles of tone-arm drive, high-frequency groove tracing, and protection against removal of preloading.

SPECIFICATION OF THE MECHANICAL PARAMETERS

The design relations based on the suggested allocation of the available driving force can be expressed as three inequalities:

$$\tfrac{1}{3} M_B g \geqq x_s/C_{LV}, \qquad (1)$$
$$\tfrac{1}{3} M_B g \geqq M_{ta}\, a_{ta} + R_d\, v_{ta}, \qquad (2)$$
$$\tfrac{1}{3} M_B g \geqq M_s\, a_s, \qquad (3)$$

where the symbols have the following meaning: g = acceleration due to gravity = 980 cm/sec^2; M_B = total bearing weight, gram; M_{ta} = equivalent mass of the tone arm referred to the position of the stylus tip, gram; M_s = equivalent mass of the stylus and its elastic support, referred to the stylus tip, gram; C_{LV} = compliance of the stylus suspension, assumed to be the same for lateral or vertical motion, cm/dyne; x_s = allowance for removal of preload = maximum vertical displacement amplitude of groove modulation, cm; a_{ta} = maximum acceleration of the tone arm, measured at the position of the stylus tip, arising during tracking of warped or eccentric grooves, cm/sec^2; v_{ta} = maximum radial velocity of the tone arm, measured at the position of the stylus tip, cm/sec; R_d = tone-arm damping resistance, referred to the position of the stylus tip, dyne-sec/cm; and a_s = maximum acceleration of the stylus tip arising in the tracing of the modulated groove, cm/sec^2.

These design relations contain four arbitrary parameters which are determined by the characteristics of the recording to be reproduced and which are beyond the control of the pickup designer, viz., x_s, a_s, v_{ta}, and a_{ta}. Two of these, x_s and a_s, are vulnerable to the apparently irresistible urge of recording engineers to modulate the record groove as fully as its dimensions will allow. The other two, a_{ta} and v_{ta}, are dictated by the standard dimensional specifications of the records and by contingency demands imposed by damaged records or unusual circumstances. However, after "reasonable" values of these record characteristics have been assigned, the design inequalities (1), (2), and (3) can be used to establish limiting values of the primary mechanical parameters of the stylus system and the tone arm in terms of any prescribed value of the total bearing weight.

Stylus Compliance

Although the vertical compliance of the stylus support is statically preloaded by the bearing force, whereas the lateral compliance is not, the dynamical requirements imposed by the recorded material are symmetrical. Thus the stylus must track either lateral, vertical, 45°-right, or 45°-left modulation with equal facility. This strongly suggests what has already been implied in defining C_{LV}, namely, that the variational compliance of the stylus suspension should be the same for lateral or vertical motion.

An important geometrical feature pertaining both to pickups and recording cutters must now be considered. For either pickup or cutter, a vibrational axis can be defined as a line which passes through the tip of the stylus when it is in playing (or cutting) position on the record and which is normal to the plane of action containing the so-called vertical and lateral excursions of the stylus tip. For structural reasons it is not ordinarily convenient for either of these axes to be parallel with the record surface, and angles of inclination ranging from 10° to 25° have been used. However, the pickup axis should obviously coincide in direction with the cutter axis since any angular divergence between them will represent an angular tracking error which can be a source of appreciable harmonic distortion. Since this type of angular tracking error can be completely eliminated by standardization, it can be hoped that some (any!) value for the cutter inclination angle may soon gain universal acceptance. In the meantime, such standardization can be anticipated by adopting tentatively a compromise value of 15° for the inclination angle of the pickup axis.

In order to use the design inequality (1) to specify the stylus compliance in terms of the bearing weight, it is necessary to assign an appropriate range of values for x_s, the vertical amplitude allowance for removal of stylus preload. This assignment can be made within relatively narrow tolerance limits on the basis of current standards governing the size and shape of record grooves. Two numerical data are relevant: 1. The maximum width of an unmodulated groove is 3.2 mil, according to the Electronic Industries Association (EIA) Standard RS-211-A (August 1959); 2. The width of a modulated groove is required to be never less than 1.0 mil in order to avoid expelling the stylus from the groove at the extreme upward excursions of full vertical modulation. These two limits define the peak amplitude of allowable width variation during vertical modulation as 2.2 mil, corresponding to a variation of total groove width at the record surface between the limits described by 3.2 ± 2.2 mil. The depth of a groove is always just half its width when the included angle of the cutting stylus is 90°, so the corresponding limits on the depth variation for a fully modulated groove are 1.6 ± 1.1 mil.

Each of the two groove walls must be assumed to carry independent modulation in a stereo recording, so the net groove modulation may at some times be all vertical and at other times all lateral. The manner in which these extremes occur can be explained by assigning half the groove width variation described above to the lateral deviations of each groove wall boundary. Then, when the lateral deviations of the inner and outer wall of the groove occur in phase, the groove width remains constant and the modulation is all lateral with a peak amplitude of 1.1 mil. On the other hand, when the lateral deviations occur in opposite phase, the groove centerline is undeviated and the modulation is all vertical with a peak amplitude of 1.1 mil, as described above.

If half the allowable variation of groove width is assigned to the lateral deviation of each groove wall, and if the modulation of adjacent grooves is uncorrelated, then the required spacing between the centerlines of adjacent grooves will be just one "maximum groove width" plus some small allowance, such as 0.1 mil, for the minimum land needed between adjacent groove edges. It follows, therefore, that the separation between adjacent fully modulated grooves must be at least 5.5 mil. This is equivalent to specifying that the groove pitch should not be finer than the reciprocal of

0.0055, or 180 lines per inch. On the other hand, if the recording engineer chooses to deal with a loud passage by momentarily opening up the groove pitch to 100 lines/inch, the mean groove depth should be correspondingly increased from 1.6 to 2.75 mil. The maximum vertical excursion from the mean depth might then be as large as 2.25 mil. No information is available as to how frequently this strategem is used in current recording practice, but prudence dictates that the pickup designer should make some provision for such a contingency.

The foregoing considerations suggest that 1.9 ± 0.2 mil defines a reasonable range of values for x_s, the vertical amplitude allowance for removal of preload. On this basis, the design inequality (1) specifies that the product $M_B C_{LV}$ should be at least $(14.7 \pm 1.6) \times 10^{-6}$, with M_B in grams and C_{LV} in cm/dyne. If the compliance were expressed in mils per gram weight, a highly mixed but sometimes useful unit, the corresponding limits on the bearing weight-compliance product would be 5.7 ± 0.6. In either form, these statements are equivalent to the specification that the compliance of the stylus suspension should be such that the total bearing weight will produce a static deflection at the stylus of at least 5.7 ± 0.6 mil.

Tone-Arm Mass

In order to use the design inequality (2) to specify the equivalent mass of the tone arm in terms of the bearing weight, it is necessary to assign appropriate maximum values for the expected velocity and acceleration of the tone arm. These assignments are largely controlled by the secondary dimensional characteristics of typical records and by a restrained desire to provide for tracking any record, damaged or otherwise, under almost any circumstances. A few quantitative examples will illustrate the interplay of these factors.

Tracking an eccentric groove gives rise to a horizontal simple harmonic motion characterized by a radial acceleration equal to half the peak-to-peak runout multiplied by the square of the angular frequency of disk rotation. The maximum runout for the eccentric stopping groove of a 78-rpm record is 0.265 in., according to EIA Standard RS-211-A. Tracking such a groove, therefore, requires a radial acceleration of the tone arm of 0.023 g.

The transition from recording pitch to the pitch of the lead-out spiral on 45 and 33⅓-rpm records calls for momentary radial accelerations that are almost as demanding. The coarsest allowable pitch of the leadout spiral for these records corresponds to a radial velocity of 0.95 cm/sec at 45 rpm and 0.705 cm/sec at 33⅓ rpm. If the transition to these coarse pitches were to be completed within 35 to 50 milliseconds (i.e., within 10 to 15° of record rotation), the required radial accelerations would be 0.026 g and 0.015 g. Such accelerations are of relatively short duration, but they last long enough to induce groove jumping unless the inertial mass of the tone arm is small enough to allow it to be correspondingly accelerated by the available driving force.

A more severe requirement would be imposed by the rather awesome "45-big-hole test," in which a 45-rpm record is displaced on the turntable so that one edge of the large center hole is in contact with the turntable spindle. At the normal speed of 45 rpm this calls for a radial acceleration as high as 0.035 g, but this relaxes to 0.019 g if the test is compromised by running it at 33⅓ rpm. This bizarre test is obviously unreasonable as well as unrealistic, but it is of interest to observe that its acceleration demand only exceeds by less than a factor of two the more temperate demands of conventional records in normal playback.

It is relatively more difficult to make quantitative estimates of the probable acceleration demand arising in the tracking of vertical bumps produced by warpage. However, as an example which is severe but not outlandish, consider a bump which rises in sinusoidal fashion to an elevation of 1/16 in. and then subsides within a total wavelength occupying ⅕ of the record circumference. Even at 33⅓ rpm this would call for a peak acceleration of 0.025 g, and the demand would be correspondingly higher if the same bump were negotiated at higher record speeds.

In view of the foregoing, it is suggested that $a_{ta} = 0.025\ g$ represents a reasonable compromise value of the tone-arm acceleration demand for use in setting an upper limit on the allowable equivalent mass of an undamped tone arm.

The second term on the right-hand side of inequality (2) accounts for the driving force requirements imposed by tone-arm damping. It is obvious that the lateral and vertical stylus compliances and the corresponding tone-arm masses will constitute a pair of high-Q resonant systems unless some kind of damping is introduced. Bachman[1] has shown definitively that such damping should be associated with the tone-arm pivots, *not* with the stylus compliances. Bachman also showed that such damping ought to be of the viscous type, with drag force proportional to velocity, rather than of the velocity-independent type ordinarily associated with bearing friction. If the damping resistance R_d is indeed of the viscous type, the Q factor for tone-arm resonance can be expressed as $Q_{ta} = M_{ta}\,\omega_{ta}/R_d$, where ω_{ta} is the angular frequency of tone-arm resonance. It can then be readily shown that $R^2 = 1 + Q_{ta}^2$, where R is the ratio of the amplitude response at resonance to the amplitude of groove modulation. These two relations can then be used to incorporate the allowable enhancement of response at resonance explicitly in the design inequality (2), as follows:

$$\tfrac{1}{3} M_B g \geqq M_{ta}\,[a_{ta} + \omega_{ta}\,v_{ta}/(R^2 - 1)^{1/2}]. \qquad (2a)$$

As suggested above, the highest value of v_{ta}, the tone-arm traverse velocity, occurs during the tracking of the lead-out spiral on 45-rpm records. At this maximum velocity, the steady drag force due to damping becomes comparable with the acceleration demand when the height of the tone-arm resonance peak is about 10 dB. The driving force requirement becomes increasingly dominated by the drag due to damping as the resonance peak is further suppressed. Thus, for example, if $v_{ta} = 0.95$ cm/sec, $a_{ta} = 0.025\ g$, and $\omega_{ta}/2\pi = 12$ c/s, it can be inferred from inequality (2a) that

[1] W. S. Bachman, *Proc. Inst. Radio Engrs.* **40**, 133-137 (1952).

the ratio of the equivalent mass of a viscous-damped tone arm to the total bearing weight should not exceed 6.8 if the height of the resonance peak is 10 dB, and that the ratio should not exceed 3.5 if the height of the peak is only 3 dB.

The foregoing examples indicate clearly the important influence of the steady-state damping on the allowable mass of the tone arm. It would obviously be advantageous if the enhanced response at tone-arm resonance could be controlled by a *tuned* vibration absorber which would not introduce a viscous drag at the very low frequencies corresponding to the tone-arm traverse motion. If such a tuned damper were employed, the term involving v_{ta} could be dropped from inequality (2a), which would then lead (with $a_{ta} = 0.025\ g$) to the specification that the ratio M_{ta}/M_B should not exceed 13. Unfortunately, this is still a moderately severe restriction on the allowable mass of the tone arm.

It might be argued that the assignment of only one-third of the available bearing force to the task of tone-arm drive could be relaxed since there is no groove modulation to protect during the tracking of the finishing grooves. Moreover, the maximum values of acceleration and of traverse velocity which appear in the inequality (2a) do not ordinarily arise concurrently. These arguments do not quite stand up, however, since vertical bumps do occur simultaneously with groove modulation and during the tracking of lead-out spirals, and such bumps have a potential acceleration demand comparable with that assumed above in the numerical example for the tone arm with tuned damper.

In summary, therefore, two conclusions can be drawn: 1. the equivalent mass of the tone arm should never exceed about 13 times the total bearing weight, and 2. the tone-arm mass should not exceed 6 times the bearing weight if the steady-state damping is comparable with the damping at tone-arm resonance.

Stylus Mass

The design inequality (3) can be used to set a firm upper limit for the allowable mass of the stylus and its suspension, provided a suitable range of values can be assigned for the stylus acceleration demand. Dimensional standardization is of no help in making this assignment. Instead, the pickup designer is at the mercy of the recording engineer who is usually, in turn, under pressure from the "front office" to modulate the recorded groove as fully as the limits of intolerable distortion will allow. The latter provides the key to the assessment of the stylus acceleration demand. The acceleration and the groove-wall curvature on which the distortion depends are each functions of the second derivative of the groove-wall displacement. It is relatively straightforward, therefore, to express the maximum acceleration demand in terms of the maximum level of tracing distortion at whatever frequency the latter may occur. In this way it is possible to avoid any necessity for detailed consideration of the shape and intensity distribution of the spectrum of recorded signals. This analysis will be presented first for the case of rigid groove walls. The further limitations on stylus mass which arise when the deformation of the groove wall is taken into account will be dealt with later in terms of the location of the upper critical frequencies.

As pointed out above, the modulation of a stereo groove may at some times be all lateral and at other times all vertical. It is obvious that in the latter case the stylus will lose contact with the groove unless the available fraction of the total driving force can impart to the stylus mass an acceleration matching that of the groove. It is less obvious but equally true that a similar intermittency of stylus-groove contact will arise unless the stylus acceleration can also meet the acceleration demands of the second-harmonic components of tracing distortion which constitute the pinch effect. The amplitude of vertical motion called for by the pinch effect at high levels of distortion will ordinarily exceed the depth of the elastic deformation of the groove wall under the stylus. As a consequence, the pinch-effect motion cannot adequately be "absorbed" in the groove-wall deformation. Of course any intermittency of stylus-groove contact is greatly to be deplored, whether it occurs at fundamental frequency or at some frequency above the system cutoff, since each restoration of contact will produce impulse excitation of the stylus system and a corresponding contribution of broad-band noise. It continues to be true, therefore, that the proper tracing of a laterally modulated groove requires that the stylus system be capable of following vertical modulation faithfully up to frequencies as high as twice the upper cutoff of the recorded material.[2]

It follows from the foregoing that the limit on stylus mass imposed by the total acceleration demand is to be reckoned in terms of the accelerations associated with the second-harmonic components of tracing distortion as well as those associated with the recorded fundamentals. The needed relations can be extracted directly from the well-known results of tracing distortion analysis.[3] Thus, for motion in the plane normal to one or the other groove wall, as specifically relevant for the playback of a single right or left stereo channel, the following relations can be tabulated:

fundamental recorded amplitude	$a \sin kVt$
recorded velocity	$kVa \cos kVt$
acceleration at fundamental frequency	$-k^2V^2a \sin kVt$
amplitude of second-harmonic component of traced trajectory	$\frac{1}{4}\, ra^2k^2 (\cos 2kVt + 1)$
second-harmonic velocity	$-\frac{1}{2}\, ra^2k^3V \sin 2kVt$
relative second-harmonic velocity distortion	$D_{2v} = \frac{1}{2}\, kakr$
acceleration at second-harmonic frequency	$-ra^2k^4V^2 \cos 2kVt$

In these relations, V is the linear velocity of the record groove (cm/sec), r is the stylus tip radius (cm), a is the

[2] Although this requirement was first pointed out more than 24 years ago [J. A. Pierce and F. V. Hunt, *J. Soc. Motion Picture Engrs.* **31**, 157 (1938) and *J. Acoust. Soc. Am.* **10**, 14-28 (1938)], it seems to have been either blandly or insistently ignored by most designers of lateral pickups.

[3] W. D. Lewis and F. V. Hunt, *J. Acoust. Soc. Am.* **12**, 348-365 (1941).

amplitude of groove modulation for the right or left channel measured along the normal to the corresponding groove wall (cm), $k = \omega/V = 2\pi/\lambda$, λ is the recorded wavelength (cm), and D_{2v} is the fractional second-harmonic (velocity) distortion. The total motion of the stylus will be the vector resultant of the contributions from each groove wall and adding these and the similar higher-order terms with due regard for phase would lead to the usual conclusions concerning the tracing distortion for lateral or vertical groove modulation. It is the acceleration of each groove wall along its own normal, however, which governs the allowable stylus mass, and this acceleration is immediately available as the sum of the fundamental and second-harmonic components of acceleration tabulated above. These components vary differently with time but once during each fundamental cycle their peak values will occur at the same time and with the same algebraic sign. The relevant peak magnitude of the required stylus acceleration a_s can, therefore, be exhibited at once in the following alternative forms:

$$a_s = k^2V^2a + rk^4V^2a^2 = (V^2/r)\ kakr\ (1 + kakr) \quad (4)$$
$$= (V^2/r)\ (2D_{2v} + 4D_{2v}^2) \quad (\text{cm/sec}^2).$$

The factor $kakr$, which appears in Eq. (4), is identified in the analysis of tracing distortion as the ratio of the radius of curvature of the stylus tip to the radius of curvature of the groove surface. The second-harmonic distortion D_{2v} reaches 25% when the radius of curvature of the groove wall becomes as small as twice the stylus radius, and a cusp occurs in the rigid-wall stylus trajectory when these curvatures become equal (at $kakr = 1$). Painful observation of the recorded levels on some "hot" records would suggest that even the value of unity for $kakr$ may sometimes be exceeded. However strongly the pickup designer may disapprove such "hot" grooves, he cannot afford to be complacent about any inability of his stylus to track them. The severity of the hot-groove tracking challenge is indicated by the fact that a stylus acceleration of nearly 3000 g must be provided in order to deal with the situation described by $kakr = 1$ ($D_{2v} = 50\%$), $r = 0.7$ mil, and $V = 20$ in./sec (outer edge of 12-in. 33⅓-rpm record). On the other hand, if the designer were inclined to be somewhat less aggressive, he might set the limit on acceleration at 1000 g, which would conform to the requirements of $V = 20$ in./sec, $kakr \doteq 0.5$, and $r = 0.7$ mil.

The foregoing can be summarized in the judgment that 1000 g is a conservative, non-extravagant estimate of the maximum acceleration demand arising in the perfect tracking of typical recorded material. The design inequality (3), with $a_s = 1000\ g$, then dictates that the effective dynamic mass of the stylus should not exceed 0.33 mg for each gram of total bearing weight. This is a necessary, but not a sufficient, upper limit on the allowable stylus mass. Other considerations to be discussed below will impose even further restrictions on the stylus mass.

A Digression on Side Thrust

It has been well known, since B. B. Bauer[4] first pointed it out in 1945, that the frictional drag on a pickup stylus sliding in a record groove will give rise to a radial component of force F_r directed toward the center of the record. The magnitude of this parasitic "side thrust" is given by $F_r = \mu M_B g \tan\theta$, where μ is the effective coefficient of friction and θ is the angle between the groove tangent and a line extending from the stylus tip to the vertical tone-arm pivot. The relevant values of $\tan\theta$ are shown by the analysis of tracking angle[4] to be $\tan\theta \doteq \theta \doteq (R/2L) + D/L$, where R is the radius of the record groove, L is the length of the tone arm from stylus to pivot, and D is the overhang. When the offset angle and the overhang are chosen in the conventional way to minimize distortion due to tracking error, D can be eliminated from the expression for θ and the radial side-thrust force at the radius R can be written as

$$F_r(R) = \mu M_B g\ (R/2L) \left[1 + \frac{2\ R_o^2/R^2}{(R_o/R_i)^2 + 2(R_o/R_i) + 2}\right] \quad (\text{dynes}).$$

This expression simplifies handily for the case of a 10-in. 33⅓-rpm record for which R_o, the radius of the outermost recorded groove is just twice R_i, the radius of the inner groove. For this case the maximum side thrust, which occurs at the outside edge, can be written explicitly as

$$F_r(R_o) = \mu M_B g\ (R_o/2L) \left[\frac{(R_o/R_i)^2 + 2(R_o/R_i) + 4}{(R_o/R_i)^2 + 2(R_o/R_i) + 2}\right] = 0.6\ \mu M_B g\ R_o/L \quad (\text{dynes}).$$

Published data on μ, the effective coefficient of friction for a stylus sliding in a V-groove, are very meager but typical values appear to lie between 0.5 and 0.25, the values tending downward as the bearing weight is reduced and upward as the level of modulation is increased. The ratio R_o/L will usually fall between 0.45 and 0.7. It follows, then, that the magnitude of the radial side thrust will range from about 0.07 $M_B g$ for long, light tone arms to 0.19 $M_B g$ for short, heavy arms. Such a radial force is by no means negligible and its direct effect on stylus tracking and its influence on the partitioning of $M_B g$ among the three design inequalities (1)—(3) need to be carefully assessed.

It is instructive to examine the vector force diagram describing the equilibrium between the vertical bearing force $M_B g$ and the normal force reactions at the two groove walls. Study of this diagram reveals that the effect of adding a radial side-thrust force F_r is to *increase* the normal force reaction at the inner groove wall by $F_r/\sqrt{2}$ (and the vertical loading by $F_r/2$) and to *decrease* the normal and vertical loading of the outer groove wall by similar amounts. The total frictional drag, which gives rise to the side thrust, is substantially unchanged by this differential shift of groove loading from one wall to the other, although the wear on both stylus and groove tends to become unsymmetrical.

It has been argued that side thrust, since it acts in the inward direction of ordinary tone-arm traverse, can serve beneficially to overcome friction in the vertical pivot bearing and thereby reduce the fraction of the driving force

[4] B. B. Bauer, *Electronics*, 110-115 (March 1945).

which must be allocated for tone-arm drive. This argument is valid but misleading since it implies that the derived restrictions on allowable tone-arm mass could be eased in this way without incurring any other penalty. It needs to be remembered, however, that the driving force requirements imposed by the stylus acceleration demand are symmetrical and that the limitation on stylus mass expressed by inequality (3) will be governed by the available force reaction at the more lightly loaded groove wall. It follows, therefore, that the action of the radial side thrust in unloading the outer groove wall will increase the severity of the restriction on stylus mass in the same proportion that it serves to ease the part of the restriction on tone-arm mass which derives from horizontal, inward velocity or acceleration demands. In addition, side thrust offers no help in meeting the symmetrical drive requirements for tone-arm motion such as those which arise in negotiating bumps or eccentric grooves or those associated with viscous damping introduced for control of the vertical tone-arm resonance. The trade-off between the restrictions on stylus mass and on tone-arm mass appears, therefore, to be less than even, and one is led to the conclusion that side thrust is an undesirable effect which ought to be eliminated or compensated.

The analytical expressions given above indicate that, for the typical case of $R_o/R_i = 2$, the variation of side thrust with groove radius is described by the simple relation $F_r(R_o)/F_r(R_i) = 4/3$. Although the variation of F_r with R is not strictly linear, the behavior of F_r between the limits R_o and R_i can be closely simulated by the restoring torque of a simple torsion spring acting around the vertical tone-arm pivot. Such a simulative spring would have a spring constant of $0.3\ \mu\ M_B g\ L$ dyne-cm/radian and an initial wind-up of $4(R_o - R_i)/L$ radians when the stylus is at R_o. The chief import of this observation is the sad conclusion that side thrust cannot be perfectly compensated by any simple counter-torsion spring unless one were available which had a negative spring constant. On the other hand, the net magnitude of the side thrust can be reduced by a factor of 6 to 8 by merely providing an essentially uniform countertorque that will just cancel F_r at the mean record radius. Various devices have been proposed from time to time for accomplishing this but most of them have lacked the elegant simplicity which earns universal adoption. Nevertheless, in what follows it will be assumed that the tone-arm designer has coped successfully with the side-thrust problem.

LOCATION OF THE CRITICAL FREQUENCIES

There are at least five critical frequencies which have a significant bearing on the over-all performance of a reproducing system consisting of tone arm, pickup, and record. These are the frequencies of tone-arm resonance, free resonance of the stylus suspension, the first compliantly restrained mode of the stylus suspension, and the two cutoffs characterizing the translation loss and the scanning loss. The first two of these can be expressed in terms of the mechanical parameters already discussed. The others call progressively for the assumption or specification of additional properties or dimensions of the pickup or the record.

Tone-Arm Resonance

The stylus compliance has already been specified to be the same for either lateral or vertical motion. The horizontal and vertical tone-arm masses will also be the same if the arm is a simple strut with coincident horizontal and vertical pivots. In this case there will be a common tone-arm resonance frequency f_{ta} given by $2\pi f_{ta} = (C_{LV} M_{ta})^{-1/2}$. This can be readily evaluated for the simple case of the tone arm with tuned damper by using the limiting equalities allowed by the design relations (1) and (2) with the term involving v_{ta} dropped from (2). When the suggested numerical values for a_{ta} and x_s are introduced, this becomes

$$f_{ta} = [1/(2\pi)]\ (a_{ta}/x_s)^{1/2} \approx 11 \text{ c/s}. \tag{5}$$

Note first that this resonance frequency is not a function of the bearing weight. Note also that inequalities (1) and (2) specify a lower limit for C_{LV} and an upper limit for M_{ta}. The numerical value yielded by Eq. (5), therefore, does not uniquely define either an upper or lower limit on f_{ta}. The designer's latitude in choosing C_{LV} is less than for the other parameters, however, since reducing the displacement allowance x_s courts tracking hazards, while increasing it invites mechanical instability and excessive deflections and strains in the stylus suspension. As a result, f_{ta} cannot safely be reduced below about 10 c/s, but it can be raised to perhaps as high as 15 to 18 c/s if the designer is successful in making the tone arm much lighter than the upper limit on mass set by the inequality (2).

Free Resonance of the Stylus Suspension

The free resonance of the stylus suspension can be defined as the fundamental mode of vibration of the stylus system when the stylus is not in contact with the record groove. It is sometimes observed by exciting the pickup electrically and looking for the maximum of stylus motion through a microscope. This resonance would be identified as a first clamped-free mode if the actual stylus suspension were replaced by an equivalent simple cantilever. The driving-point impedance at the stylus tip, and hence the dynamic groove-wall reaction force, passes through a minimum at $f_{1,cf}$, the resonance frequency for this mode. There is almost never any trace of this resonance in the electrical output of a pickup in normal use, however, since the stylus motion is positively controlled by the modulation of the groove. This free resonance frequency is chiefly useful, therefore, as a vehicle for defining the equivalent mass of the stylus as the lumped mass M_s which would resonate with the static compliance C_{LV} at the frequency $f_{1,cf}$. With the help of the limiting equalities allowed by the design relations (1) and (3), and the suggested numerical values for x_s and a_s, this frequency can be written as

$$f_{1,cf} = [1/(2\pi)]\ (M_s C_{LV})^{-1/2} = [1/(2\pi)]\ (a_s/x_s)^{1/2} \approx 2300 \text{ c/s}. \tag{6}$$

As in the preceding case, Eq. (6) does not define uniquely either an upper or lower limit on $f_{1,cf}$. Also as in the preceding case, the acceptable choices available to the designer can lower this frequency by only a few percent but can raise it by as much as a factor of two or three if substantial reductions of equivalent stylus mass can be realized.

It is often suggested that the free resonance frequency should be placed at the geometric mean of the upper and lower boundaries of the frequency range of interest. This dictum is based on the presumption that such a placement of the resonance will produce the same groove loading at either limiting frequency. The foregoing discussion has indicated, however, that the limiting values of groove loading at the two extremes of frequency are dictated by independent and unrelated considerations which lead to the proper placement of this resonance well above the geometric mean of the limits of the frequency range of interest.

The Restrained Modes of the Stylus-Groove System

In normal use, of course, the stylus is not "free" but is "restrained" by its contacts with the groove walls. It is necessary, therefore, to analyze and assess the natural modes of vibration of the composite system comprising the stylus suspension and the stylus-groove contact. This analysis can be carried out by considering separately the driving-point impedance of the stylus suspension, the finite compliance of the stylus-groove contact, and the effects of their interaction.

The driving-point impedance of the stylus suspension as observed at the stylus tip, Z_{ll}, is characterized by an alternating succession of maxima and minima. These would be the typical real-frequency poles and zeros of a multimesh reactance if there were no dissipation. The reactance of the static compliance, $1/j\omega C_{LV}$, produces the pole of Z_{ll} at zero frequency. This is followed by the first zero of Z_{ll}, which occurs at the frequency defined above as the frequency of free resonance. These are followed in turn by the pole which characterizes the first "restrained" or clamped-supported mode of vibration, the zero characterizing the second clamped-free mode, the pole characterizing the second clamped-supported mode, and so on.

The driving-point impedance of any stylus system, no matter what its configuration, can always be described in this way in terms of the location of its poles and zeros. If the stylus system were extremely simple, the distribution of these critical frequencies could be calculated in advance. Thus, for example, if the stylus system were to consist of a simple, straight wire or tube, the frequency of its first clamped-supported resonance $f_{1,cs}$ would be just 4.38 times $f_{1,cf}$, its frequency of free resonance; its second clamped-free mode would occur at 6.27 $f_{1,cf}$, and so on. On the other hand, practical stylus systems usually differ from this over-simplified idealization in several ways: The support of the stylus tip at the groove walls is offset from the structural axis of the suspension; the mounting tube or wire may not be uniform or straight, and it is usually end-loaded by the inertia of the stylus rocking about its tip; a resilient connection to the transducing mechanism may be attached at some intermediate point; and the so-called fixed end of the suspension is often less-than-rigidly clamped between visco-elastic blocks. These structural complexities do not alter the sequence in which $f_{1,cf}$ and $f_{1,cs}$ appear; they only alter the spacing between them and make it impractically difficult to calculate either frequency in advance of construction. For this reason it is often useful to prepare a large scale-model of the stylus suspension so that all its critical frequencies can be brought for study into the range which is accessible for microscopic and/or stroboscopic observation.

The relevant elastic property of the groove wall is the force with which it resists indentation by the spherical stylus tip, and the ratio of the variational component of this force to the variational indentation defines the effective dynamic stiffness of the stylus-groove contact. The classical Hertz theory of the contact between elastic solids[5] is usually relied on for the evaluation of this effective stiffness. The applicable result of the Hertz theory, for the present case in which an ideally hard spherical stylus attacks a cylindrically-warped elastic surface, can be stated as follows:

$$a^3 = (9F^2/16E_v^2 r)\,[1 \pm (r/2R')], \tag{7}$$

where a is the total indentation (cm) measured normal to the groove wall, F is the normal force (dynes) pressing the stylus against the groove wall, r is the radius of the stylus (cm), R' is the principal radius of curvature of the groove wall *before indentation* (cm), v is Poisson's ratio, and $E_v = E/(1-v^2)$ is the constrained or plane-stress elastic modulus of the record material. The plus sign is to be used when the groove surface is convex toward the stylus, the minus sign when it is concave. The dynamic stiffness of such a single stylus-groove contact, for small displacements normal to the groove wall, can be found by differentiating both sides of Eq. (7) and rearranging to yield

$$\frac{dF}{da} = K_r = \left[\frac{6E_v^2 rF}{1 \pm (r/2R')}\right]^{1/3} \quad \text{(dynes/cm)}. \tag{8}$$

For the practical case of two-wall stylus support, the corresponding dynamic stiffness for small lateral or vertical displacements can be computed in the same way after first combining the lateral and vertical components of the normal forces and displacements at each groove wall in the proper phase to represent lateral or vertical stylus motion. It turns out, interestingly enough, that the effective lateral and vertical stiffnesses for two-wall support in a 90°-groove are each identical with the normal stiffness described by Eq. (8) for a single groove contact. The relevant value of the effective stiffness of the actual stylus-groove contact can, therefore, be expressed by rewriting Eq. (8) in the form

$$K_r = 1/C_r = 16.08\, E_v^{2/3}\, r^{1/3}\, M_B^{1/3} \; [1 \pm (1/2\sqrt{2})\, kakr]^{-1/3} \quad \text{(dynes/cm)}, \tag{8a}$$

in which the numerical factor represents $(3\sqrt{2}\,g)^{1/3}$, the ratio r/R' has been replaced by $(1/\sqrt{2})\,kakr$ where a is the

[5] See, for example, S. Timoshenko and J. N. Goodier, *Theory of Elasticity,* 2nd ed., pp. 372-377 (McGraw-Hill Book Company, New York, 1951).

peak amplitude of the lateral or vertical modulation, and the normal force F has been replaced by $M_B g/\sqrt{2}$ so that the result can be expressed in terms of M_B, the total bearing weight (grams) supported by both walls of the groove. A numerical example will illustrate the typical magnitudes of this compliance and the associated indentation. If $E_v = 3.76 \times 10^{10}$ dynes/cm^2, as measured for one sample of vinylite, and if $r = 0.7$ mils $= 17.8 \times 10^{-4}$ cm, the product $(E_v^{2/3} r^{1/3})$ will be 1.36×10^6. For these values, and for the case of an unmodulated groove ($kakr = 0$), Eqs. (7) and (8a) reduce to

$$a = 0.475 \times 10^{-4} M_B^{2/3} \quad \text{(cm)} \tag{7a}$$
$$C_r = 0.046 \times 10^{-6} M_B^{-1/3} \quad \text{(cm/dyne)}. \tag{8b}$$

Equation (8b) indicates, as might be expected, that the stylus-groove compliance is smaller than the stylus compliance by two or three orders of magnitude. As a result, the groove-driven motion of the stylus will not be significantly altered by the finite stiffness of the stylus-groove contact except near the poles of Z_{ll}, where the driving-point mass reactance can become comparable with the elastic reactance of the stylus-groove contact. Equality of these reactances would correspond to a resonance, the nature of which can be explained by considering first an oversimplified case. Suppose, for example, that the stylus system comprises a single concentrated mass M_s supported by an ideal, weightless, elastic hinge. The driving-point impedance of such a system would have a pole $(1/j\omega C_{LV})$ at zero frequency, a pole $(j\omega M_s)$ at infinite frequency, and a zero at the frequency of free resonance. If such a system were driven by an elastic groove having compliance C_r, a "stylus-groove resonance" would occur at a frequency f_0, which can be evaluated, with the help of Eq. (8a), as

$$f_0 = [1/(2\pi)]\,(M_s C_r)^{-1/2} = 0.6382\, E_v^{1/3} r^{1/6} M_B^{1/6} M_s^{-1/2} \quad \text{(c/s)}. \tag{9}$$

This is the stylus-groove resonance identified by F. G. Miller[6] in his definitive analysis of the effect of groove-wall elasticity on stylus tracking. As Eqs. (8) and (9) indicate, both the effective stiffness of the stylus-groove contact and the frequency of the stylus-groove resonance are increased when the area of the contact circle on which the stylus bears is enlarged by an increase of either the bearing weight or the radius of the stylus. On the other hand, the indicated effect of bearing weight on the resonance frequency is more than counterbalanced when the design precept expressed by inequality (3) is used to specify the lumped stylus mass in terms of the bearing weight. Thus, when the limiting equality allowed by Inequal. (3) is introduced in Eq. (9), the stylus-groove resonance frequency is found to vary inversely with the cube root of the bearing weight, according to

$$f_0 = 0.03530\, E_v^{1/3} r^{1/6} a_s^{1/2} M_B^{-1/3} \quad \text{(c/s)}. \tag{10}$$

This stylus-groove resonance frequency can also be expressed usefully in terms of the frequency of free resonance since each varies inversely as the square root of the effective mass. For this purpose, Eqs. (9) and (10) can be divided by Eq. (6) to yield

$$f_0/f_{1,cf} = (C_{LV}/C_r)^{1/2} = 0.222\, E_v^{1/3} r^{1/6} x_s^{1/2} M_B^{-1/3}. \tag{11}$$

The quantitative predictions of Eqs. (10) and (11) can be demonstrated with a numerical example. If again $E_v = 3.76 \times 10^{10}$ dynes/cm^2 and $r = 0.7$ mil, and if $a_s = 1000\,g$, $x_s = 1.9$ mil, and M_B is in grams, Eqs. (10) and (11) reduce to

$$f_0 = 40.7\, M_B^{-1/3} \quad \text{(kc/s)} \tag{10a}$$
$$f_0/f_{1,cf} = 18.0\, M_B^{-1/3}. \tag{11a}$$

In practice, of course, it is seldom valid to assume that the effective stylus mass is a constant independent of frequency, as implicitly assumed in writing Eqs. (9) through (11). The actual stylus suspension will almost always qualify as a distributed system for which the driving-point impedance will exhibit the alternating sequence of maxima and minima described above. In this case the effect of C_r, the finite compliance of the stylus-groove contact, will be twofold. As viewed from the terminals of a constant-current (velocity) generator representing the recorded groove modulation, the compliance C_r will appear in parallel with the driving-point impedance Z_{ll}. Its first effect, therefore, will be to convert the overall impedance from Type C-L (capacitive at zero frequency, inductive at infinite frequency) to Type C-C. Its second effect, however, is more relevant in the present context. Although the zeros of Z_{ll} will not be shifted, the poles of Z_{ll}, which correspond to the series of clamped-supported modes of the stylus suspension, will all be shifted downward in frequency. As a net result of this interaction of C_r with Z_{ll}, the single stylus-groove resonance identified above will be replaced by a series of resonances, each member of the series representing a restrained, or clamped-supported, mode of the stylus system for which the "restraint" has been modified by the elasticity of the groove-wall "support." These resonances will be called the "modified stylus-groove resonances," or the "compliantly restrained modes" of the stylus system, and their frequencies will be designated as $f_{1,cr}$ $(< f_{1,cs})$, $f_{2,cr}$ $(< f_{2,cs})$, etc.

These are troublesome resonances because relatively little that is harmless can be done to suppress them. The pattern of vibration in such compliantly restrained modes is characterized by a relative maximum in the lateral or vertical motion of all parts of the stylus suspension, including especially its intermediate portions. The transducing mechanism is usually arranged to sense the motion at some such intermediate point, and hence these resonances will always appear in the electrical output as spurious responses. Moreover, if one of them falls within the frequency range of interest, the enhanced lateral and/or vertical response near resonance will affect the channel separation adversely and may even cause it to disappear or reverse. It is mandatory, therefore, for these resonances either to be suitably damped or to be located well above the frequency range of interest, or both.

The record itself provides one component of damping for the compliantly restrained modes. The plastic materials used in making records are less than perfectly elastic, and so the stylus-groove compliance C_r can be regarded as inherently lossy. Measured values of Young's modulus for such plastics commonly reveal imaginary (dissipative) components corresponding to values of Q ranging from 5 to 25. Of course, if such values of Q were representative of the total damping of the vibratory system, they would lead to resonance peaks which could not be tolerated if they were to fall within the frequency range of interest.

Additional damping of the stylus system is ordinarily supplied, in amounts ranging from moderate to excessive, by the viscoelastic blocks in which the base of the stylus suspension is clamped or by damping blocks bearing directly on the suspension. Unfortunately, the direct application of damping to the stylus system in this way exacts an unexpected penalty. Since the stylus-groove compliance is much smaller than the stylus compliance, it follows that when there is enough damping to suppress the restrained-mode resonances involving C_r, the stylus suspension itself will be grossly overdamped at other frequencies. As a consequence, adequate suppression of these resonances by damping alone can be achieved only at the cost of adding a substantial in-phase component to the stylus driving-force requirements throughout the mid-frequency range. This conclusion is similar to that reached in discussing the problem of tone-arm damping. In this case, however, the extra components of stylus reaction force will not alter the conclusions expressed by the design inequalities (1) and (3), since these inequalities are based on reactance relations at the extremes of frequency. On the other hand, one can expect that another apparently unrelated effect will make its appearance. The increased force reaction normal to the groove walls will lead to an increase in the frictional work performed at the sliding contacts between the stylus and the groove walls and, consequently, to an increase in the rate of wear of both stylus and record!

The foregoing can be summarized by stating that "stylus-groove resonances" will occur, that they constitute the clamped-supported modes of vibration of the stylus suspension as modified by the finite compliance of the "support" at the stylus-groove contact, that these resonances will occur at frequencies slightly lower than the corresponding natural frequencies for the same stylus system when it is rigidly clamped at the base and rigidly supported at the stylus tip, and that a moderate amount of damping of these compliantly restrained modes may be useful, necessary, or even unavoidable. It can also be concluded, however, that enough damping to suppress these modes completely is almost sure to be excessive, and that relief from their harmful effects should be sought by placing the resonances above the frequency range of interest. How far above involves other considerations which will be discussed in more detail below.

Translation Loss

Playback loss is an omnibus term describing any failure of the reproducer to recover fully the recorded signal. The major component of playback loss is usually the translation loss, which describes the effect of groove-wall elasticity that manifests itself as a variation of playback response with groove velocity. For any given amplitude of groove modulation, the playback level is usually lower for the inside grooves than it is for the higher velocity grooves near the outer edge of the record, and this outside-to-inside drop in playback level is defined explicitly as the translation loss (N_g, dB).

This loss has its physical origin in the fact that the groove-wall stiffness defined by Eq. (8) is slightly altered by the wall curvature produced by groove modulation. In effect, the groove wall becomes slightly stiffer on the concave side and less stiff on the convex side. As a result, the indentation of the groove wall produced by the total bearing force is reduced on the concave side and increased on the convex side. This differential shift of indentation from one groove wall to the other produces a slight displacement of the stylus with respect to the centerline of the groove, always in the direction corresponding to a reduction of the stylus excursion. The fact that these wall curvatures act to produce only small variations about the mean total indentation at each groove wall has two interesting consequences: 1. even a small net shift of indentation may represent a large reduction of the output signal when the recorded amplitude is much smaller than the total indentation, as it ordinarily is at very high frequencies; and 2. the nonlinear relation between stylus force and indentation can safely be "linearized" by retaining only the first term of its power-series expansion about the mean static indentation. This is the procedure followed by F. G. Miller,[6] who showed that the first-order effects of translation loss can be described by a translation-loss function G which enters as a multiplying factor in the fundamental-frequency output of the pickup; explicitly,

$$G(f/f_g) = 1 - (f/f_g)^2, \tag{12}$$

in which f_g is a characteristic cutoff frequency given by

$$f_g = 0.0485\, V\, (E_v/rM_B)^{1/3} \quad \text{(c/s)}, \tag{13}$$

where V is the groove velocity (cm/sec). If the sample values used above for E_v and r are introduced at once, Eq. (13) can be reduced to

$$f_g = 1.34\, V\, M_B^{-1/3} \quad \text{(kc/s)}. \tag{13a}$$

This relation can be particularized further by introducing velocities (21 and 50.5 cm/sec) appropriate for the inside and outside grooves of a 12-in. 33⅓-rpm record. The corresponding inside and outside cutoff frequencies will then be

[6] F. G. Miller, "Stylus-Groove Relations in Phonograph Records," Doctoral dissertation, Harvard University, 1950; issued as Tech. Memo. No. 20, Acoustics Research Laboratory, Harvard University, March 15, 1950 (only photo-copies now available, Ref. No. PB 125454, Office of Technical Services, U. S. Dept. of Commerce, Washington, D.C.). A brief summary of the results was presented in F. V. Hunt, *Acustica* 4, 33-35 (1954).

$$f_{gi} = 28.1\, M_B^{-1/3} \text{ and } f_{go} = 67.7\, M_B^{-1/3} \quad (\text{kc/s}). \quad (13b)$$

A translation loss factor τ can now be defined as the ratio of the response at the inside grooves to the response at the outside grooves. Equation (12) can be used to write this in the form

$$\tau = [1 - (f/f_{gi})^2]/[1 - (f/f_{go})^2]. \quad (14)$$

Since the inside cutoff is always the more restrictive, and since Eq. (13a) indicates that $f_{gi}/f_{go} = V_i/V_o$, Eq. (14) can be arranged in the more suggestive form

$$f^2_{gi}/f^2 = [1 - \tau\, (V_i^2/V_o^2)]/(1 - \tau). \quad (15)$$

If, as before, this relation is particularized for a 12-in. 33⅓-rpm record by setting $V_i/V_o = 21/50.5$, Eq. (14) reduces to

$$f_{gi}/f = [(1 - 0.173\,\tau)/(1 - \tau)]^{1/2}. \quad (15a)$$

All these relations, as well as the qualitative argument given above, indicate that translation loss will operate most drastically on signals lying in the upper portion of the frequency range of interest. For design guidance, therefore, these relations should be evaluated for a frequency f_u representing the upper cutoff of the recorded signal spectrum. On this basis, Eq. (15a) can be used to deduce that for the translation loss not to exceed 6 dB ($\tau = 0.5$) for frequencies $f \leqq f_u$, it is required that $f_{gi} \geqq 1.35\, f_u$. Similarly, for a translation loss limit of 2 dB, $f_{gi} \geqq 2.05\, f_u$.

One singular feature of the foregoing deserves emphasis. The translation loss, to the first order of approximation presented here, is entirely independent of the structural details of the pickup and the dynamics of the stylus suspension. This can be illustrated explicitly by combining Eq. (15a) and the first relation of Eq. (13b) to yield a design inequality relating M_B, f_u, and τ as follows:

$$M_B \leqq [(22.3 \times 10^3)/f_u^3] \quad (16)$$
$$[(1 - \tau)/(1 - 0.173\,\tau)]^{3/2} \quad (\text{gram})$$

where f_u is in kc/s. This relation leads at once to the impressive conclusion that the bearing weight must not exceed 1.3 gram if the design specification calls for a translation loss of 3 dB or less at an upper cutoff frequency of 15 kc/s. This is the first instance in this study in which a design criterion for adequate performance has dictated a specific numerical boundary on the designer's freedom of choice, but the implied thrust toward lighter bearing weight is clear and persuasive.

Scanning Loss

The low-pass filtering action of a finite scanning aperture is well known in other circumstances. It manifests itself, for example, as the gap-length effect in magnetic recording, as the slit-width effect in the photoelectric playback of motion-picture sound tracks, and as the aperture-loss effect in optical scanning for facsimile transmission. It can be expected, therefore, that the finite area of the circle of contact between the stylus and the groove wall will give rise to a similar scanning-loss effect in the phonograph playback process—and it does.

It is an understatement to say that the aperture effect in the stylus tracking problem has defied complete analysis. The real problem here, as in the previous cases involving the "elasticity" of the groove wall, is that of a hard sphere *sliding* over a *viscoelastic* surface of *unrestricted* curvature. Each of the emphasized words indicates by contrast a major inadequacy of present theory, which deals with the static contact between elastic solids whose radii of curvature are much larger than any dimension of their area of contact. Unfortunately, the repair of these inadequacies of present theory is a problem of intimidating difficulty which has not yet been solved. In the meantime, therefore, it is necessary to continue to pursue results of interim utility in terms of approximate solutions of restricted, but still relevant, problems.

In the only analysis of stylus-groove scanning loss so far undertaken, Miller[6] attacked the problem by assuming that the wall curvature due to modulation will produce a perturbation of the local stylus pressure acting on different parts of the area of contact. It is useful to contrast this approach with that used in dealing with translation loss. In the latter case, attention was focused on the change of indentation, or of transmitted force, as a result of changes of wall curvature from point to point along the groove; in effect, the amplitude of the loss was measured by the difference of wall curvature at points separated by half a recorded wavelength along the groove. It is not surprising, therefore, that a linearized analysis so oriented should yield a translation-loss factor pertaining only to the fundamental-frequency component of the pickup output.

On the other hand, the perturbation of stylus force accounted for as scanning loss is reckoned in terms of the change of wall curvature occurring in a distance extending only from one edge of the circle of contact to the other. As a first approximation, the change of curvature is reckoned from an initially plane condition, so that the change of curvature becomes just the curvature itself. Then, under the further restriction that the amplitude of modulation is small in comparison with the total indentation (which is satisfied at high frequencies if the distortion is modest), Miller's analysis led him to an evaluation of the scanning-loss factor as a power series in the argument $\pi\delta/\lambda = \pi f\delta/V$, where 2δ is the diameter of the circle of contact. The latter is given explicitly by

$$2\delta = 16.08\,(r\, M_B/E_v)^{1/3} = K_r/E_v \quad (\text{cm}). \quad (17)$$

The scanning-loss function is then given by the series

$$S(\pi\delta/\lambda) = 1 - \tfrac{1}{2}\,(\pi\delta/\lambda)^2 + \tfrac{1}{12}\,(\pi\delta/\lambda)^4 - \tfrac{1}{144}\,(\pi\delta/\lambda)^6 + \tfrac{1}{2880}\,(\pi\delta/\lambda)^8 - \ldots \quad (18)$$

The first zero of this function occurs when the argument $\pi\delta/\lambda = 1.90$, and this critical value can be used to define a characteristic cutoff frequency for scanning loss as

$$f_s = (1.9/\pi)\ (V/\delta) = 0.0755\ V\ (E_v/r\ M_B)^{1/3} \quad \text{(c/s)}. \tag{19}$$

The close kinship between scanning loss and translation loss can be indicated by comparing Eqs. (13) and (19). Such a comparison shows that f_s always equals 1.55 f_g. This relation can be cast in even more suggestive form by using Eq. (17) with Eqs. (13) and (19) to express the cutoff wavelengths in terms of the diameter of the contact circle; thus

$$V/f_g = \lambda_g = 1.29\ (2\delta) \tag{20}$$
$$V/f_s = \lambda_s = 0.83\ (2\delta) \quad \text{(cm)}.$$

These results provide an interesting comparison with the case of optical scanning (as of a sound track, for example) with a uniformly illuminated slit of width 2δ, for which the first scanning-loss null would occur when the modulation wavelength is just equal to the slit width. It is also interesting to observe that, although the analysis of translation loss did not take into account the finite size of the circle of contact, the conclusions drawn are very similar to those which would have been reached by considering the problem as one of scanning with a finite aperture. The reasons for choosing a different basis for analysis in these two cases, and the differences in the limiting wavelengths expressed by Eq. (20), are presumably lodged in the geometrical complications associated with tracing distortion.

The foregoing can be summarized by stating that groove-wall curvature and the finite size of the circle of contact cooperate to give rise to two loss factors: one is a translation-loss factor G which operates on the fundamental-frequency output, and the other is a scanning-loss factor S which operates on all output components. These loss factors are nearly equal in magnitude when each is very small ($\leqq$ 1 dB), but G decreases more rapidly with increasing frequency and carries the fundamental-frequency output to extinction at $f = f_g$. At this same critical frequency, S has a value corresponding to a scanning loss of about 8 db and an attenuation slope which is about –12 dB/octave at $f = f_g$ and which increases rapidly thereafter. As a result of this behavior, the abatement by scanning loss of all distortion products, including the pinch effect, can be characterized as the action of a low-pass filter having a nominal cutoff frequency approximately equal to f_g, the extinction frequency of the translation-loss function.

THE STRATEGY OF CRITICAL-FREQUENCY PLACEMENT

The distribution of the upper critical frequencies along the frequency scale is not uniquely determined by the considerations discussed above, except for the relations $f_{1,cf} < f_{1,cr}$ and $f_s = 1.55\ f_g$. It will be useful, therefore, to consider carefully what the designer can, or should, do about this frequency pattern.

In most cases the pickup designer has no direct control over the elastic modulus of the record material or the magnitude and range of variation of the groove velocity. The requirements of standardization and the hazards of tracing distortion also restrict severely his latitude of choice in selecting the stylus tip radius. Moreover, the stylus compliance can be presumed to be fixed within relatively narrow limits by the low-frequency tracking requirements expressed in the design inequality (1). It would appear, therefore, that the only mechanical variables remaining at the disposal of the designer are the bearing weight and the mass and structural form of the stylus suspension. The purpose of this section is to examine in detail how the designer can "spend" these remaining variables to best advantage.

Inequality (16) has already put in evidence a relation between bearing weight, translation loss ($N_g = -20 \log \tau$), and f_u, the upper boundary of the frequency range of interest. Since the designer *can* suppress translation loss almost completely by choosing these parameters appropriately, he has a clear obligation to do so. "Almost complete suppression" can reasonably be interpreted as a requirement that $N_g \leqq 2$ db for all frequencies below f_u. Then, for the value $\tau = 0.794$ corresponding to $N_g = 2$ dB, it follows from Eq. (15a) that $f_{gi} = 2.05\ f_u$. Inequality (16) can then be rearranged, with the help of Eq. (13b), to exhibit the relation between M_B and the frequency f_u below which $N_g \leqq 2$, as follows:

$$f_u = 13.74/M_B^{1/3} \quad \text{(kc/s)}, \tag{21}$$

where the numerical constant subsumes the same typical record material, groove velocity, and stylus radius used for Eq. (13a) and elsewhere in this text. The variation of f_u with the inverse cube root of M_B indicates that bandwidth free from translation loss comes at a high price in terms of bearing weight. Thus, for example, M_B must be reduced from 5 grams to 0.77 grams in order to extend the bandwidth without translation loss from 8 kc/s to 15 kc/s. As noted above, the implied thrust toward lighter bearing weights is clear and it should be persuasive.

It may be pertinent to interject here a comment about compensation for translation loss by pre-equalization in recording. It has been suggested that this could be accomplished by increasing the already high pre-emphasis at high frequencies as recording proceeds toward the inner grooves. Unfortunately, the recorded level at high frequencies cannot be further increased in this way without intolerable increases of distortion. While it is true that scanning loss would offset part of the increase, the analysis given above indicates that the net distortion will always be at least doubled by the added pre-emphasis needed to compensate for as much as 6 to 10 db of translation loss. There is also an even more persuasive reason for deprecating this practice. Advances in the state of the phonographic art will certainly be hampered by freezing into the recording characteristic a variable frequency response that would act as a barrier to further improvement of reproducing systems. For these reasons, the pre-equalization of recordings for compensation of playback loss should be stoutly opposed by all who seek to preserve or promote a high standard of playback quality.

After the bearing weight has been fixed, on the basis of a compromise with the demands of translation loss, the designer finds himself down to his last disposable variable with one more critical frequency to deal with. The modified stylus-groove resonance $f_{1,cr}$ must now be properly placed by a suitable choice and modification of the mass and structural form of the stylus suspension. An upper limit on the stylus mass is furnished by the design inequality (3), and it has already been suggested that $f_{1,cr}$ should be placed above the frequency range of interest. How much smaller the stylus mass should be and how far this resonance should be placed above f_u are questions which must now be considered in detail.

Two considerations influence the placement of $f_{1,cr}$: one is concerned with avoiding trouble and the other with achieving superior performance. Channel separation and frequency response are each degraded if $f_{1,cr}$ lies within the frequency range of interest, and a tracking hazard arises if it lies within the octave extending from f_u to $2f_u$. This is the octave containing the upper half of the spectrum of second-order distortion products which dominate in controlling the stylus acceleration demand. When $f_{1,cr}$ falls within this range, the enhanced stylus motion at resonance increases the acceleration demand and reduces in proportion the upper limit on stylus mass imposed by inequality (3). This consideration alone would be sufficient to dictate that $f_{1,cr}$ should be at least as high as $2f_u$.

The placement of the first compliantly restrained modes of the stylus system above the frequency $2f_u$ serves not only to avoid a tracking hazard but also to extend the pickup's range of uniform response toward $2f_u$. Such an extended response might appear at first glance to be a useless and expensive luxury since the recorded material and the electrical output circuit are each assumed to be sharply bounded by a cutoff frequency at f_u. The luxury may be expensive but it is by no means useless. In fact, it can be stated unequivocally that the response behavior of a pickup at frequencies beyond f_u can exert an important influence on its output at frequencies below f_u. Two prime examples, one old and one new, can be cited in support of this important basic principle.

It was pointed out more than twenty years ago that the second-order distortion which is ordinarily characteristic of vertical recording can be almost completely eliminated by the procedure of re-recording the vertical masters with reversed polarity before pressing. The efficacy of this method of reducing distortion was demonstrated in practice before World War II, and the basic validity of the procedure and the necessary conditions for its success were established analytically by Lewis and Hunt.[3] As a cardinal result of this analysis, it was shown that the passband of both the re-recording and the final reproducing channel must extend to an upper limit of $2f_u$ in order to achieve the desired cancellation of distortion at all frequencies below f_u. In effect, when the push and the pull of push-pull are to be achieved in two successive steps, each step must preserve intact *all* the second-order distortion in order for its final cancellation to be complete. The fact that 45-45 stereo records comprise inherently two orthogonal "vertical" recording channels does not impair the applicability of this technique. It would appear, therefore, that the almost-forgotten trick of inverted re-recording should by all means be revived. The major reduction of even-order distortion which can be realized in this way ought to provide ample incentive for the effort required to achieve the wider passbands needed.

The second example of an in-band effect of a pickup's out-of-band response has its roots in the nonlinearity of the stylus-groove contact and can be characterized as a "carrier effect." The nature of this effect can be illustrated by an analogy which will be familiar to anyone who has operated a high-gain radio receiver on which the AVC can be disabled, such as a "communications" receiver. Assume first that no input signal is present and that the rf gain is advanced until appreciable noise is observed in the audio output beyond the second detector. If the receiver is now tuned through the frequency of an *un*modulated carrier, the presence of this carrier will be announced by a substantial increase of the noise output. The cause of this noise increase can be explained as follows: in the absence of any carrier, the noise in the audio output consists of a summation of the difference tones caused by each component of the noise in the rf passband beating with every other component. However, since the rf noise is a continuous distribution, each "component" is infinitesimal and the resulting demodulated noise is of second order. On the other hand, when a large unmodulated carrier is present, each component of the rf noise can beat directly with the carrier to produce a first-order spectrum of noise in the audio output.

An exactly similar effect can, and presumably does, occur in the phonograph playback process. The first compliantly restrained mode of the stylus system, for example, can be excited at its resonance frequency either by surface noise or by the higher-order products of tracing distortion. Such excitation will be intermittent and asynchronous, and hence the stylus motion at its resonance frequency will be relatively steady and will vary in amplitude with the effective Q which characterizes the height and width of the resonance. Since this stylus motion at resonance is not synchronously locked with any component of the recorded signals, it qualifies as an externally introduced and essentially unmodulated carrier.

If the compliance of the stylus-groove contact were strictly linear, the presence of this vibrational carrier signal would be of no concern since neither the carrier nor its neighboring noise components could reach the band-limited output circuit. The stylus-groove contact *is* nonlinear, however, and so the noise and distortion components lying within f_u of the excited resonance frequency can intermodulate with the vibrational carrier to produce a contribution to the in-band noise below f_u. Quantitative data that would bear on the efficiency of this intermodulation process are elusive. In dealing with translation loss and scanning loss, it was necessary to "linearize" the stylus-groove compliance and to assume that the amplitude of modulation was small with respect to the total indentation. Sample calculations indicate that this assumption of smallness is just barely satisfied in many cases of interest. As a consequence, only a small resonance enhancement of the stylus motion beyond that dictated by the groove modulation will produce excursions that are wide enough to extend into the nonlinear regime of the stylus-groove contact. It is pertinent to recall at this juncture that the stylus-groove resonance cannot be completely suppressed except by an amount of damping which is otherwise excessive. It can be concluded, therefore, that the stylus-groove resonance will almost always be excited sufficiently to produce by intermodulation a substantial contribution to the in-band noise—unless something else is done about it.

Happily, something else *can* be done about it, and a particular arrangement of the upper critical frequencies appears to offer a novel opportunity to achieve uniquely superior performance. The key to this situation resides in the beneficial use of the scanning loss to prevent the excitation of stylus resonance. Consider the following premises: The inside cutoff frequency for translation loss f_{gi} should be placed at or above $2\,f_u$ in order to suppress translation loss throughout the range below f_u. The corresponding outside cutoff frequency f_{go} will be higher than f_{gi} by the ratio of the groove velocities, so that f_{go} will occur at approximately $5\,f_u$. It has also been pointed out that the cutoff, or extinction, frequencies for translation loss can be taken as the effective cutoff frequencies for the low-pass filtering action of the scanning loss. It follows then that the specification

$$f_{1,cr} \geqq f_{go} \tag{22}$$

will be sufficient to insure that the excitation of the stylus-groove resonance by either surface noise or signal distortion products will be effectively interdicted by the scanning loss and that the associated contribution of in-band noise will be eliminated. Inequality (22) can be designated, therefore, as a "low-noise" design specification.

The difficulty of precalculating $f_{1,cr}$ in practical cases has already been mentioned and model studies will usually be required to furnish the guidance needed in applying inequality (22). Its numerical implications can be exhibited, however, for the case of a lumped-constant stylus system, by dividing Eq. (9) by Eq. (13) and squaring, to yield

$$f_o^2/f_{go}^2 = 198\, r\, M_B/(V_o^2\, M_s) \geqq 1. \tag{23}$$

It should be noted in passing that this relation is independent of the record material. The limitation on stylus mass implied by the inequality in Eq. (23) can be put in evidence by assuming as before that $r = 17.8 \times 10^{-4}$ cm $= 0.7$ mil and that $V_o = 50.5$ cm/sec (outside of 12-in. 33⅓-rpm record), whereupon

$$M_s \leqq 138 \times 10^{-6}\, M_B \quad \text{(gram)}. \tag{24}$$

This inequality calls for a dynamic stylus mass of less than 0.14 mg for each gram of total bearing weight, a requirement that is more than twice as severe as the one established previously on the basis of acceleration demand.

No opportunity has yet been afforded to conduct a definitive test that would confirm the noise reduction predicted to flow from meeting the design specification expressed by inequality (22). However, encouraging support of the proposed hypothesis is furnished by the pre-World War II experience of Pierce and Hunt with an experimental pickup of 1940 vintage which they identified as Model HP-26A.[7] So far as the writer is aware, this is the only pickup ever constructed, prior to the present study program, which could qualify under the criterion expressed by inequality (22). Model HP-26A used a 2.85-mil stylus in a mounting which was elastically pivoted and which did vibrate as a lumped-constant system having an effective stylus mass of about 0.75 mg, and it tracked 78-rpm records satisfactorily at a bearing weight of 5 to 10 gm. Although we could not have known it at that time, since the stylus-groove resonance and the translation-loss cutoff frequencies were not identified until several years later, this experimental pickup must have operated with a critical-frequency ratio f_o/f_{go} in excess of unity, even at the outside edge of 12-in. 78-rpm records. One of the salient features of this pickup, which takes on new relevance in the present context, was the fact that its playback performance on either shellac or vinylite records was "quieter" than we then knew how to explain or justify on the basis of its over-all response characteristics. The considerations discussed above appear to furnish now, albeit two decades late, a plausible if not a proven explanation of its low-noise performance. Until a contrary alternative is established, therefore, it is proposed that the arrangement of the upper critical frequencies in accordance with the inequality (22) constitutes a proper goal for pickup design.

SUMMARY AND DISCUSSION

Some illustrative numerical conclusions based on the foregoing analysis are summarized in Table I, in which the stylus compliance, upper limits on the equivalent tone-arm mass and the stylus mass, and a few derived parameters are tabulated for bearing weights ranging from 20 mg to 28 gm.

A comparison of the entries in the first four columns of Table I with the published descriptions of commercially available tone arms and pickups supports two general conclusions: 1. almost all the commercially available pickups *do* meet the suggested specification for stylus compliance; 2. commercially available tone arms and styli, almost without exception, *do not* meet the suggested specifications for allowable mass. In fact, most of the tone arms and styli appear to be afflicted with equivalent overweight to a degree that would be called obesity if it occurred to the same extent in people!

In the face of such universal disregard of what has been described above as a necessary provision for the stylus acceleration demand, it may be useful to inquire further about the origin of this studied unconcern. The design inequalities (1) through (3) were formulated on the assumption that all the peak demands on the available driving force might need to be satisfied simultaneously. In practice, of course, each of the three separate requirements on the available driving force will have a statistical distribution in time and their peak demands will only coincide intermittently.

[7] F. V. Hunt and J. A. Pierce, *J. Acoust. Soc. Am.* **12**, 474 (A) (1941); also F. V. Hunt and J. A. Pierce, U. S. Patent No. 2,369,676 issued Feb. 20, 1945. The pickup described in these references, alas, was never marketed commercially.

TABLE I. Coordinated mechanical parameters and performance features for stereo pickups designed for different bearing weights.

	Primary parameters			Stylus-groove resonance (See Note 1)	Translation loss: outside/inside 12-in. 33⅓-rpm (See Note 2)	(See Note 3)	Low-noise design (See Note 4)	(See Note 5)
	C_{LV}	M_{ta}	M_s	f_o	N_g	f_u	M_s	f_o
Total bearing weight	(for $x_s = 1.93$ mil)	(for $a_{ta} = 0.026\ g$)	(for $a_s = 1000\ g$)	(for $a_s = 1000\ g$)	(for $f_u = 15$ kc/s)	(for $N_g \leq 2$ dB)	(for $f_o = f_{go}$)	($= f_{go}$)
gram	cm/dyne	gram	microgram	kc/s	dB	kc/s	microgram	kc/s
0.02	$750. \times 10^{-6}$	0.26	6.7	150.	0.15	50.6	2.8	249.
0.05	300.	0.65	16.7	110.5	0.28	37.3	6.9	184.
0.07	214.	0.90	23.3	98.8	0.36	33.3	9.7	164.
0.10	150.	1.3	33.3	87.8	0.46	29.6	13.8	146.
0.20	75.	2.6	66.7	69.6	0.74	23.5	27.6	116.
0.50	30.	6.5	167.	51.3	1.4	17.3	69.	85.3
0.70	21.	9.1	233.	45.8	1.9	15.5	97.	76.2
1.0	15.0	13.0	333.	40.7	2.5	13.7	138.	67.7
2.0	7.5	26.	667.	32.3	4.5	10.9	276.	53.7
3.0	5.0	39.	1000.	28.2	6.8	9.5	414.	46.9
5.0	3.0	65.	1667.	23.8	14.1	8.0	690.	39.6
7.0	2.1	91.	2333.	21.3	$f_u > f_{gt}$	7.2	966.	35.4
10.	1.5	130.	3333.	18.9		6.4	1380.	31.4
28.	0.5	324.	9333.	13.4		4.5	3860.	22.3

Notes: 1. From Eq. (10a); for lumped-constant stylus system. 2. From Eqs. (13b) and (15a). 3. From Eq. (21). 4. From Eq. (24). 5. From Eq. (13b).

At almost any given instant, therefore, any one of the reproducer functions can draw on a larger fraction of the total driving force than the one-third allocated to it in writing inequalities (1) through (3). These considerations allow two predictions to be made with equal confidence about a reproducer system that is overweight by no more than a factor of two or three: it *will* track satisfactorily for a large fraction of its playing time, and it *will fail* to track satisfactorily for a small and non-vanishing fraction of its playing time. These predictions are confirmed by many commercial pickups which are described as capable of operating with a range of bearing weights. Experience usually reveals that the maximum quoted weight must be used in order to avoid occasional "breakup" on "hot" records. For example, a few pickups are now available in which the reduction of stylus mass has been carried to the one-milligram level. These are commonly said to "track at one or two grams," but users usually concede that they "perform better" when operated at a bearing weight of two to four grams. A more realistic rating of such units would bring them into reasonable conformity with the specifications set forth in Table I.

There is a regrettable dearth of published information about the equivalent mass of commercial tone arms, most of which are merely described as "capable of taking any standard cartridge." There is some reason to think that many of these have not been appreciably lightened, since the "standard cartridge" was expected to operate at bearing weights of 7 to 12 gm. A whole new family of lightweight tone-arm designs will obviously be needed when pickups finally succeed in breaking through the "weight barrier" of 1 to 3 gm.

The stylus-groove resonance frequencies listed in the fifth column of Table I appear to lie safely above the frequency range of interest in all cases. It needs to be remembered, however, that these numbers apply only to lumped-constant stylus systems and that they are based on a stylus-mass specification that most pickups fail to meet by a factor of two to five. The more pertinent values of $f_{1,cr}$ will ordinarily be lower than the frequencies listed.

The magnitude of the translation loss for an upper cutoff frequency $f_u = 15$ kc/s, and the translation-loss-free bandwidth for $N_g \leq 2$ dB, are tabulated in Columns 6 and 7. The trend of these data shows clearly why substantial amounts of this kind of playback loss are routinely accepted as a phonographic way of life, but they show with equal clarity that reduced bearing weights can completely eliminate this nuisance.

The last two columns of Table I exhibit the allowable stylus mass for a lumped-constant system and the required frequency of the compliantly restrained mode for designs incorporating the proposed low-noise arrangement of the upper critical frequencies. As suggested above, these are intimidating specifications which currently available pickups fail to meet by a wide margin. However, the quest for an ultra-lightweight pickup which gave rise to this study has now yielded a group of experimental units which were designed for a bearing weight of 0.1 gm and which appear to meet these low-noise specifications. The problems arising in the construction of these units and in the assessment of their performance will be described in a subsequent paper.

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During World War II, Professor Hunt founded and served as Director of the NDRC-sponsored Harvard Underwater Sound Laboratory. In recognition of the distinguished record of this Laboratory, Professor Hunt received the Presidential Medal for Merit in 1947. In addition to his academic duties, he continues to serve on various governmental advisory committees dealing with undersea warfare and to act as a private consultant in the fields of electroacoustics and underwater sound.

Professor Hunt was one of the first group of four to be elected as Honorary Members of AES. He became the second recipient of the Emile Berliner Award in 1953, and the first to receive the AES Publications Award in 1956. He is a Fellow and past-President of the Acoustical Society of America, and a Fellow of the American Physical Society, the Institute of Radio Engineers, and the American Academy of Arts and Sciences. He is the author of numerous research papers in acoustics and applied electronics, and of the book *Electroacoustics* (Harvard University Press, 1954).