LETTERS TO THE EDITOR

NOTES ON PICKUP DESIGN AND RESPONSE

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Various factors influencing pickup response curves are discussed. Experimental results suggest that the high-frequency peak frequently seen on response curves of cantilever-type pickups is not the stylus–groove resonance, but a cantilever bending resonance. This resonance falls in the audio range due to the weight of the magnet or other material at the end of the cantilever. Torsional resonance gives a dip in response and is preceded by a peak. This peak is readily damped and goes unnoticed in rubber suspensions. Translation or differential curvature causes loss of treble. Deformation of the groove walls makes scanning loss of academic interest only at present.

This letter has been prompted by a recent paper by Sank [1], [2]. In discussing the response curve of a certain pickup, he suggests that the high-frequency peak may not be due to the stylus–groove resonance. This view is supported by tests made by this author in 1962. Other aspects of the response will also be discussed.

Stylus–Groove Resonance

Most moving magnet and induced magnet designs, where the magnetic material is near the compliance and is driven by a relatively long cantilever, have a response curve of the general shape shown in Fig. 1. Similar curve shapes are shown in Sank's Figs. 4 and 6 [1]. In early models of moving magnet pickups, the high-frequency peak was at 12–15 kHz; more recently, by reduction of the moving masses, the peak has been raised to 20 kHz. In some designs, the moving mass at the stylus tip was several milligrams, and this may well have put the resonance of the tip mass and the groove compliance in this same region of 12–15 kHz. However, other models with a tip mass of 2 mg or less likewise gave a peak at 12–15 kHz, with no evidence of any resonance higher up.

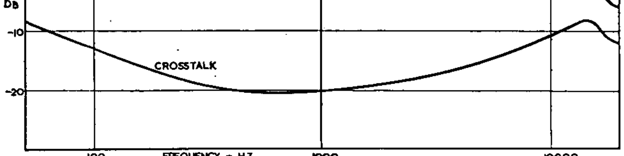

Fig. 1. Typical response curve of moving magnet pickup.

Previous experience over several years with a mono moving coil pickup also suggested that the stylus–groove resonance should be higher than this. The armature of this pickup was a stubby plastic moulding operating in torsion (Fig. 2). Bending resonances would therefore be unlikely in the audio range. With the simpler system in a mono pickup, with basically only one mode of movement, the only high-frequency resonance expected would be the stylus–groove resonance. As the system moved in torsion, there would be no other torsional resonance than the series midfrequency one of the torsional tip mass and torsional compliance of the suspension. Many thousands of these pickups were made and the response of each one was measured. There was one high-frequency peak only, at 19–21 kHz. The mass at the stylus tip, calculated from the dimensions of the moving parts, was 2.5 mg.

Hunt [3] gives the formula for the stylus–groove resonance as

$$f_0 = 0.6382\, E_v{}^{1/3} r^{1/6} M_b{}^{1/6} M_s{}^{-1/2}$$

where E_v is the plane strain elastic modulus of the record material, $E_v = E/(1-v^2)$, E being Young's modulus and v Poisson's ratio, r is the stylus radius, M_b the playing weight, and M_s the tip mass.

The value of E_v given by Hunt is measured as 3.76×10^{10} dyn/cm^2. Similar values have been obtained by the author on samples cut from records, and similar values are quoted by manufacturers of the vinyl compound.

For the mono pickup, with a 0.001-in stylus radius and a 3.5-g playing weight, the calculated stylus–groove resonance is 18.4 kHz, agreeing with the measured peak. It will be noted that the frequency is proportional to the

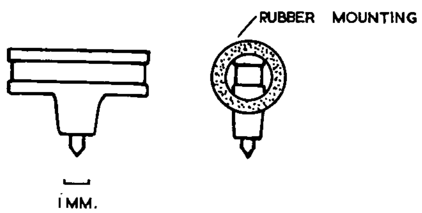

Fig. 2. Armature of mono moving coil pickup.

sixth root of the stylus radius, so that changes in stylus radius will not alter the frequency unduly. Fitting a 0.0025-in radius stylus raised the resonance from 19 to 22 kHz on vinyl, again agreeing with Hunt's formula.

The value of modulus used in these calculations is the static one. Under dynamic conditions, the effective modulus will be raised by any internal mechanical hysteresis in the record material. The vinyl record compound has good elastic properties, with very little creep, so that hysteresis losses will be small. The formula might underestimate the resonant frequency slightly, but would not overestimate it.

The formula is derived from the Hertz equations for the elastic contact of two curved surfaces and is thus only valid for loads not exceeding a few milligrams. Above this, plastic deformation will take place, at first below the surface, later reaching the surface to give a permanent concave track. Deformation continues until the area of contact is large enough to support the stylus without further plastic deformation. The material then behaves elastically, provided the load is not exceeded. The stylus–groove resonance constitutes elastic deformation of the plastically formed groove. Allowance should be made for the curvature of the permanent track, but the correction would not be large, and again Hunt's formula will not overestimate the resonant frequency.

If the author is correct in guessing the identity of Sank's pickup X (a moving coil model, made in Denmark about five years ago), it has a calculated tip mass of 1 mg. With a playing weight of 2.5 g and a stylus radius of 0.0007 in, the calculated stylus–groove resonance is 29 kHz, agreeing with the 30-kHz peak in the response curve. The other peak in the response curve of pickup X, at 15 kHz, is evidently due to some other cause. Obviously it cannot be the stylus–groove resonance, as the response would cut off above resonance. It may be assumed therefore that Hunt's formula will predict the stylus–groove resonance with reasonable accuracy.

Cantilever-Magnet Resonance

One of the earlier pickups examined had a magnet in the form of a 4-mm diameter sphere of ferrite, which was glued to a long tubular aluminum cantilever. The response was reasonably smooth, and extended to 15 kHz. The ferrite was mounted in rubber, and when this was changed for neoprene, the response showed a 10-dB peak at 8 kHz, and still continued to 15 kHz. A variety of different rubbers were tried, but only one gave a smooth response without the 8-kHz peak, viz., butyl—obviously a very useful material for pickup construction.

It seemed likely that the 8-kHz peak was associated in some way with the long cantilever and the very large magnet. The mass of the magnet was 150 mg, yet because of the cantilever length of some 10 mm, the effective mass of this magnet at the stylus tip was only 1.5 mg, the total tip mass being about 4 mg. The calculated groove–stylus resonance was 15 kHz.

At moderately high frequencies, the cantilever is approximately simply supported at the stylus. At these frequencies, the rubber mounting will be highly compliant, so that the cantilever can practically be considered free of support. The fundamental resonance of such a bar is zero. The first overtone is given by the formula

$$f_1 = \frac{2.45}{l^2}\sqrt{\frac{Qk^2}{\rho}}$$

where Q is Young's modulus of the cantilever in dyn/cm^2, k is the radius of gyration, $= \sqrt{r_o^2 + r_i^2/2}$, r_o is the outside radius, r_i the inside radius, l the length, and ρ the density.

This resonance is supersonic, but the weight of the magnet would lower the frequency considerably. It would also have the effect of reducing the amplitude at that point, due to its inertia, so that the actual motion of the cantilever would approach the condition of a beam simply supported at both ends. The formula for this resonance is

$$f_0 = \frac{\pi}{2l^2}\sqrt{\frac{Qk^2}{\rho}}.$$

Again, the magnet weight would reduce this considerably. Rangabe [4] has reached a similar conclusion.

Test results on assemblies of different proportions, originally made for other purposes, are of interest. The five assemblies had identical tubular aluminum cantilevers, and the magnets were all of the same weight, 15 mg. The length to cross-sectional area ratios of the magnets varied between 2 and 12.8. This varied the effective length of the moving assembly without altering cantilever and magnet weight, and with only a small change in tip mass. The assemblies were mounted in

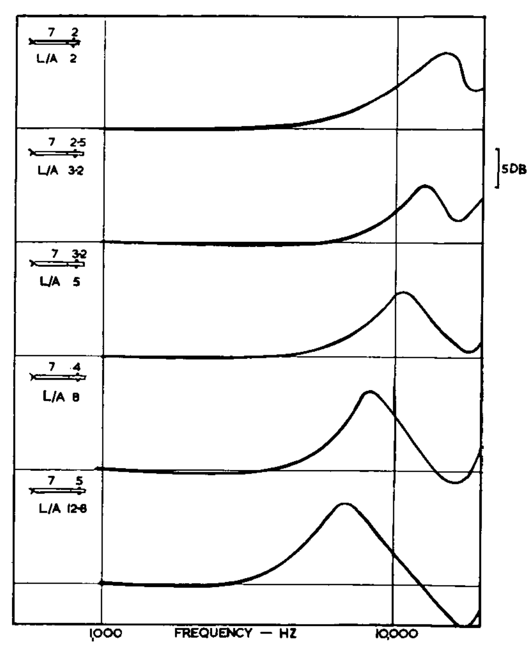

Fig. 3. Effect of magnet proportions on frequency response.

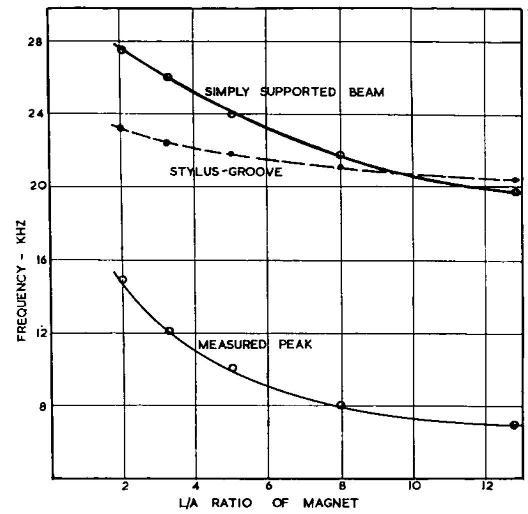

Fig. 4. Measured and calculated resonances for various magnet proportions.

turn in the same piece of neoprene, chosen so that resonances would not be heavily damped, and the same static assembly was used for each. Response curves are shown in Fig. 3. In Fig. 4, the $1/A$ ratios of the magnets are plotted against frequency of peak in the response, calculated stylus–groove resonant frequency, and calculated resonance of a simply supported beam.

It will be seen that there is a rapid decrease in frequency of the first peak with increasing effective length of assembly. There appears to be another peak just above 20 kHz, agreeing with the calculated stylus–groove resonance. The calculated resonant frequencies of the simply supported beam show the same trend as the measured first peak, but are much higher, as the weight of the magnet is not readily allowed for.

A magnet mass of about 15 mg and a length of 2–3 mm, with a cantilever length of about 7 mm and a pole spacing of 2–2.5 mm, represents the limit for this type of pickup. The first peak is at 15 kHz and the output is 1–2 mV/cm/s; yet the effective mass of the magnet only at the stylus tip is only a fraction of a milligram, most of the mass being that of the cantilever. This cannot be reduced very much without making it too slender. In recent years, the peak has been raised to 20 kHz by reducing the weight of magnetic material, especially in induced magnet designs, and the output has dropped as low as 0.6 mV/cm/s, giving a lower signal to amplifier noise ratio. Merely making everything smaller reduces output considerably; e.g., halving all dimensions resulted in a signal loss of 20 dB. The output could be increased by using more turns of finer wire for the coils, but the shunt capacity of the leads and contacts of most arms is too high, up to 200 pF per channel. This resonates in the audio range with the inductance of the coils, giving a peak, although this may be damped by the resistance present. Thick low-capacity coaxial cable is to be preferred for the leads.

The alternative to a heavy piece of magnetic material at one end of the cantilever is to dispense with it and use a magnetic cantilever in a moving iron design. The moving parts are usually very much smaller and tolerances on the position of the cantilever under load became critical, say ± 0.002 in, and this must be maintained throughout the working life of the assembly.

Torsional Stiffness

The two principal moving iron pickups use torsion as one of the modes of movement. Cantilever types have the advantage of symmetry of movement in all directions, but must have great torsional rigidity. At torsional resonance, the element twists without generating any signal, so that there is a dip in the response curve, and this should obviously be placed outside the working range, although this is not easy. In most designs, the dip comes after the cantilever peak and the stylus–groove resonance may be obscured by the dip. Like flexing of the cantilever, it has been claimed that the apparent tip mass may thus be reduced, but it is obviously better to place these resonances well above the working range.

Where the suspension is made of a material with very little internal damping, e.g., metal or rigid plastic, a peak occurs immediately before the torsional dip. An example is shown in Fig. 5, in which there was no magnet or other weight to give a bending resonance in the audio range. The peak is very easily damped and would be unsuspected in any type of rubber mounting. The cause of the peak is obscure, but it is clearly bound up with the dip and is always close to it, even though numerous parameters may be varied. A somewhat related type of resonance is well known to designers and users of vibration testing machines, but is seldom explained. With the specimen in place, a 'notch-peak' resonance occurs, sometimes several pairs with a complex specimen [5]. A dip followed immediately by a small peak also occurs in some loudspeaker systems, when the absorbent wadding is removed from the cabinet.

The suspension of the pickup element should be small, so that all movement takes place about a point, or nearly so. Any form of distributed compliance is likely to give pronounced overtones. The hinge must permit movement in two principal directions without movement in the third, and without torsional movement. Devices which would do this include bellows (with a tie wire), gimbals, and perhaps edge-wound springs and the Archimedian screw. Materials with preferred orientation might also be used to provide the different required compliances in the different directions. Some pickups use a short wire as the hinge. For a given load, the deflection in torsion varies as the length; in bending, the deflection varies as the cube of the length. If a long wire is used, high compliance will be obtained with high torsional rigidity. In practice, the wire is usually sufficiently long to give trouble with overtones, sometimes as low as 8 kHz, to which heavy damping has to be applied.

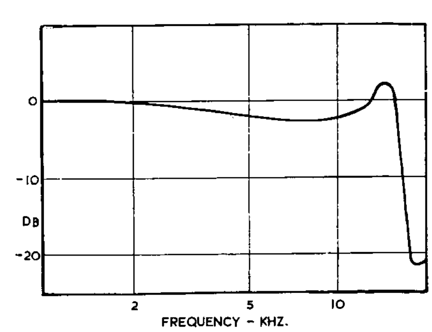

Fig. 5. Lightly damped moving iron cantilever pickup, showing peak prior to torsional dip.

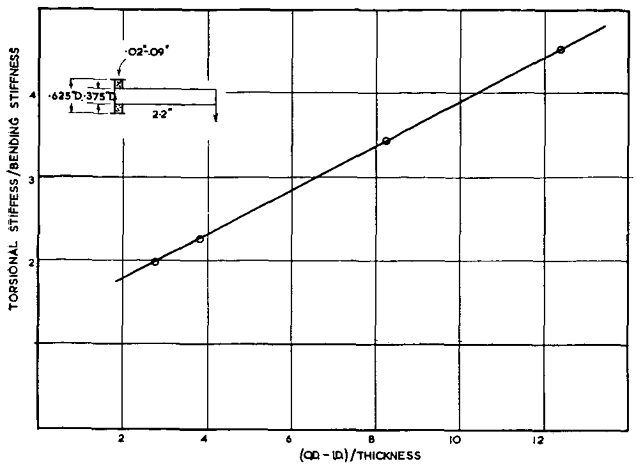

Fig. 6. Effect of thickness of rubber mounting on torsional stiffness to bending stiffness ratio.

A rubber mounting is frequently used, with or without a tie wire to prevent longitudinal movement. Bonding of the rubber to the cantilever is preferable, as it appears to raise the torsional resonance. To obtain the highest torsional rigidity with a given bending compliance, the rubber should approximate to a disk rather than a tube. This is shown in tests which were made on bonded rubber models, approximately 10 times full size. The cantilever length, outside and inside diameter of rubber, and hardness of rubber were constant, and the thickness of the rubber disk (i.e., in the longitudinal direction of the cantilever) was varied. The stiffness in torsion and bending were measured in g · cm/rad, and the ratios are plotted in Fig. 6 against the (outer diameter — inner diameter)/thickness ratios of the rubber. The advantage of using thin rubber is clear. In order to obtain the desired compliance with a thin rubber mounting, it may be necessary to use rather hard rubber, perhaps 70–80 shore *A*. To obtain this hardness in most rubbers, fillers have to be used, and elasticity and creep are usually poor. Certain grades of polyurethane are among the few materials with good elastic properties in the range of hardness.

Damping

Because of peaks in the response at high frequencies, damping is incorporated in most pickups, usually in the form of the internal mechanical hysteresis of the rubber suspension. A smooth response may thus be obtained, but the presence of damped resonances is usually betrayed by peaks in the cross-talk curves, sometimes becoming positive. The damping lowers the cross-talk figures, and if sufficiently heavy, reduces peaks in the cross-talk curves also. In this way, a pickup with numerous resonances in the audio range may have very good cross-talk figures. On the other hand, a basically good design, giving low cross talk due to avoidance of resonances, will require very little damping, so that its cross-talk figures, though quite adequate, may not appear as good as those of the first pickup. It will, however, be greatly superior to listen to.

To provide adequate damping at high frequencies, the damping at mid and low frequencies may be excessive. At the low-frequency resonance of the head mass and the element compliance, usually around 10 Hz, there is a peak in the response. On a normal response curve, not going below, say, 30 Hz, this can be seen as a gradual rise in the bass below 100 Hz. There have been some designs in which the internal damping of the rubber or other elastomer has been so high that the bass actually falls in this region. If the damping is excessive in the mid range, the load on stylus and groove walls will be high, in the very region where most of the information resides, and where it might be expected that tracking would be easiest. Hunt has pointed out that this will result in increased groove and stylus wear. In extreme cases, the loads required may be higher than at the ends of the spectrum. This is particularly true of some piezoelectric pickups, where heavy damping has to be used to suppress the resonances which fall in the audio range.

The three constituents of playing weight described by Hunt are 1) acceleration of the head mass at low frequencies, 2) suspension stiffness at low frequencies, and 3) acceleration of the tip mass at high frequencies.

For many pickups, a fourth component must be added, viz., internal hysteresis at mid frequencies, where velocity is highest. These four constituents will be additive under some conditions, and the playing weight must exceed the sum total. If internal hysteresis could be dispensed with, item 4) would of course disappear, 1) would be reduced, 2) would be easier to achieve, and 3) might be improved.

Ideally, any damping necessary should work only at the required frequency. It might be possible to choose materials which are more effective in the desired range than elsewhere. The damping properties of materials need to be measured over a range of frequencies, and over a range of static and dynamic stress. Some damping materials are effective only at high stress; others are the reverse, giving a pickup which measures well on test at low levels, but sounds bad on peaks. Full damping measurements are made on plastics and rubbers for civil engineering purposes at very low frequencies. Similar measurements in the audio range would be useful in all branches of electroacoustics.

A further consequence of heavy damping in the element compliance has been shown by Snell and Rangabe [6]. They measured the frictional drag on the stylus by the record groove, using an arm floating in oil. This is a radial tracking device, in which the float is held in position in the oil bath, in the axial direction of the arm, by means of magnetic repulsion [7]. By modifying the magnet system, the forward movement can be made appreciable, and can easily be measured by means of a scale. The system is easily calibrated by means of weights, and thus gives an accurate measure of forward drag under actual playing conditions. The drag was measured at various modulation levels. With pickups with very little damping, the drag varied very little with amplitude of signal; those with heavy damping gave an increase of over 80% drag on heavy modulation, due to the internal hysteresis of the suspension. Thus accurate bias compensation cannot be made for pickups with heavy damping, and a compromise value of bias must be used (or a satisfactory radial tracking device).

Element Compliance

The internal hysteresis in the element mounting reduces the effective compliance under dynamic conditions. In some designs, a static compliance of 30×10^{-6} cm/dyn has to be used to get a dynamic compliance of only one tenth of this value. Using rubber chosen for its good elasticity and freedom from creep, the dynamic compliance is likely to be only 40% of the static value. A very high static compliance is a disadvantage. The pickup must then be designed so that variations in playing weight do not degrade the performance, due to the magnet not being in the optimum position between the poles. One early design would not track at the recommended load of 1.5 g: this could not be increased very much, as the cantilever collapsed on to the poles at 2 g. A small percentage creep of the rubber over a period of the time also means considerable drift of the moving element from the correct position.

Low-Frequency Resonance

If the static compliance is very high, the dynamic compliance at very low frequencies is likely to be also very high. If the mass at the pickup head is high, due to shell, clamp, contacts, screws, etc., of a universal arm, instead of using an integrated design, the resonance will be very low. In one design Snell [8] found that this was as low as 3 Hz, so that it was very difficult to avoid groove jumping and acoustic feedback.

The low-frequency resonance should be lightly damped to prevent groove jumping and transmission of rumble. Bachman [9] has demonstrated that this should be done in the pivots of the arm, not on the element compliance. Critical damping of the resonance is not necessary and would require excessive playing weight to keep the stylus in contact with the groove on warps, runout grooves, etc.

An alternative is the tuned vibration absorber. This replaces one large peak with two smaller ones above and below. It may take the form of a lossy compliance between the counterbalance weight or head and arm. Tests with a compliance between counterbalance weight and arm showed that if the internal damping was excessive, very little effect was obtained; if the damping was insufficient, the two peaks were pronounced. Even with optimum damping, the upper peak may be high enough to give trouble on extreme bass notes. Light damping of the pivots was therefore preferred. The compliance on the counterbalance weight on some arms is of a different order of stiffness and would not affect the resonance. Construction of only moderate rigidity will give dips and peaks in the 200–400-Hz region.

Treble Droop

On many pickups there is a droop in the response above 1 kHz. Sank's Figs. 4 and 6 [1] are typical. The sag may be offset by high-frequency resonances, or may be accentuated by heavy damping. Scanning loss has been suggested as the cause. This is a geometrical effect when the trace becomes too small to be followed by the stylus, i.e., there is curvature overload, the minimum trace radius is less than, say, 0.0007 in. Ignoring deformation, there would be no scanning loss up to curvature overload, as the peak amplitude would remain unaltered. Constant voltage output would be obtained if measured on a peak-reading voltmeter, although there would be distortion. The sag in the response curves from record

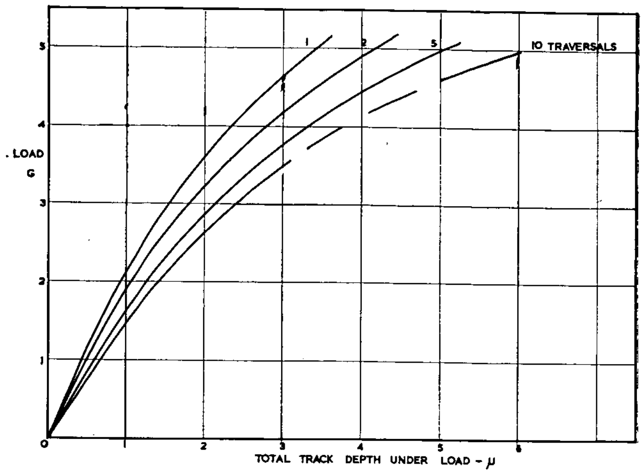

Fig. 7. Track depths on vinyl; 0.0007-in radius stylus.

STR 120 cannot be due to scanning loss, as there is no curvature overload except at the inner band above 20 kHz. In practice, when curvature overload approaches, deformation will be severe, with considerable ploughing of the trace, which cannot be calculated.

There may be translation loss. Under a given load on a flat surface, the stylus will penetrate elastically and plastically by a certain amount. On convex surfaces, i.e., the crests of the modulation, the penetration will be deeper; on concave surfaces, i.e., the hollows of the modulation, the penetration will be less for a given load. This gives a net loss of amplitude, and this is defined by Hunt as translation loss; a better term might be differential curvature loss. As much of the deformation is plastic with any practical pickup, the deformation cannot be obtained from equations based on the elastic case, although the general trend may be similar. With any practical pickup at high frequencies, the load on the crests of the modulation will be less than nominal, and in the hollows, it will be more than nominal, by the force required to accelerate the tip mass; i.e., the acceleration forces have the opposite effect to differential curvature. For a given playing weight, the effect of a significant tip mass is to reduce the load on the crests, to 'preserve' them compared with zero tip mass. Furthermore, the increased load in the hollows is taken by the areas most resistant to deformation. This is doubtless one of the reasons why records sound as well as they do, in spite of the deformation of the groove. Of course, this does not mean that a low tip mass should not be aimed at.

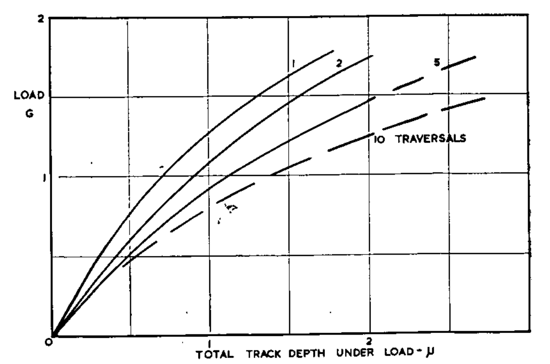

Fig. 8. Track depths on vinyl; 0.0008/0.0002-in radius stylus.

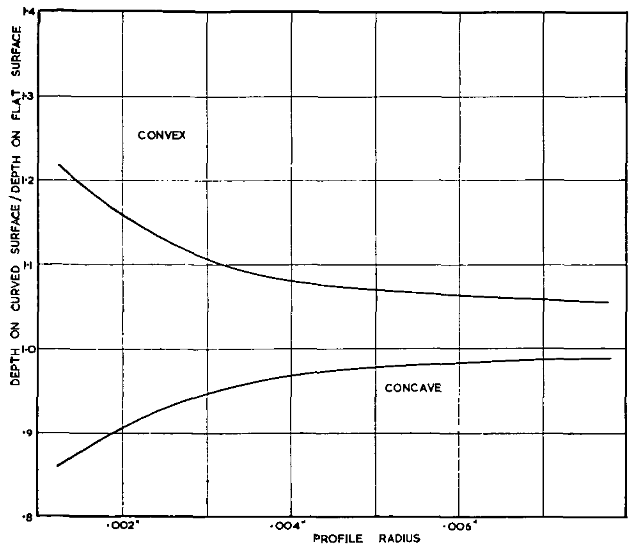

Fig. 9. Curvature ratios; 0.0007-in radius stylus.

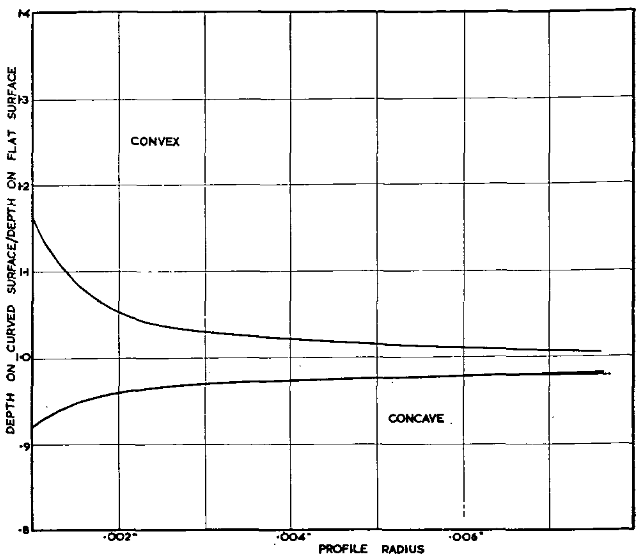

Fig. 10. Curvature ratios; 0.0008/0.0002-in radius stylus.

As part of a recent general investigation on groove deformation and distortion, measurements have been made on large-scale models of deformation under a sliding indenter, on both flat and curved surfaces, and curvature loss can be obtained from these. ⅜-in thick slabs of vinyl record compound were used, slowly cooled to avoid residual stresses. They were mounted on the faceplate of a lathe and rotated at the same linear speeds as records. A 1-mm radius indenter was used, mounted on a lever arm on the saddle of the lathe, and loads were applied by means of weights. The depth of penetration under load was measured by means of a dial gauge. This was mounted on the lever arm, close to the indenter, and the zero taken with a very small load (less than 1% of the major load). The depths under load, scaled down to a 0.0007-in radius stylus, are plotted in Fig. 7. Corresponding curves for a 0.0008/0.0002-in radius stylus are plotted in Fig. 8.

To measure depths on curved surfaces dynamically by the same method was obviously impossible, due to inertia forces, etc. Depths were therefore measured at slow speed on convex and concave profiles, and also on adjacent flat surfaces on the same piece of material under the same conditions. The ratios of depths on curved and flat surfaces were taken. Values are plotted in Figs. 9 and 10.

Using these values, the curvature loss has been calculated for one playing of the STR 120 record with 0.0007-in and 0.0008/0.0002-in radius styli, a playing weight of 1.5 g, and zero tip mass, i.e., constant load over the cycle. Figure 11 gives response curves for the first playing of STR 120, using a Shure M75-6 type 2 pickup, with 0.0006-in nominal (measured 0.0007-in) radius stylus, at a playing weight of 1.5 g. Figure 12 gives results for a 0.0007/0.0002-in stylus and calculated values for a 0.0008/0.0002-in radius stylus. The 0.0007/0.0002-in stylus curves were obtained on the above STR 120 record after the one playing with the 0.0007-in radius stylus. The deformation from the spherical stylus is much less than for the biradial one (about 60% of the depth), so that errors should be small. Also plotted are calculated values for a tip mass of 1 mg, forces other than the acceleration of the tip mass being ignored.

The forces required to accelerate the tip mass are not large, the acceleration at 8 kHz is only 102 g. Nevertheless, they have more than canceled the curvature loss. The net gain or loss in any given case will depend on conditions, including the amplitude of the recorded modu-

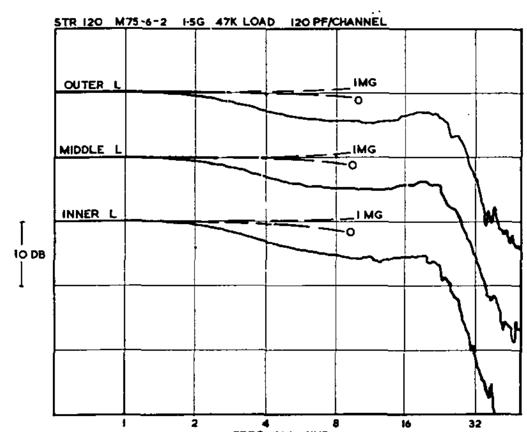

Fig. 11. Response curves and differential curvature loss; 0.0007-in radius stylus.

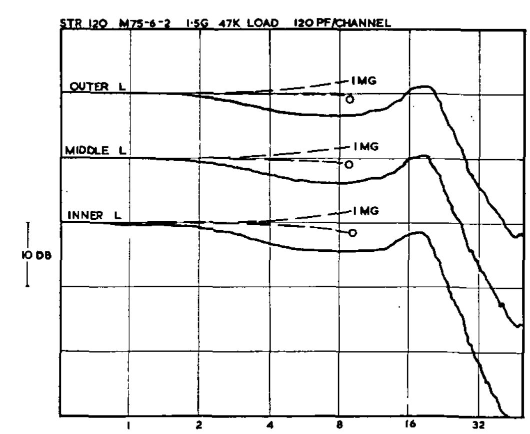

Fig. 12. Response curves and differential curvature loss; 0.0007/0.0002-in radius stylus.

lation. In general, the deformation from a biradial stylus is greater than from a spherical one, but the curvature loss is less, as the smaller radius is less affected by a given trace radius. This would explain the higher level of treble output often obtained with biradial styli. The main cause of the treble droop must lie elsewhere. The most likely cause is the shift of the actual center of oscillation from the nominal position, due to out of balance effects. Some designs of pickup give less droop than conventional moving magnet designs, perhaps because the axis of oscillation is more closely controlled.

Conclusions

1) The stylus–groove resonance may be calculated with reasonable accuracy from Hunt's formula [3].

2) The high-frequency peak present in many cantilever-type pickups is a function of the cantilever and the mass at the end remote from the stylus.

3) In symmetrical designs, torsional resonance occurs as a dip, preceded by a peak. The dip may obscure the stylus–groove resonance.

4) Differential curvature gives a loss of treble when tip masses are very low. The inertia forces have the opposite effect and may increase the treble with current tip masses. The main cause of treble droop is probably shift of the axis of oscillation of the generating element. Scanning loss is mainly of academic interest.

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