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(Abstract)

In the past , many studies have been made, and reports published, on the low-frequency resonance of tonearms [1], [2], [3]. But it seems that no analysis has been made yet of where the best position is on a tonearm for its pivot.

This paper introduces the new concepts of a static center of gravity and a dynamic center of gravity on a tonearm, and the optimum pivot position of a tonearm is analyzed in relation to those two points.

It has been found that, when a tonearm receives a force from the stylus tip or when any vibration is conveyed to the tip from the pivot position, the translational motion and rotational motion of the tonearm cancel each other out. This observation has enabled us to compute the optimum pivot position of a tonearm, and to couple our optimum pivot position theory with the theory of dynamically damping a tonearm, as proposed by other scholars and engineers in the past, to achieve more effective damping of the low-frequency resonance of a tonearm.

Introduction

Numerous studies and reports have been made about the design of tonearms in the past. For instance, methods of attenuating the low-frequency resonance are discussed in Reference [1] and [2], the design of a tonearm which would provide superior transient response characteristics in Reference [2] and [3], and the design of an especially lightweight tonearm in Reference [4]. Each in its own way deals with the theories and designs for a tonearm which would accurately trace the record groove without generating undesirable noise or resonance.

Of late, the attention of tonearm designers has been focused especially on the low-frequency resonance which is caused by the inertia moment of a tonearm and its stylus tip compliance. They have found that such resonance often results in a kind of frequency modulation of the audio signal [5].

In another development, the so-called dynamic damping theory [1] has emerged as a way of attenuating the low-frequency resonance of a tonearm, and several products incorporating the benefits of this theory have already appeared on the marketplace.

Such dynamically damped tonearms seem to be either of a design which divides the cartridge-fitted front portion of a tonearm and inserts a rubber damper at the junction, or of a design which divides the counterweight-equipped rear portion of a tonearm and damps it with a spring and silicon oil.

However, no argument has been made so far about where to position the pivot of such a dynamically damped tonearm and what percentage of the mass of the rear portion of a tonearm should be used as a damping force and what percentage to leave on the entire tonearm itself.

This paper makes the observation that a certain amount of the mass left on the rear portion of a tonearm works to prevent any undesirable vibration applied to the tonearm's pivot from reaching the stylus tip. From this observation a conclusion is derived about where the optimum pivot position on a tonearm should be, not only to avoid the deterioration of tonal quality normally occurring from the pivot vibration but also to attenuate the low-frequency resonance of the tonearm to a great extent.

It is expected that if a tonearm is actually developed to have such an "optimum pivot position," any vibration applied to its pivot will scarcely reach its stylus tip, thus effectively reducing and eliminating the possibility of howling.

Moreover, the tonal quality of reproduced sound is known to be affected by how the tonearm base is attached. Even this problem, however, can be solved by designing the optimum pivot position as subsequently described in this paper. This is simply because the relationship from the pivot of a tonearm to its stylus tip is reciprocal to the relationship from the stylus tip to the pivot, and the fact that no vibration received at the pivot reaches the stylus tip obviously means that no part of the signal (vibration) picked up by the stylus tip will escape to the pivot to create delicate changes in the tonal quality of reproduced sound.

On the Relationship Between the Center of Gravity of a Tonearm and Its Pivot Position

Vibration applied from outside to the pivot of a tonearm travels differently to the stylus tip depending upon the relationship between the pivot and the center of gravity of a tonearm, and this relationship is first reviewed for several conventional types of tonearms.

To simplify our discussion, we are going to pretend that each tonearm is shaped as a piece of straight tube, but the conclusions derived from the discusssion would also theoretically apply to any tonearm of complex shape.

First, Fig. 1 shows a dynamically balanced tonearm whose pivot is at its center of gravity, with the required tracking force applied by a spring. On such a tonearm, any vibration given the center of gravity,(H), appears at stylus tip (a) and counterweight (b) as identical amounts of vibration.

On a statically balanced tonearm such as is shown in Fig. 2, the pivot position is slightly shifted from the center of gravity toward the counterweight to permit application of a tracking force in the range of a few grams. Since the pivot is closer to the counterweight in relation to the center of gravity, any vibration received at the pivot tends to appear in a greater proportion at the counterweight than it appears at the stylus tip. Generally speaking, however, the displacement of the pivot position from the center of gravity is so small on most statically balanced tonearms that the vibration appearing at the stylus tip does not differ much from that received at the pivot.

Fig. 3 shows a tonearm where the pivot is shifted much closer to the counterweight and away from the center of gravity. Here the tracking force is adjusted with a spring attached to the pivot (H). The result is that any vibration given the pivot (H) appears at the stylus tip in antiphase.

The one thing that is clear from these observations is that most conventional tonearms incorporate no special measure of preventing the vibration received at the pivot from reaching the stylus tip. Conversely, there is no special concept incorporated in the tonearm design to prevent the vibration of the stylus tip from escaping to the tonearm pivot.

In our explanation of Fig. 3, we made an important observation -- that the vibration appearing at the stylus tip is in antiphase to the vibration received at the pivot. This fact presumes that there is a node of vibration somewhere between the pivot and the stylus tip.

In this paper we wish to introduce a theory of an optimum pivot position which would bring this "node of vibration" exactly to the stylus tip, working within the frequency range in which the tonearm can be considered a rigid body. At the same time, we propose to substitute a spring and a mass for the spring in Fig. 3, whereby the resonance relationship between this spring and mass is connected with the previous theory of dynamically damped tonearms. Results of the experiments we made to substantiate our theory are also presented along the way.

Theory of the Optimum Pivot Position

In physics, it is accepted that any force applied to any part of a rigid body will break down into a force that will translate the rigid body and force that will rotate it.

Now, if we could fix the pivot of a tonearm exactly where this translational force and rotational force cancel each other out at the stylus tip, then we would have a tonearm available whose stylus tip is completely immune to vibration from the pivot.

Fig. 4 shows a tonearm with the tubing, counterweight, phono cartridge and stylus, where the distribution of density is shown to be in effect quite arbitrary. We now designate the stylus tip as A, the center of gravity of the entire tonearm system as G and its pivot as P. We also fix the co-ordinates, and set G = $(x_G, 0)$, A = (0, 0) and P = $(x_i, 0)$.

Let us then assume that an external force F = $(0, F_y(t))$ is applied to P. From the laws of physics, we can assume that the tonearm system will then produce a movement that combines a translational motion and a rotational motion around the center of gravity (G).

We now proceed to compute the position of the pivot (P) which keeps the stylus tip position (A) where it is when the external force F is applied to the pivot (P).

(A) Translational Motion

If the mass of the entire tonearm system is m_T , the motion equation of the center of gravity (G) of the tonearm system in direction y is

$$m_T \frac{d^2 y_G}{dt^2} = F_y(t) \qquad --(1)$$

Therefore, the displacement y_G of the center of gravity (G) of the tonearm system in direction y by the force Fy (t) is

$$y_G = \frac{1}{m_T}\int_0^t \int_0^t F_y(t)\,dt\,dt + v_0 t + y_0 \qquad \text{--(2)}$$

If then the initial velocity $v_0 = 0$ and the initial position of the system $y_0 = 0$ at time $t = 0$, the displacement y_G is expressed as

$$y_G = \frac{1}{m_T}\int_0^t \int_0^t F_y(t)\,dt\,dt \qquad \text{--(3)}$$

Thus by the external force F, applied to point P, the tonearm system translates by the distance of y_G.

<u>(B) Rotational Motion</u>

The rotational motion equation of the tonearm system around its center of gravity(G)is

$$I_G \frac{d\omega}{dt} = M \qquad \text{--(4)}$$

where I_G = the moment of inertia around the system's center of gravity G, ω = angular velocity, and M = the moment of force. Since the external force, expressed as Fy (t) in equation (1), applies to point(P)only, the moment of force M around the system's center of gravity (G) is

$$M = (X_i - X_G)\cdot F_y \qquad \text{--(5)}$$

When terms in equation (4) are substituted for terms in this equation and rearranged, we have another equation:

$$\frac{d\omega}{dt} = \frac{M}{I_G} = \frac{x_i - x_G}{I_G} F_Y \qquad \text{--(6)}$$

Therefore, the rotary angle θ caused by the external force, around the system's center of gravity (G) is

$$\theta = \frac{x_i - x_G}{I_G} \int_0^t \int_0^t F_Y(t)\,dt\,dt + \omega_0 t + \theta_0 \qquad \text{--(7)}$$

Then if the initial angular velocity ω_0 = 0 and the initial rotary angle θ_0 = 0 at time t = 0, then equation (7) is expressed as

$$\theta = \frac{x_i - x_G}{I_G} \int_0^t \int_0^t F_Y(t)\,dt\,dt \qquad \text{--(8)}$$

In order that the stylus tip (A) remain stationary when the system's center of gravity (G) has displaced by the distance of y_G in direction y ,the tip should be at the rotational center of the system itself. Therefore, if the rotary angle θ is infinitely small, the displacement y_G is

$$y_G \fallingdotseq x_G \theta \qquad \text{--(9)}$$

From equations (3), (8) and (9) the following equation is obtained:

$$\frac{1}{m_T} \int_0^t \int_0^t F_y(t)\,dt\,dt = x_G \frac{x_i - x_G}{I_G} \int_0^t \int_0^t F_y(t)\,dt\,dt$$

$$\therefore \quad x_i = \frac{I_G}{m_T x_G} + x_G \qquad \text{--(10)}$$

Thus if three parameters--the position of the center of gravity X_G , the mass m_T and the moment of inertia I_G around the center of gravity (G) of the sytem--are determined, then it is possible to calculate the optimum position X_i of the pivot (P) of the tonearm system, from equation (10).

Here in Fig. 4 the moment of inertia I_G around the system's center of gravity (G) is expressed as

$$I_G = \int \rho(r)\, r^2\, dr \qquad --(11)$$

where $\rho(r)$= the density of the volume element of the system at a point away from the center of gravity (G) by the distance r .

From the aforesaid, it follows that if the pivot (P) is set at the point X_i calculated from equation (10), the force at the pivot does not interfere with movement of the stylus tip (A) and, conversely, the force at the stylus tip (A) does not affect the movement of the pivot (P).

Two conclusions may be drawn from this statement. First, as you see from Fig. 5, the spurious vibrations, as applied to the pivot, are converted into circular motion of the system around the stylus tip (A) and therefore do not apply undue force at the tip, eliminating the chance of adding audible noise to sound reproduction. Second, the movement of the stylus, caused by applied signal, is turned into circular motion of the system centered at the pivot (P) and therefore does not add undue force at the pivot. From these two conclusions, it is clear that no tonal degradation occurs with this system in whatever physical condition its pivot is.

Yet the optimum pivot position X_i of the system calculated from equation (10) is usually away from the position of its center of gravity X_G, resulting in application of excessive tracking force to the stylus tip unless some means is used to avoid it. We need some mechanism in the system to approximate the position of its static center of gravity to its optimum pivot position, without changing the position of its dynamic center of gravity x_G.

In the tonearm in Fig. 6, static balance is achieved by a spring-loaded weight, designed to apply required tracking force to the stylus tip. In this arrangement, it is conceivably possible, by successfully tuning the spring-loaded weight system to the resonant part of the tubing system, to damp low-frequency resonance of the entire tonearm system. Here our theory of the optimum pivot position of a tonearm and the theory of dynamic damping are effectively combined.

Next, let us consider how to determine the mass of the weight for static balance of a tonearm in Fig. 6 when its pivot is set at the optimum position calculated from equation (10).

Fig. 7 represents the tonearm model with which the pivot is set at the optimum position calculated from equation (10). Point (A) is the stylus tip which forms the origin; point (G), the center of gravity of the section of the tonearm from (A) to (B); point G_1, the center of gravity of the section of the tonearm from (A) to (P); point (G_2), the center of gravity of the section of the tonearm from (P) to (B); (G_3), the center of gravity of the weight (C); and P the optimum pivot position, given from equation (10).

Then where

m_T = the mass of the section of the tonearm form (A) to (B)

m = the mass of weight (C)

$\rho(r)$ = the density of the tonearm at an arbitrary point anywhere from (A) to (B) which is away by the distance r from the pivot (P).

g = acceleration of gravity

the balance equation of the tonearm system, with its static balance achieved around the pivot (P), is

$$(x_i - x_{G_1})\, g \int_0^{x_i} \rho(r)\, dr = (x_{G_2} - x_i)\, g \int_{x_i}^{x_B} \rho(r)\, dr + (x_{G_3} - x_i)\, g\, m \qquad --(12)$$

$$m = \frac{(x_i - x_{G_1})\int_0^{x_i}\rho(r)\,dr - (x_{G_2} - x_i)\int_0^{x_B}\rho(r)\,dr}{x_{G_3} - x_i}$$

$$= \frac{x_i\int_0^{x_B}\rho(r)\,dr - \left\{x_{G_2}\int_0^{x_B}\rho(r)\,dr - (x_{G_2} - x_{G_1})\int_0^{x_i}\rho(r)\,dr\right\}}{x_{G_3} - x_i}$$

$$\therefore \quad m = \frac{(x_{G_2} - x_{G_1})\int_0^{x_i}\rho(r)\,dr - (x_{G_2} - x_i)\, m_T}{x_{G_3} - x_i} \qquad \text{--(13)}$$

Thus when the pivot is set at the optimum position in Fig. 7, the mass m of the weight required to achieve static balance of the system, is given by equation (13).

Theory of Dynamic Damping

Now let us apply to this tonearm the theory of an electric equivalent circuit we have developed collaterally for this paper, to analyze the theory of dynamically damping a tonearm. Fig. 8 shows the model of a tonearm dynamically damped.

When substituting an electric equivalent circuit for a mechanical vibration system, one in series is often preferred. But since we believe that the equivalent circuit in parallel far more fittingly corresponds to the mechanical vibration system, we have developed the following discussion around the electric equivalent circuit in parallel.

Here in this parallel circuit, the velocity of the mechanical system is replaced by the voltage of the electric system and the force, by the current. Fig. 9 represents the electric equivalent circuit as seen from the stylus tip of the tonearm, where

e_0 = true input to the cartridge (velocity of the cartridge body, relative to the stylus tip)

e_s = signals recorded on record (constant-velocity amplitude)

and

L_1 = Ca (stylus-tip compliance)
R_1 = $1/r_a$, c = $1/\ell_a^2$ (effective tonearm mass)
L_2 = $n^2 C_b$, $R_2 = n^2/r_b$, $C_2 = n^{-2} m$
= $\ell a/\ell b$

--- (14)

From Fig. 9 the following equation is obtained:

$$\frac{e_o}{e_s} = \frac{S^4 L_1 C_1 R_1 L_2 C_2 R_2 + S^3 L_1 L_2 R_1 (C_1 + C_2) + S^2 L_1 R_1 R_2 (C_1 + C_2)}{S^4 L_1 C_1 R_1 L_2 C_2 R_2 + S^3 L_1 L_2 \{R_1 (C_1 + C_2) + C_2 R_2\} + S^2 \{L_1 R_1 R_2 (C_1 + C_2) + L_1 L_2 + R_1 L_2 C_2 R_2\} + S (L_1 R_2 + R_1 L_2) + R_1 R_2} \quad \text{--(15)}$$

where: $S = j\omega$ (j : Imaginary unit).

Therefore, the frequency response of the signal from the stylus tip is

$$R = 20 \log_{10} \left| \frac{e_o}{e_s} \right| \quad \text{--(16)}$$

<u>Experiment on the Optimum Pivot Point--Experiment I</u>

The entire system arranged in our experiment is shown in Fig. 10. Observations were made on how the physical discrepancy between the theoretical optimum pivot position (P) and the actual pivot position (H) would affect the response of a signal from the stylus tip when the pivot (H) was vibrated. In this experiment, we gradually sawed off the rear end (Q) of the tonearm in Fig. 10 until the actual pivot position H best approximated to the optimum pivot position (P), calculated from equation (10).

Tracking force was adjusted by moving the weight m . Table 1 charts the experimentally obtained optimum pivot position of each of five tonearms, where m_T = the mass of the entire tonearm system (stylus, cartridge, shell, tubing) but the dynamic damping section, X_G = the position of the system's dynamic center of gravity, I_G = the moment of inertia of the tonearm system around point(G), X_i = the optimum pivot position, computed from equation (10), X_h = the actual fixing position of the tonearm, I_h = the moment of inertia of the tonearm system around the pivot (H).

Fig. 11 through 15 show the frequency response of the signal from the stylus tip, derived from five tonearms--No. 1 through No. 5--all pivoted at X_h . In each case, the pivot of the tonearm was subjected to a vibration, caused by signals with frequencies covering a 2Hz to 1kHz range and in 0.3cm/sec constant-velocity amplitude.

Since our experiment was intended to cover the theory of dynamic damping as well, all except the No. 1 tonearm had a dynamic damping system at their rear. Silicon oil of 10,000 pcs formed the damping system. Of course, the same cartridge was mounted on each of the tonearms in Table 1.

What we found out from Table and Figs. 11 through 15 was this: With the No. 4 tonearm that had the smallest discrepancy ΔX among five tonearms, the effect of the vibrated pivot point (H) on the frequency response was smallest, notably in the range below 150Hz. It was thus proved that our theory of the optimum pivot point of a tonearm was experimentally correct as well.

Experiment on Dynamic Damping--Experiment II

Figs. 16 and 17 represent the response of the No. 1 and No. 4 tonearms we made for the above-described experiment intended to determine the optimum pivot position of a tonearm. In Fig. 16 you see a pronounced low-frequency resonance peak of about 14dB, because the No. 1 tonearm lacked a dynamic dapming system.

In Fig. 17, then, you see a damped low-frequency resonance peak of only 9dB--about 5dB less than that of the No. 1 tonearm--because the No. 4 tonearm had a dynamic damping system.

Then we had a computer simulate the low-frequency resonance, so that it would repeat the patterns obtained in the experiment.

The equivalent circuit of the No. 1 tonearm was as shown in Fig. 18. From Table 1, the effective mass m_a, as seen from the stylus tip of this tonearm, is

$$m_a = \frac{I_h}{X_h^2} = \frac{14787}{(23.3)^2} = 27.2\ (g) = C_1 \qquad --(17)$$

If we set the low-frequency resonance at 8Hz from Fig. 16, then the following equation is obtained.

$$f = \frac{1}{2\pi\sqrt{C_1 L_1}} = 8\ (Hz) \qquad --(18)$$

Therefore, the compliance Ca of an unknown cartridge may be calculated by the following equation.

$$Ca = L_1 = \frac{1}{(2\pi f)^2 C_1} = 14.6 \times 10^{-6}\ (cm/dyne) \qquad --(19)$$

Using the figures thus obtained, we had the computer describe a pattern looking like the one in Fig. 16. Then we determined the mechanical resistance r_a of the stylus tip, that is, r_a = 280 (dyne sec/cm) = $1/R_1$. From the equivalent circuit in Fig. 18, the frequency response of the signal from the stylus tip R(s) is

$$R = 20 \log_{10}\left|\frac{e_o}{e_s}\right| \qquad --(20)$$

where

$$\frac{e_o}{e_s} = \frac{S^2 L_1 C_1 R_1}{S^2 L_1 C_1 R_1 + S L_1 + R_1} \qquad \text{--(21)}$$

The response in Fig. 18 is the result described by the computer.

Fig. 19 is the computer-simulated response of the No. 4 tonearm. Since the same cartridge as mounted on the No. 1 tonearm was used again, we deemed compliance $C_a = 14.6 \times 10^{-6}$ (cm/dyne) and mechanical resistance $r_a = 280$ (dyne sec/cm). Therefore from Fig. 9 and equation (14), $L_1 = 14.6 \times 10^{-6}$ (cm/dyne) and $R_1 = 1/280$ (cm/dyne sec). And from equations in (14) where $\ell_a = x_h$ and $I = I_h$, and from Table 1, the equivalent mass m_a, as seen from the stylus tip of this tonearm, is

$$m_a = \frac{I_h}{x_h^2} = \frac{13156}{(23.3)^2} = 24.2\ (g) = C_1 \qquad \text{--(22)}$$

The mass of the weight m for low-frequency damping actually measured 47.4g, and ℓ_2, 6.8cm. Thus from equation (14), the equivalent mass at the stylus tip is

$$C_2 = \frac{\ell_b}{\ell_a} m = 4.0\ (g) \qquad \text{--(23)}$$

Based on the figures thus obtained, we had the computer describe a pattern looking like the one for low-frequency resonance response in Fig. 17. From the pattern we determined the compliance and mechanical resistance of the spring for dynamic damping at 80×10^{-6} (cm/dyne) and 180 (dyne sec/cm) respectively. Therefore, $L_2 = 80 \times 10^{-6}$ (cm/dyne) and $R_2 = 1/180$ (cm/dyne sec).

Substituting the figures thus obtained for the parameters in Fig. 9, we had the computer calculate equation (16), the result of which is shown in Fig. 19. Fig. 20 is the result of computation with all the parameters remaining the same as in Fig. 19, but $R_2 = \infty$. This response represents the case where the mechanical resistance for damping--silicon oil--is removed from the dynamic damping mechanism.

Fig. 21 shows the result of our computation when $R_2 = 0$. It represents the case where the dynamic damping weight is fixed on the tonearm, that is, where there is no dynamic damping with the pivot set away from the optimum position.

Conclusions

We have thus proved both theoretically and experimentally that a tonearm has an optimum pivot position.

It is also proved that when the tonearm is optimally pivoted, vibrations barely travel from the pivot to the stylus tip and therefore the tonearm is immune to cabinet-borne vibrations, resulting in dramatic reduction of so-called "howling." The reciprocal consequence is that the influence of the stylus tip on the pivot would likewise be lessened.

Furthermore, it is possible, by coupling a matched dynamic damping mechanism with an optimally pivoted tonearm, to attenuate low-frequency resonance.

In this paper we have shown a method for calculating the optimum pivot position for a tonearm. Yet there may be those who think that our theory does not hold true any more when the cartridge or shell is replaced by different ones, often with different weights from the original ones as in actual turntable operation. Our theory does hold true, however; our calculations and experiments have shown that no change in mass around the stylus tip point or the optimum pivot point affects the validity of our theory, because, in a word, these points are at the "nodes of vibration."

Also from the observations above, it is clear that the parameters involved are less affected by external influences when the mass is concentrated at the optimum pivot point, as Fig. 22 shows, and that then by approximating the actual to the theoretical values, the optimally pivoted tonearm may be designed even if the parameters in equation (10), such as I_G/m_T or X_G, are changed.

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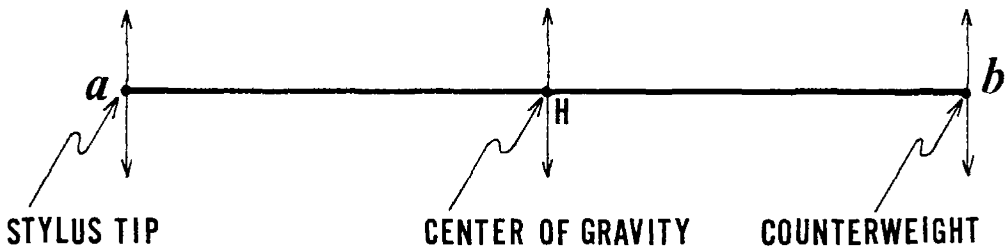


FIG. 1

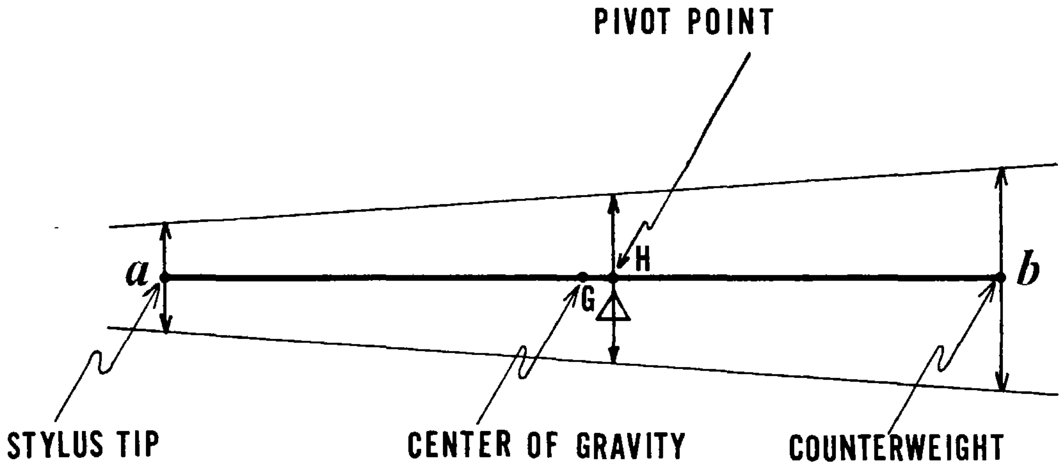

PIVOT POINT
H
a
G
b
STYLUS TIP
CENTER OF GRAVITY
COUNTERWEIGHT

FIG. 2

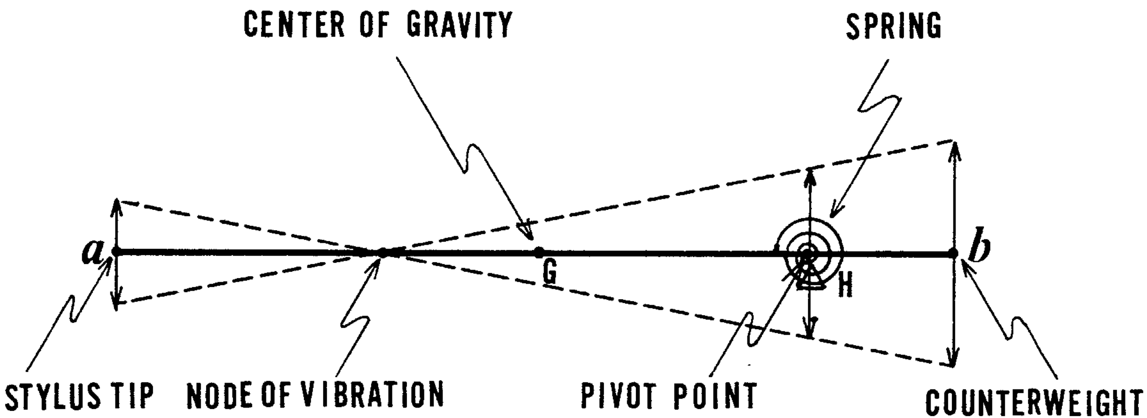

CENTER OF GRAVITY
SPRING
a
G
H
b
STYLUS TIP
NODE OF VIBRATION
PIVOT POINT
COUNTERWEIGHT

FIG. 3

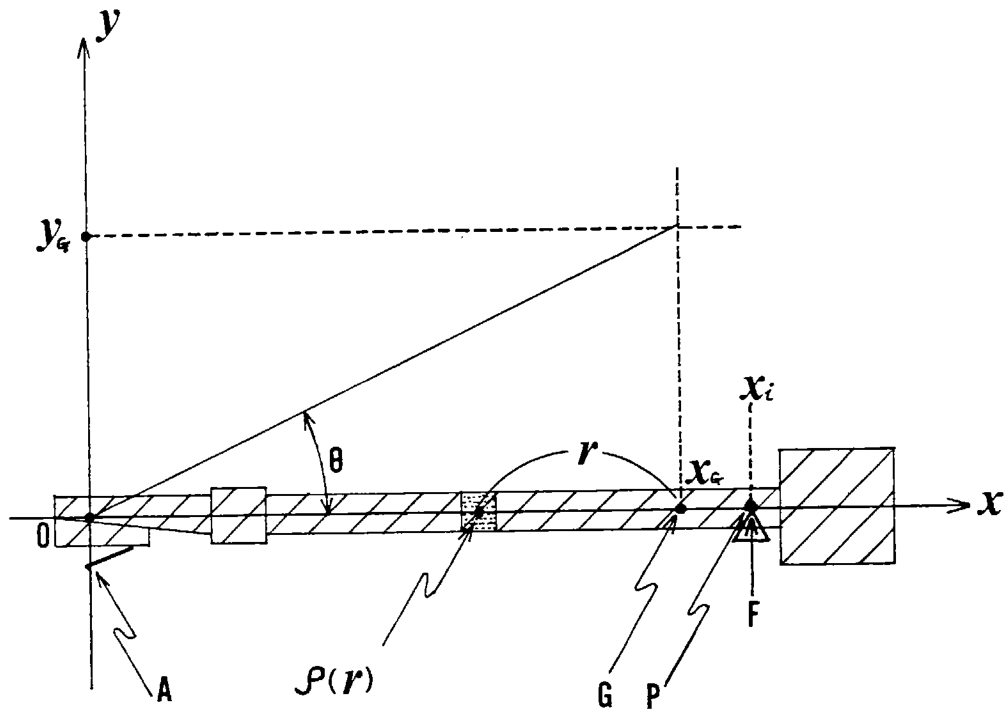


FIG. 4

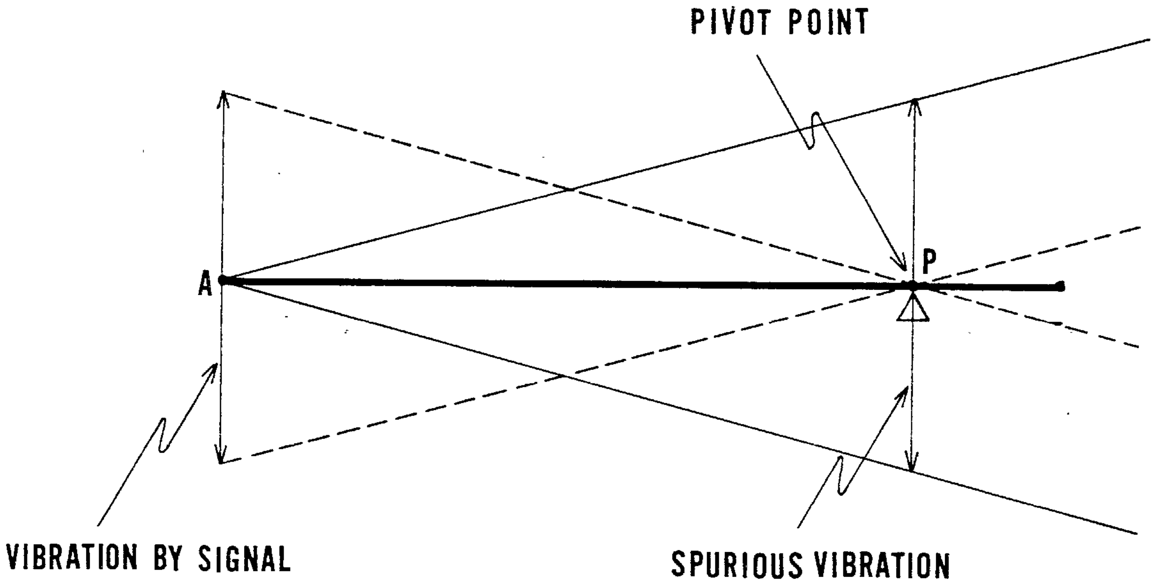


FIG. 5

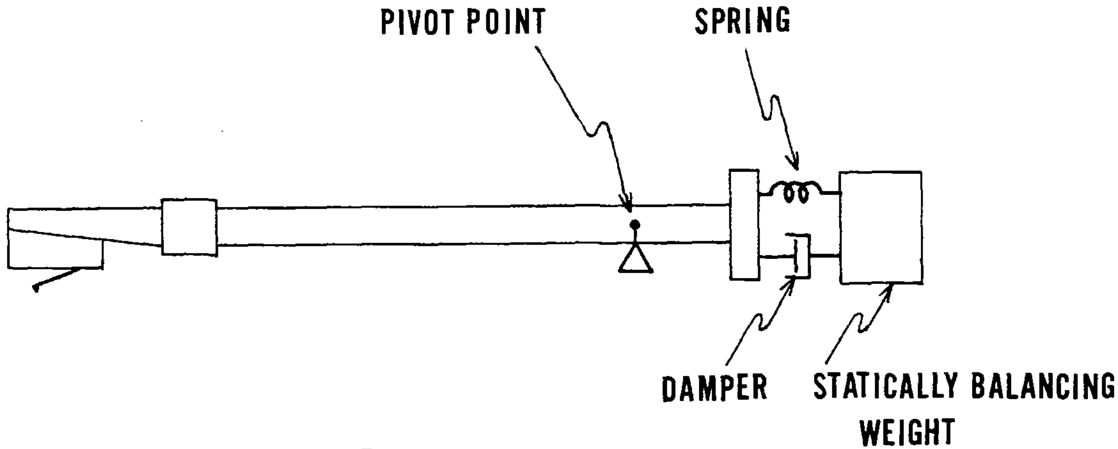


FIG. 6

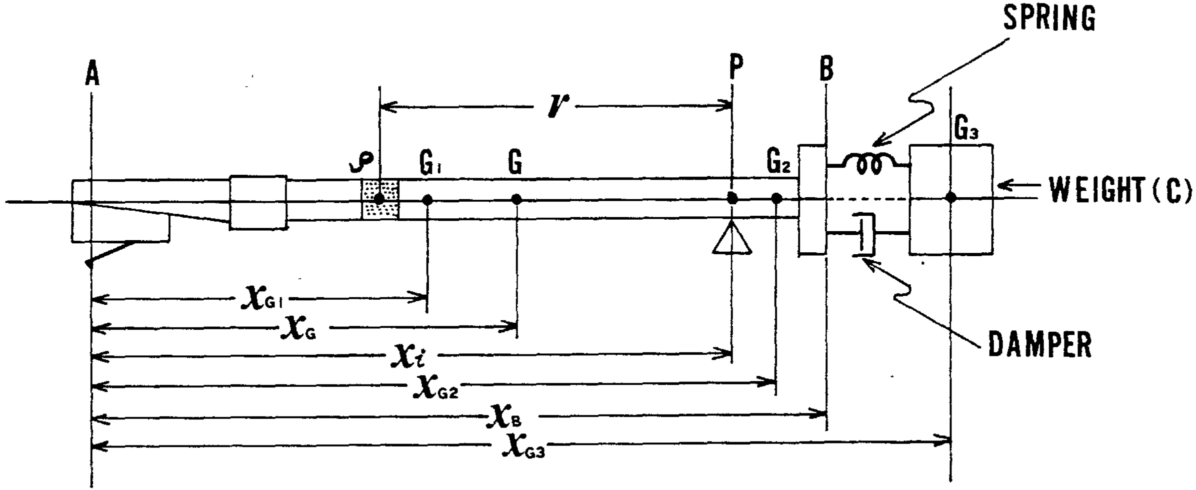
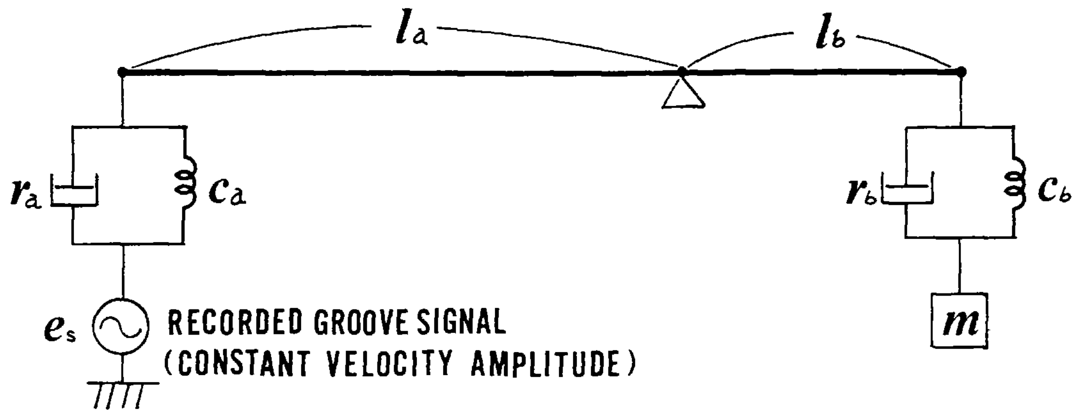


FIG. 7



l_a
l_b
r_a
c_a
r_b
c_b
e_s
RECORDED GROOVE SIGNAL
(CONSTANT VELOCITY AMPLITUDE)
m

FIG. 8

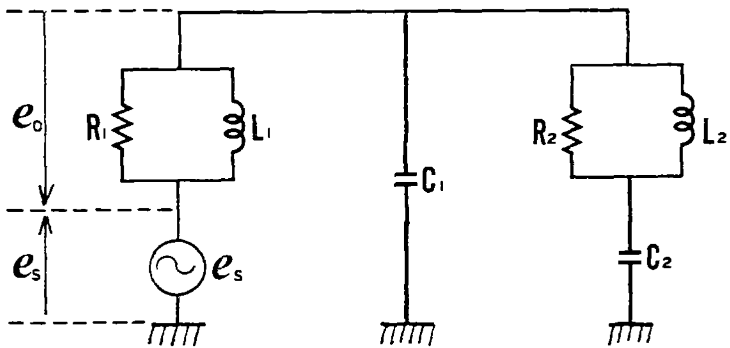


FIG. 9

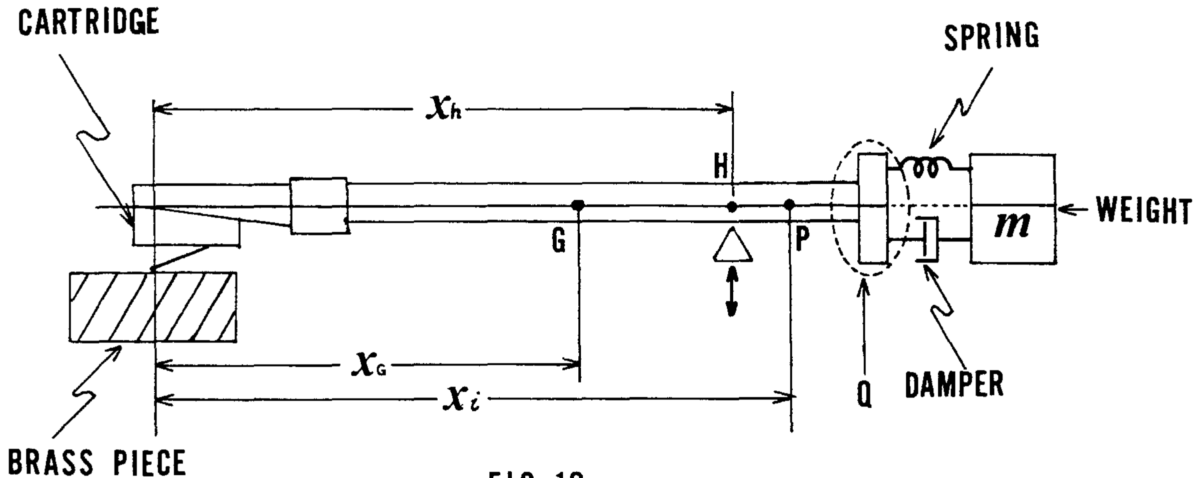


FIG. 10

No.	m_T (g)	I_G (gcm^2)	X_G (cm)	X_i (cm)	X_h (cm)	ΔX (cm)	I_h (gcm^2)
1	262.5	14692	22.7	25.2	23.3	1.9	14787
2	206.2	12995	21.8	24.7	23.3	1.4	13459
3	180.0	11784	20.9	24.0	23.3	0.7	12821
4	169.4	11731	20.4	23.8	23.3	0.5	13156
5	155.2	10036	19.3	22.7	23.3	0.6	12519

TABLE 1

$$\Delta X = |X_i - X_h|$$

$$I_h = I_G + m_T (X_h - X_G)^2$$

FIG.11 TONEARM No.1

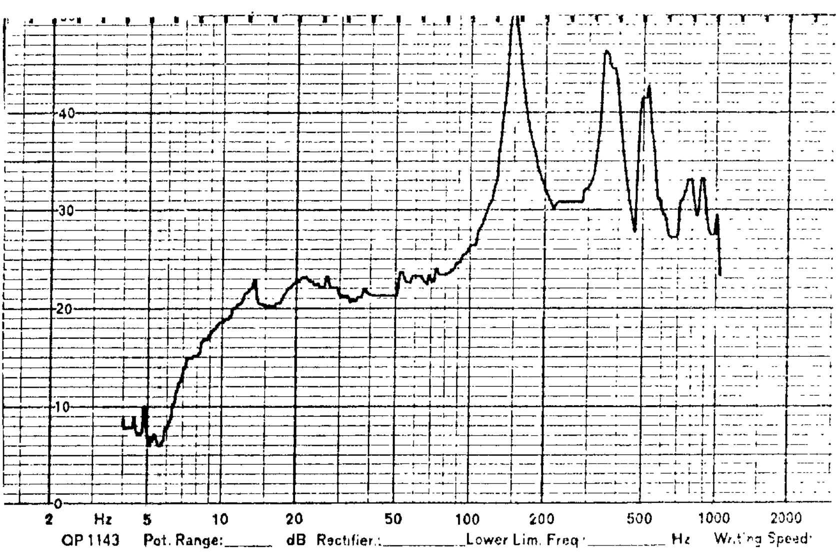


FIG.12 TONEARM No.2

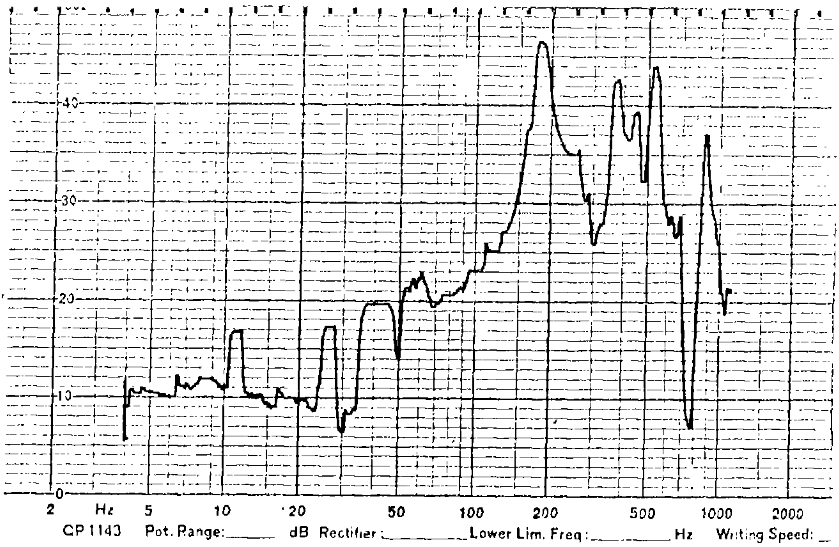


FIG.13 TONEARM No.3

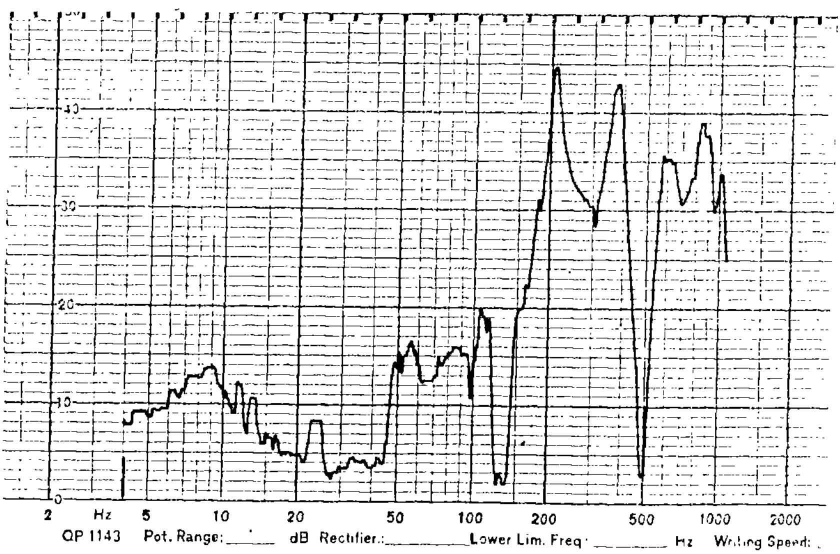


FIG.14 TONEARM No.4

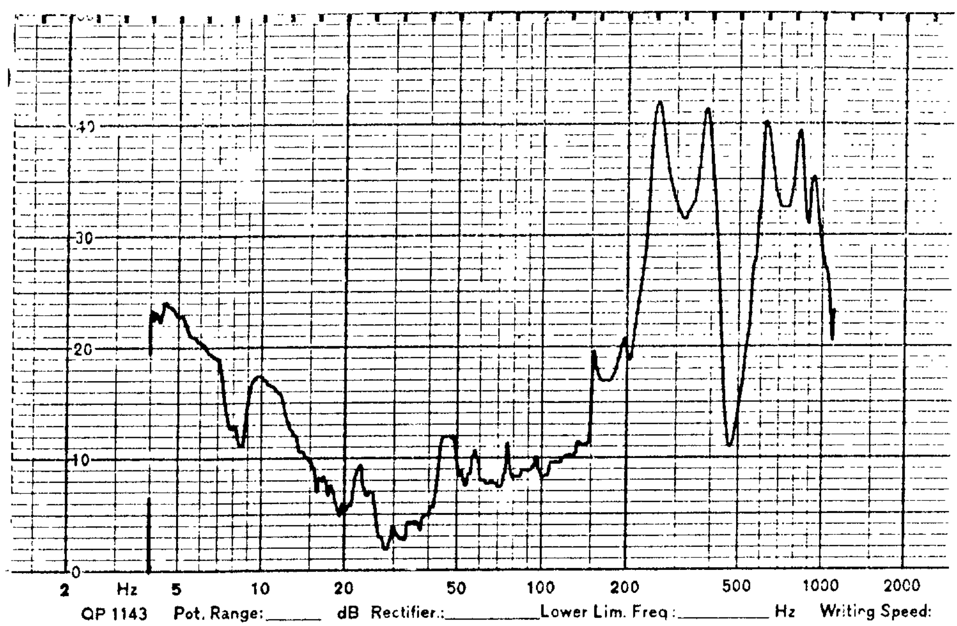


FIG.15 **TONEARM No.5**

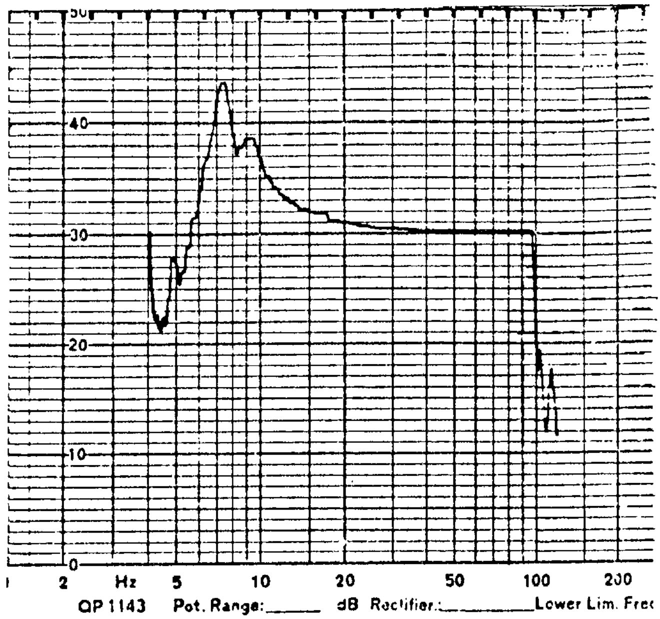


FIG.16 TONEARM No.1

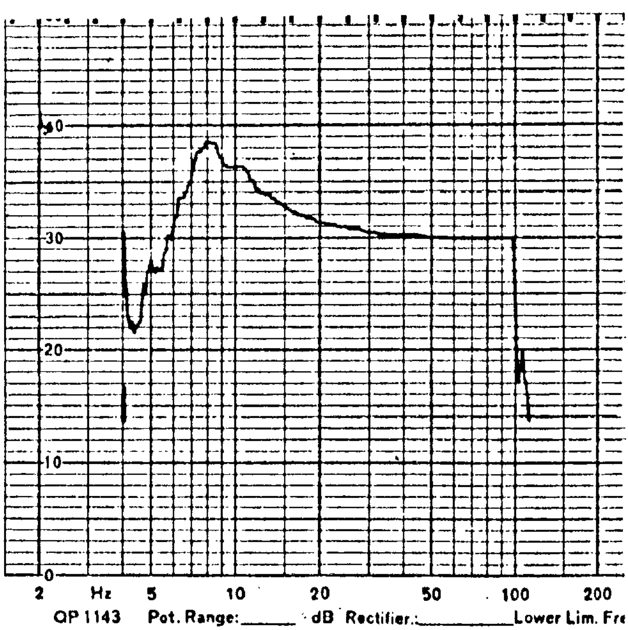


FIG.17 TONEARM No.4

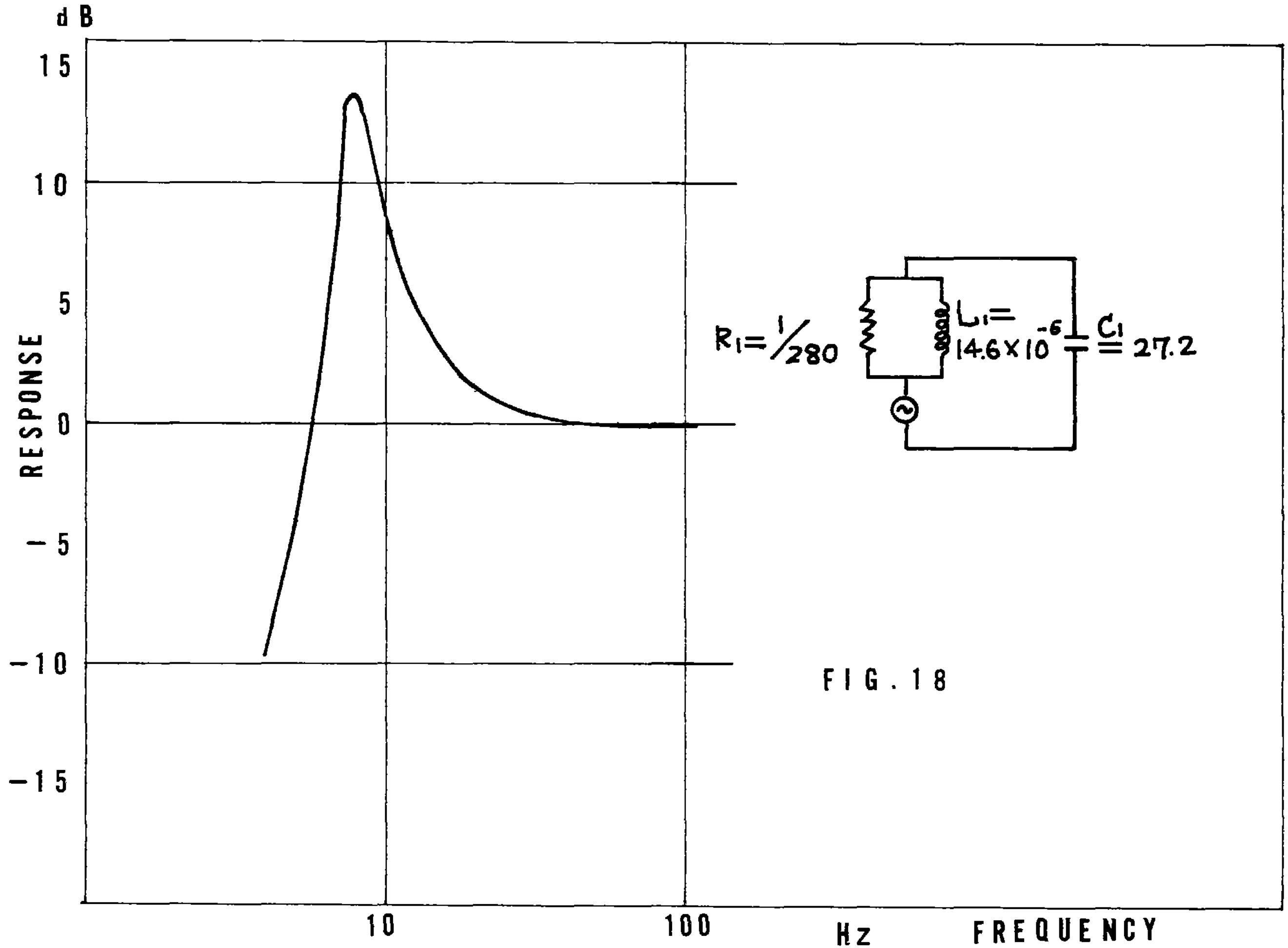

dB
15
10
5
0
− 5
−10
−15
RESPONSE
10
100
Hz
FREQUENCY
R1=1/280
L1=
14.6×10^-6
C1
=27.2

FIG. 18

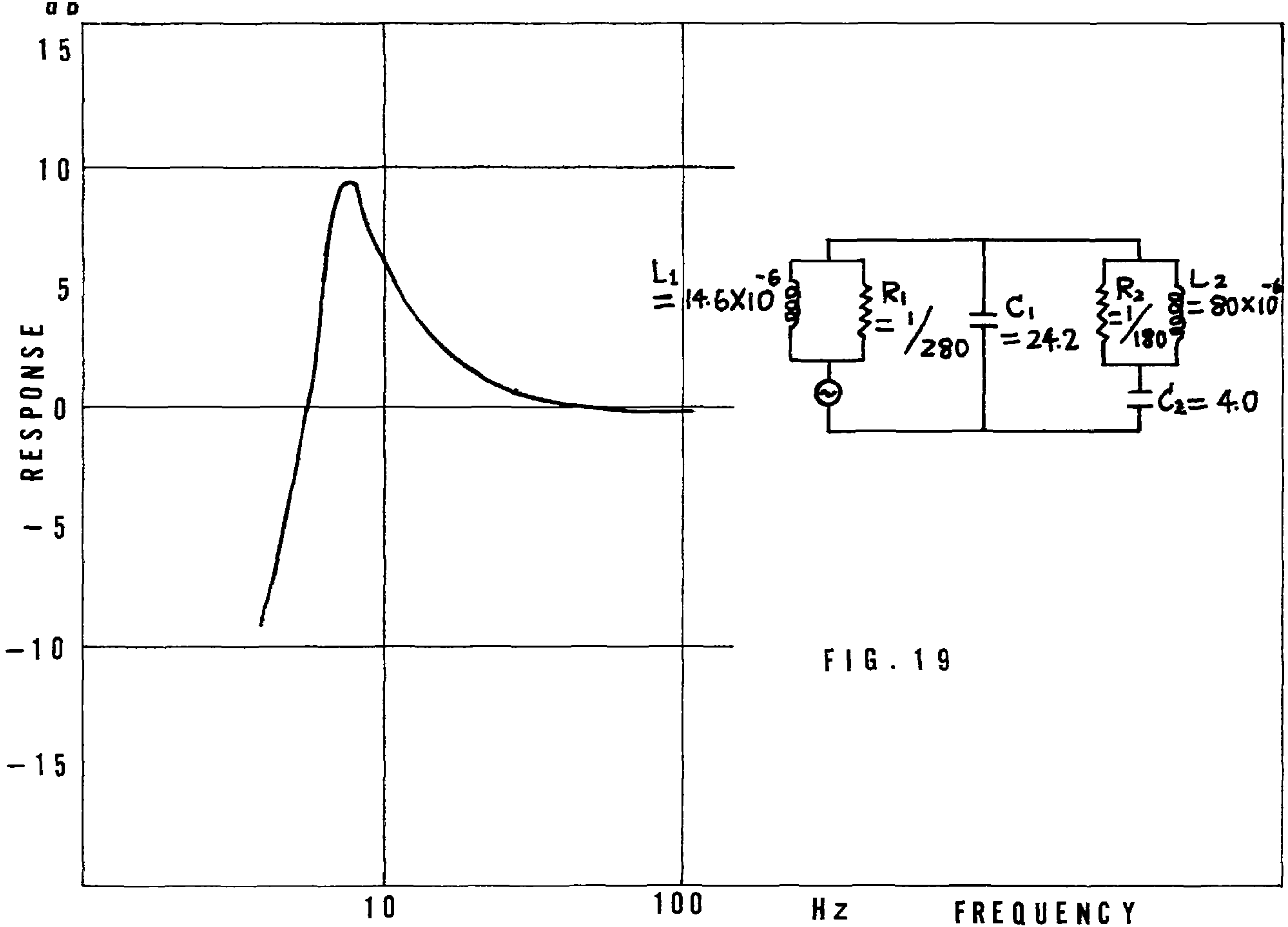

dB
15
10
5
0
-5
-10
-15
RESPONSE
10
100
Hz
FREQUENCY
L1 = 14.6X10-6
R1 = 1/280
C1 = 24.2
R2 = 1/180
L2 = 80X10-6
C2 = 4.0

FIG. 19

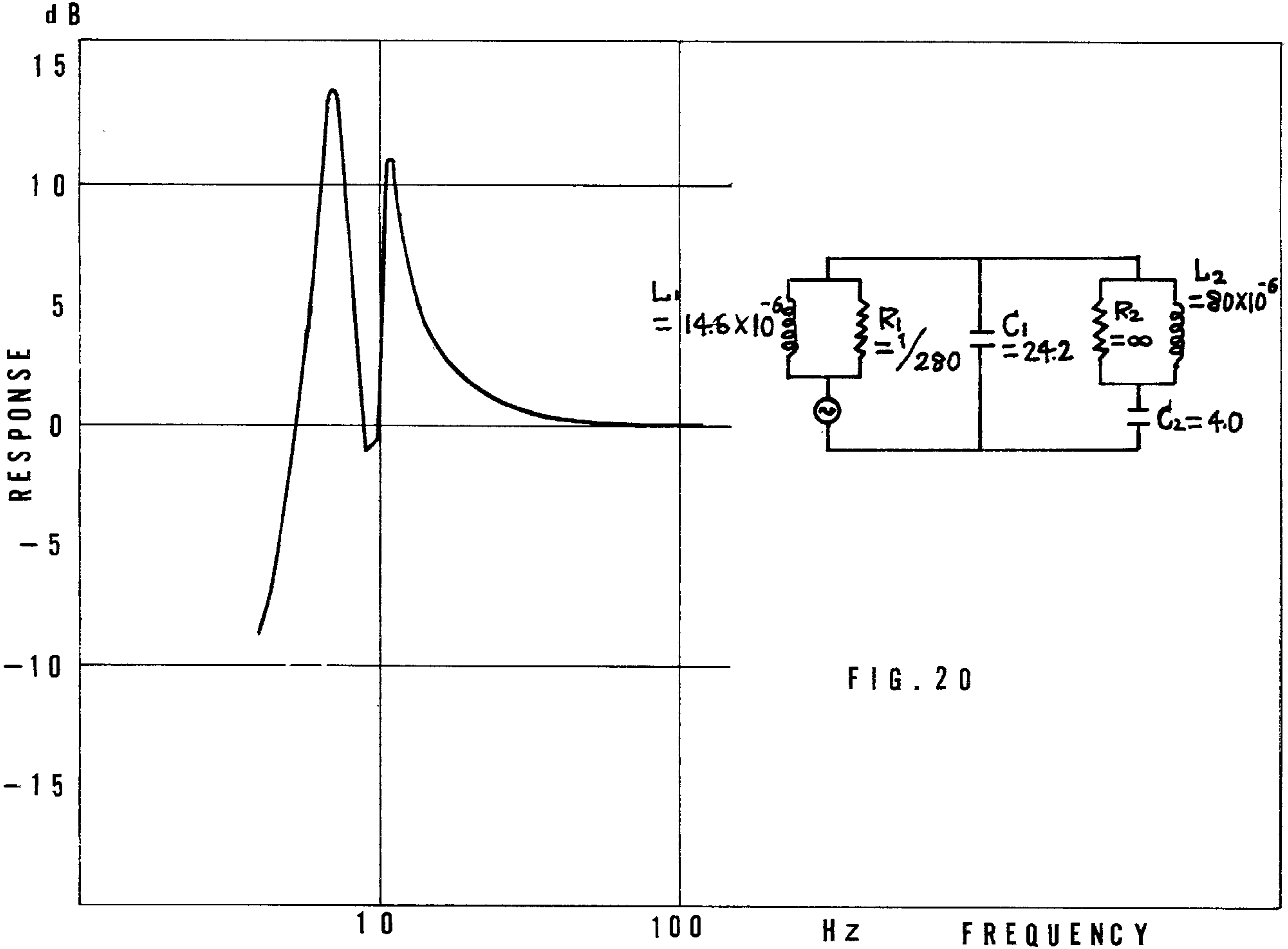
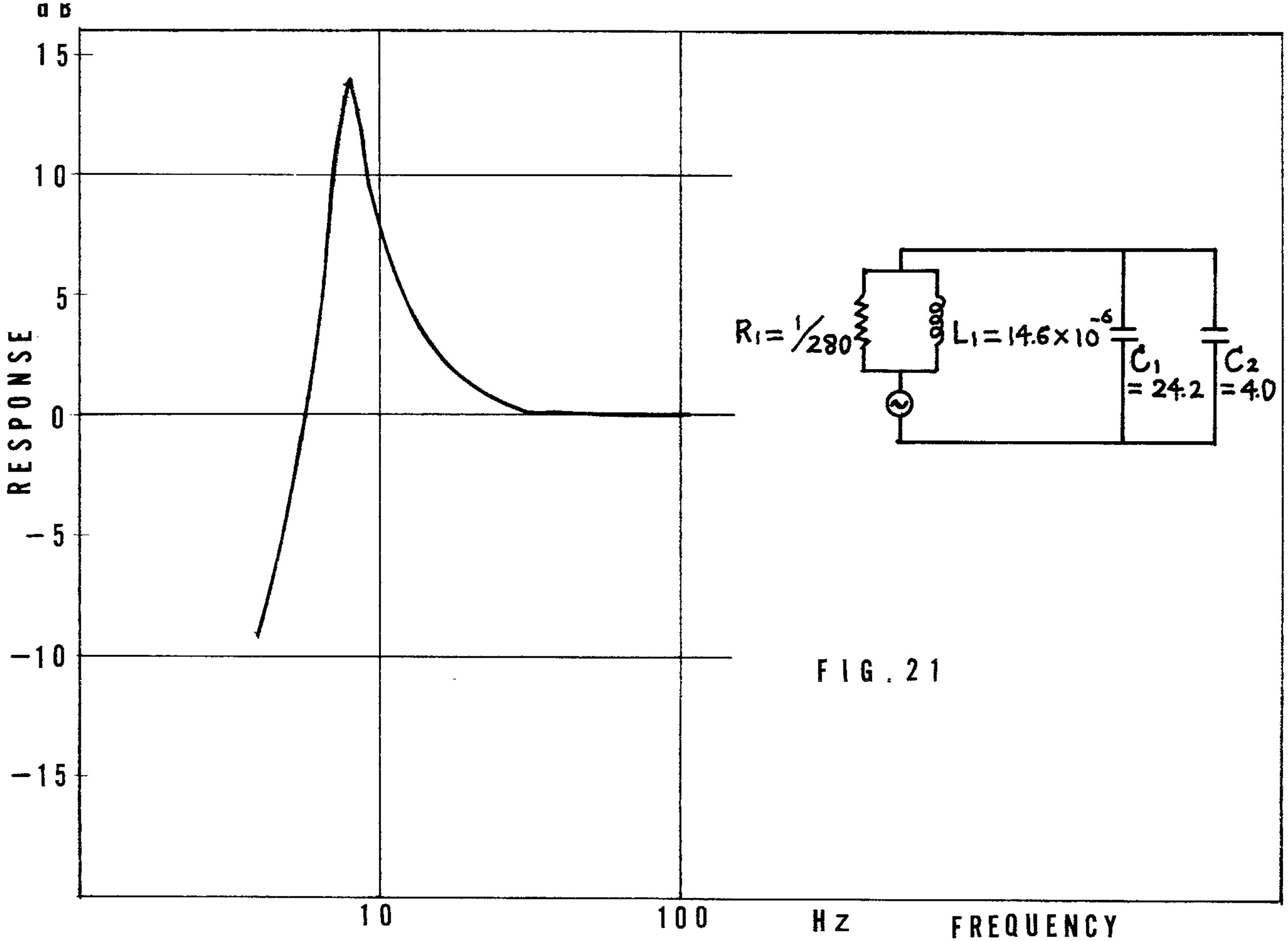

dB
15
10
5
0
−5
−10
−15
RESPONSE
10
100
Hz
FREQUENCY
L1 = 14.6×10⁻⁶
R1 = 1/280
C1 = 24.2
R2 = ∞
L2 = 80×10⁻⁶
C2 = 4.0

FIG. 20



dB
15
10
5
0
-5
-10
-15
RESPONSE
10
100
Hz
FREQUENCY
$R_1 = 1/280$
$L_1 = 14.6 \times 10^{-6}$
$C_1 = 24.2$
$C_2 = 4.0$

FIG. 21

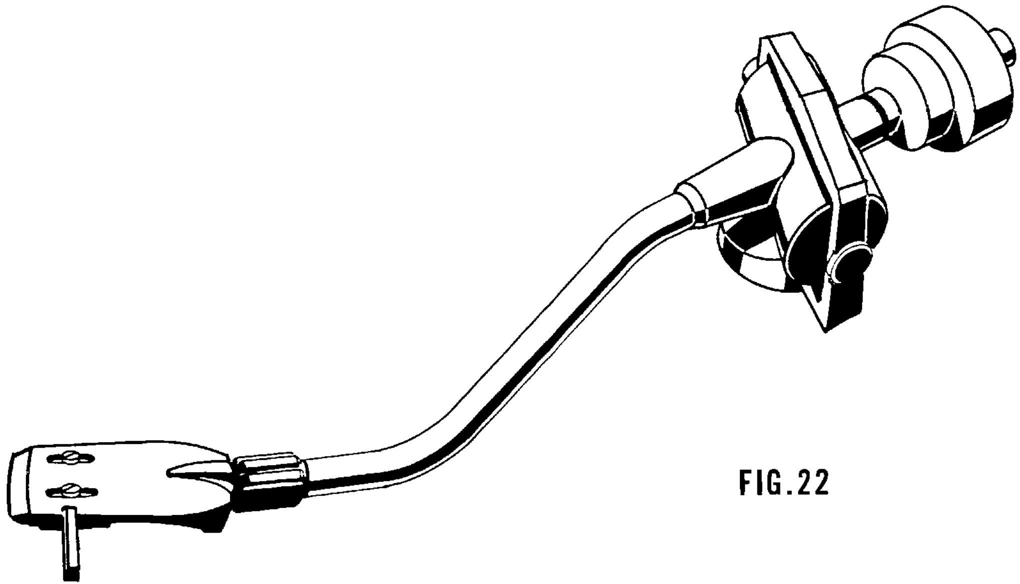


FIG.22