Force-Domain Considerations in Vibration Isolation for Analog Disc Playback

Authors:

Mark Doehmann, George Moraitis and Dallas Clarke

Döhmann Audio Pty Ltd

Email: mark@dohmannaudio.com

ABSTRACT

Vinyl playback spans extreme mechanical scales-from macroscopic structural vibrations to nanometer-scale groove modulations-where minute force and velocity perturbations affect cartridge output. Traditional turntable isolation relies on displacement attenuation metrics, yet at nanometer levels, the stylus-groove interface is governed by force-induced velocity modulation rather than displacement.

This paper proposes a force-domain analytical framework to evaluate vibration transmission in analog disc playback, emphasizing interactions among low-frequency structural motion, bearing noise, drive control, and stylus transduction. Nanometer groove modulation is analyzed linearly below unity depth for small-signal superposition. Low-frequency velocity modulation and sidebands are related to rumble, wow, and flutter measurements.

Findings reveal shortcomings of displacement-centric metrics in high-end analog replay and demonstrate that force-domain isolation more effectively reduces stylus reaction forces and low-frequency modulation without depending solely on ultra-low natural frequencies. Record warp, eccentricity, vacuum hold-down, and isolation-drive interactions are discussed regarding sub-audible modulation.

The framework offers a physically grounded approach to interpreting nanometer disturbances and assessing isolation architectures beyond conventional displacement-based methods, without specifying audibility thresholds.

Keywords: vibration isolation, force-domain analysis, analog disc playback, stylus transduction, low-frequency modulation, nanometer-scale groove, turntable architecture

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1. Introduction

1.1 Background and Motivation

Analog disc replay represents one of the most mechanically demanding signal recovery problems in audio engineering. Musical information is encoded as extremely small geometric variations in vinyl groove walls, which must be recovered by a mechanical transducer operating under substantial static load. Unlike digital playback systems, where signal integrity is primarily governed by clock stability and noise shaping, vinyl playback is fundamentally constrained by mechanical interactions occurring at vastly different spatial and temporal scales.

Modern vibration isolation systems used in turntable design are frequently derived from technologies developed for inertial instrumentation, optical metrology, or semiconductor fabrication. These systems are typically characterized using displacement transmissibility metrics, which describe the relative motion between a payload and its supporting structure. While such metrics are appropriate for position-sensitive instruments, their relevance to force- and velocity-sensitive transducers such as phonograph cartridges has not been rigorously examined.

This paper addresses that gap by re-examining vibration isolation for analog disc playback from a force-domain perspective. The analysis demonstrates that isolation strategies optimised for minimal displacement do not necessarily minimise the mechanical force variations experienced at the stylus-groove interface, and that these force variations can produce measurable signal consequences even when platform motion is negligible.

In this paper, a ‘force-domain’ criterion refers to minimizing low-frequency reaction force variation transmitted into the tonearm/cartridge support and hence the stylus force perturbation $\Delta F(t)$, rather than minimizing platform displacement alone.

1.2 Scope and Assumptions

The discussion in this paper is intentionally limited in scope. It concerns only conventional lateral-cut vinyl records replayed using moving-coil or moving-magnet cartridges operating within typical tracking-force ranges. The analysis assumes linear elastic behavior of vinyl within normal operating limits and does not address record wear, damage, or long-term plastic deformation.

Electrical aspects of cartridge and phono preamplifier design are considered only insofar as they define practical noise floors and bandwidth constraints. Psychoacoustic thresholds are not re-derived; instead, established modulation and signal-to-noise concepts are used to assess detectability. No claims are made regarding listener preference or subjective superiority of any specific implementation.

Where manufacturer-published vibration isolation data are referenced, they are used solely for illustrative comparison of architectural behavior. No attempt is made to validate or dispute published specifications.

1.3 The Nanometer-Scale Constraint in Vinyl Replay

The fundamental challenge of vinyl playback arises from the scale mismatch between the encoded signal and the surrounding mechanical environment. Groove wall modulation corresponding to audible musical content typically occurs at displacement amplitudes on the order of a few nanometers in the mid-band. At the standard reference velocity of 5 cm/s RMS at 1 kHz, the corresponding groove displacement is approximately 11 µm. Nanometer-scale groove displacements occur for low-level program content (e.g, ≈7 nm corresponds to ≈ -64 dB relative to the 5cm/s RMS reference (e.g RMS lateral velocity)).

A formal derivation of the displacement equivalence used here is provided in Appendix A.

By contrast, structural vibrations originating from building motion, footfall, or airborne excitation commonly occur at displacement amplitudes several orders of magnitude larger, particularly at low frequencies below 20 Hz. While such disturbances may be far below audibility when considered as direct acoustic signals, their interaction with the stylus-cartridge system is governed not by absolute displacement alone, but by the forces transmitted through compliant mechanical interfaces.

This scale disparity motivates a reassessment of how vibration isolation performance is evaluated for analog disc playback systems. In the following sections, the stylus-groove interface is examined as a force-biased mechanical system, and the consequences of low-frequency force modulation are analysed in detail.

2. Physical Limits of the Stylus-Groove Interface

2.1 Groove Geometry and Modulation Scale

In lateral-cut vinyl records, musical information is encoded as geometric modulation of the groove walls. For a sinusoidal tone of frequency f and peak velocity v_{peak}, the corresponding peak groove displacement x_{pk} is given by:

$$x_{\mathrm{pk}} = \frac{v_{\mathrm{peak}}}{2\pi f} \tag{1}$$

Using the standard reference velocity of 5 cm/s RMS at 1 kHz, the resulting peak groove displacement is approximately 11 µm (exact 11.25 µm). For low-level signals, displacement scales linearly with velocity: therefore, a peak displacement of 7 nm corresponds to approximately -64 dB re 5 cm/s RMS at 1 kHz (e.g RMS lateral velocity).

This displacement scale is representative of mid-band musical content and decreases further at higher frequencies.

These values place the encoded signal well below the scale of typical mechanical disturbances encountered in domestic environments, particularly at low frequencies. As a result, any mechanical system involved in vinyl playback must preserve nanometer-scale modulation integrity in the presence of significantly larger structural motions.

Displacement transmissibility alone does not uniquely determine the mechanical impedance seen at the stylus interface. The validity of this linear treatment for nanometer-scale displacement and force perturbations, operating well below unity modulation depth, is addressed explicitly in the Appendix. Throughout this paper, all force and displacement perturbations are treated within the small-signal regime, where linear superposition remains valid (see Appendix).

2.2 Stylus Contact Mechanics

The stylus-groove interface constitutes a highly localized elastic contact between the diamond stylus tip and the vinyl groove wall. Modern stylus profiles typically produce a contact patch on the order of one to several square micrometers, depending on stylus geometry, groove curvature, and instantaneous load conditions.

Under nominal tracking forces of approximately 1.5–2.5 g (≈15–25 mN), the resulting contact pressures are in the gigapascal range due to the small contact area. Despite these high local pressures, experimental and analytical studies indicate that vinyl deformation during normal playback remains predominantly elastic, provided tracking force and groove velocity remain within standard operating limits. [1][2]

Factors affecting the stylus–groove relationship—including load distribution, compliance, and geometric alignment—have been shown to influence playback fidelity. [3]

The stylus does not respond directly to groove displacement alone, but to the local force distribution and resulting relative motion between stylus and groove wall. The cantilever and suspension assembly provide finite compliance, so the instantaneous stylus motion reflects the combined effects of groove geometry, suspension stiffness, damping, and applied load. Classical analyses of pickup design treat this interaction as a force-biased mechanical system operating about a static equilibrium point. [4]

Because the system operates under a substantial static tracking-force bias, small dynamic perturbations in force or acceleration can alter the instantaneous equilibrium condition of the stylus–cantilever assembly. In the small-signal regime considered throughout this paper, these perturbations are treated using linearized incremental stiffness about the DC operating point. Under this assumption, stylus motion remains governed by elastic compliance and velocity sensitivity, without requiring loss of contact or plastic deformation.

The present analysis builds on this framework by examining how low-frequency force variations superimposed on the static tracking force may influence the dynamic response of the stylus–cantilever system.

2.3 Tracking Force as a DC Bias Condition

During playback, the stylus is subjected to a constant downward force set by the tracking force adjustment of the tonearm. This static tracking force F_0 may be expressed as:

$$F_0 = mg \tag{2}$$

where $\boldsymbol{m}$ is the effective tracking mass and $\boldsymbol{g}$ is the acceleration due to gravity.

For a tracking force of 2 g, the corresponding static force F_0 is approximately 19.62 mN. This force is continuously applied and establishes the operating point of the stylus-cantilever system. In this sense, the tracking force functions as a DC bias condition about which all dynamic motion occurs, a concept central to early analyses of pickup design.[4]

Musical information is not encoded by large excursions relative to this bias, but by extremely small variations in force and velocity superimposed upon it. These variations are typically several orders of magnitude smaller than F_0, yet they are responsible for the entire audio signal recovered by the cartridge.

2.4 Force Versus Displacement as Governing Variables

It is common to describe vibration isolation performance in terms of displacement transmissibility. However, for the stylus-groove interface, displacement alone does not fully characterize the mechanical state of the system. The cartridge responds primarily to velocity and force variations at the stylus tip, not to absolute position.

Because the stylus operates under a large static force bias, small changes in applied force can alter its dynamic response even when displacement remains minimal. In a compliant mechanical system, force modulation may arise from changes in acceleration, stiffness, or reaction forces transmitted through supporting structures.

This distinction is critical: a support system may exhibit low displacement transmissibility while still injecting low-frequency force variations into the tonearm-cartridge assembly. Such force variations are not necessarily evident in platform motion measurements but can directly perturb the stylus operating point.

2.5 Low-Frequency Structural Excitation

Structural vibrations in domestic environments are dominated by low-frequency components. Common domestic disturbances often include substantial energy below ~20 Hz. Sources include building sway, footfall, and long-wavelength floor resonances. These excitations often overlap the fundamental resonance of the tonearm-cartridge system, which commonly lies in the 5-12 Hz range.

When low-frequency excitation coincides with this resonance region, even modest accelerations can produce force variations at the stylus through inertial coupling. Because these forces act on a system already biased near its elastic limits, they have the potential to modulate the stylus response to groove wall motion.

The implications of this interaction are examined in the following section, where low-frequency force modulation is shown to produce mid-band modulation products through established modulation mechanisms.

3. Low-Frequency Force Modulation and Signal Consequences

3.1 Inertial Force Perturbation About the Tracking-Force Bias

During playback, the stylus operates under a static tracking force F_0, which establishes the DC operating point of the cantilever–suspension system (Section 2.3). When the turntable–tonearm assembly is subjected to low-frequency base acceleration $a(t)$, inertial forces arise in the effective moving mass $m_{\text{eff}}$of the tonearm–cartridge system:

$$\Delta F(t) = m_{\text{eff}} a(t) \tag{3}$$

The instantaneous stylus force therefore becomes:

$$F(t) = F_0 + \Delta F(t) \tag{4}$$

For sinusoidal low-frequency excitation at modulation frequency f_m:

$$F(t) = F_0 + \Delta F \sin(2\pi f_m t) \tag{5}$$

where $\Delta F \ll F_0$ under normal operating conditions.

When the turntable–tonearm–cartridge system is subjected to low-frequency base excitation, inertial forces arise within the effective moving mass of the tonearm and cartridge assembly. These forces act in addition to the static tracking force defined in Section 2.3 and may be treated as incremental perturbations about the DC bias condition.

Low-frequency excitation commonly overlaps the tonearm–cartridge resonance (typically 5–12 Hz). Near resonance, base acceleration may be amplified by the mechanical transfer function of the arm–cartridge system, increasing ΔFwithout requiring large platform displacement.

Record-induced geometric disturbances provide an additional source of low-frequency force variation. Disc warps (≈0.5–5 Hz at 33⅓ rpm) and record eccentricity introduce cyclic acceleration of the effective mass as the stylus traverses geometric deviations. [5] These effects likewise modulate stylus force about the static bias F_0..[5]

In all cases considered here, $\Delta F / F_0 \ll 1$, and the stylus remains in continuous contact with the groove. The perturbation is therefore treated as incremental loading about a biased equilibrium condition.

3.2 Amplitude Modulation of Groove-Derived Velocity

The electrical output of a phonograph cartridge is proportional to relative stylus–groove velocity. Because the cantilever suspension exhibits finite compliance and weak load dependence, incremental force perturbation alters the operating point of the stylus–cantilever system.

Linearizing about the DC bias F_0, the recovered velocity may be expressed to first order as:

$$v_{out}(t) \approx v_g(t)\,[1 + k_F \Delta F(t)] \tag{6}$$

where:

• $v_g(t)$is the groove-derived velocity component
• $k_F$is the incremental force-to-velocity sensitivity coefficient (units: N^{-1})
• $\Delta F(t) \ll F_0$

For sinusoidal perturbation:

$$\Delta F(t) = \Delta F \sin(2\pi f_m t) \tag{7}$$

the output becomes:

$$v_{out}(t) \approx A[1 + m \sin(2\pi f_m t)] \sin(2\pi f_c t) \tag{8}$$

where:

$$m = k_F \Delta F \tag{9}$$

is the amplitude modulation index.

In the small-signal regime $m \ll 1$, this produces first-order sidebands at $f_c \pm f_m$with amplitude approximately $m/2$relative to the carrier, consistent with classical AM analysis and phonograph intermodulation theory.[6]

3.3 Order-of-Magnitude Scaling

To illustrate the scale of the mechanism, assume:

Effective mass:

$$m_{\mathrm{eff}} = 15\ \mathrm{g} = 0.015\ \mathrm{kg} \tag{10}$$

Nominal tracking force:

$$F_0 \approx 20\ \mathrm{mN} \tag{11}$$

Low-frequency base acceleration produces incremental stylus force:

$$\Delta F = m_{\mathrm{eff}} a \tag{12}$$

In Section 3.2, the AM modulation index is defined as:

$$m = k_F \Delta F \tag{13}$$

In the small-signal AM regime ($m \ll 1$), the amplitude of each first-order sideband is approximately:

$$\frac{m}{2} \tag{14}$$

relative to the carrier. Therefore, the expected first-order sideband level is:

$$\Delta L \approx 20\log_{10}\left(\frac{m}{2}\right)\ \text{dBc} \tag{15}$$

Case A: 1 mg Base Acceleration

$$a = 1\ \text{mg} = 0.00981\ \text{m/s}^2 \tag{16}$$

$$\Delta F = 0.015 \times 0.00981 = 1.47 \times 10^{-4}\ \text{N} = 0.147\ \text{mN} \tag{17}$$

$$\frac{\Delta F}{F_0} = \frac{0.147}{20} \approx 0.00735\ (0.735\%) \tag{18}$$

If a conservative upper-bound sensitivity $k_F F_0 \sim 1$is assumed (i.e., modulation index comparable to the fractional force perturbation), then:

$$m \sim \frac{\Delta F}{F_0} \approx 0.00735 \tag{19}$$

$$\Delta L \approx 20\log_{10}\left(\frac{0.00735}{2}\right) = 20\log_{10}(0.003675) \approx -48.7\ \text{dBc} \tag{20}$$

Under a more realistic small-signal sensitivity $k_F F_0 = 0.05$:

$$m \approx 0.05 \times 0.00735 = 3.68 \times 10^{-4} \tag{21}$$

$$\Delta L \approx 20\log_{10}\left(\frac{3.68 \times 10^{-4}}{2}\right) = 20\log_{10}(1.84 \times 10^{-4}) \approx -74.7\ \text{dBc} \tag{22}$$

Case B: 0.1 mg Base Acceleration

$$a = 0.1\ \text{mg} = 0.000981\ \text{m/s}^2 \tag{23}$$

$$\Delta F = 0.015 \times 0.000981 = 1.47 \times 10^{-5}\ \text{N} = 0.0147\ \text{mN} \tag{24}$$

$$\frac{\Delta F}{F_0} = \frac{0.0147}{20} \approx 0.000735\ (0.0735\%) \tag{25}$$

Upper-bound sensitivity $k_F F_0 \sim 1$:

$$m \sim 0.000735 \tag{26}$$

$$\Delta L \approx 20\log_{10}\left(\frac{0.000735}{2}\right) = 20\log_{10}(3.675 \times 10^{-4}) \approx -68.7\ \text{dBc} \tag{27}$$

Realistic sensitivity $k_F F_0 = 0.05$:

$$m \approx 0.05 \times 0.000735 = 3.68 \times 10^{-5} \tag{28}$$

$$\Delta L \approx 20\log_{10}\left(\frac{3.68 \times 10^{-5}}{2}\right) = 20\log_{10}(1.84 \times 10^{-5}) \approx -94.7\ \text{dBc} \tag{29}$$

These examples demonstrate:

- Linear scaling of $\Delta F$with base acceleration ($\Delta F = m_{\text{eff}} a$)

- Linear scaling of modulation index with force perturbation ($m = k_F \Delta F$)
- First-order sideband levels scaling as $20\log_{10}(m/2)$in the AM small-signal regime
- Sub-milligravity excitation remains comfortably within the linear regime ($m \ll 1$) and does not require loss of stylus–groove contact

For context, low-frequency building vibrations in domestic environments commonly range from fractions of a milligravity to several milligravities depending on structure and excitation conditions.

3.4 Spectral Consequences and Detectability Considerations

In addition to amplitude modulation, incremental force variation can alter effective suspension stiffness and damping, producing small perturbations in instantaneous stylus velocity relative to groove geometry. This corresponds to phase or frequency modulation of the recovered signal.

In the small-signal regime, first-order FM/PM sidebands scale linearly with modulation index (via $J_1(\beta) \approx \beta/2$for $\beta \ll 1$), producing sideband components around the carrier frequency without requiring large displacement (see classical small-signal FM theory [6]).

Thus, both amplitude and phase modulation arise naturally from incremental perturbation of a force-biased mechanical system operating within linear elastic limits.[7]

3.5 Implications for Vibration Isolation Evaluation

The foregoing analysis demonstrates that low-frequency force modulation constitutes a plausible mechanism by which structural vibration may influence recovered audio signal content.

Evaluation methods based solely on displacement transmissibility do not fully characterize this pathway, because they quantify relative motion rather than dynamic force exchange within the system. An isolation architecture may exhibit low displacement transmissibility while still permitting low-frequency force perturbations at the stylus through inertial and structural coupling.

For playback systems operating near nanometer-scale resolution, assessment of isolation performance therefore benefits from consideration of both kinematic (displacement-based) and dynamic (force-based) behavior in the low-frequency regime identified above.

4. Electrical Noise, Rumble, and Masking Considerations

4.1 Velocity Reference and Signal Scaling

The output of a phonograph cartridge is conventionally specified with reference to a groove velocity of 5 cm/s RMS at 1 kHz. As shown in Section 2.1, the corresponding peak groove displacement at this reference level is on the order of 10 µm (exact 11.25 µm).

For a sinusoidal tone of frequency f and peak velocity v_{peak}, the corresponding peak groove displacement x_{pk} is given by:

$$x_{\text{pk}} = \frac{v_{\text{peak}}}{2\pi f} \tag{30}$$

This relationship highlights an important characteristic of vinyl playback: for a constant velocity reference, groove displacement decreases inversely with frequency. As a result, mid-band and high-frequency musical content is encoded at extremely small displacement amplitudes, while remaining well within the velocity

4.2 Cartridge Output at Nanometer-scale

For a cartridge producing an output voltage V_{ref} at the standard reference velocity, the output corresponding to a smaller groove displacement scales linearly with velocity. A mid-band tone encoded at a peak displacement of approximately 7 nm produces an output level roughly 64 dB below reference.

7 nm @ 1 kHz corresponds to 0.00311 cm/s RMS, i.e., -64.1 dB *re* 5 cm/s RMS (this value assumes sinusoidal motion and RMS conversion per IEC velocity reference conventions).

And it yields ~0.187-0.211 µV RMS from a ~0.3-0.34 mV cartridge @ 5 cm/s. (Note: This is a low but valid output and quite typical for some MC cartridge models).

When amplified by a typical phono preamplifier with 60-70 dB of gain, such signals remain within the measurable and reproducible range of modern low-noise electronics. This establishes that nanometer-scale groove modulation is not, in itself, below the electrical resolution of the playback chain.

The limiting factors therefore shift from absolute signal amplitude to the spectral distribution of noise and interference within the relevant bandwidth.

4.3 Noise Metrics and Resolution Bandwidth

Noise specifications for phono preamplifiers and turntables are commonly expressed as integrated RMS values over wide bandwidths. While such figures are useful for general characterisation, they do not directly indicate the detectability of narrowband signals.

The relationship between total integrated noise level L_{total} and noise spectral density L_{density} within a resolution bandwidth B may be expressed as

$$L_{\text{density}} = L_{\text{wideband}} - 10\log_{10}(B) \tag{31}$$

This relationship demonstrates that reducing the analysis bandwidth substantially lowers the effective noise floor against which narrowband signals are evaluated, consistent with fundamental acoustics principles.[8] For example, −60 dB integrated over 1 kHz corresponds to −90 dB/Hz (since 10 log10(1000) = 30 dB). Consequently, the presence of low integrated noise figures does not imply that small tonal components are masked across all frequencies.

4.4 Rumble and Low-Frequency Noise

Mechanical rumble and bearing noise in turntable systems are typically concentrated at low frequencies and are often reported using weighted metrics such as DIN-B weighting. While such weighting schemes correlate reasonably with subjective annoyance, they intentionally suppress low-frequency components that may be relevant to modulation mechanisms.

Low-frequency noise components, even when heavily attenuated by weighting filters, can contribute to force modulation at the stylus through inertial coupling. As discussed in Section 3, such modulation need not be directly audible to influence mid-band signal integrity.

It is therefore important to distinguish between noise that is objectionable as an audible artefact and noise that is capable of perturbing the mechanical operating point of the stylus-cartridge system.

4.5 Implications for Masking Assumptions

The preceding analysis demonstrates that masking arguments based solely on wideband noise or weighted rumble figures are insufficient to dismiss the potential impact of low-frequency disturbances. Detectability depends on local noise density, modulation mechanisms, and the nonlinear response of the stylus-cantilever system.

Accordingly, the absence of clearly audible low-frequency noise does not ensure reduced susceptibility from modulation-induced distortion.[7] This reinforces the need to consider force-domain interactions and low-frequency behavior when evaluating vibration isolation strategies for analog disc playback.

5. Mechanical Noise Sources in Turntable Systems

5.1 Bearing-Related Noise and Vibration

Turntable main bearings inevitably generate mechanical noise due to surface roughness, lubrication behavior, and load variation during rotation. While well-designed bearings can reduce audible artefacts to very low levels, residual vibration is always present and is transmitted into the platter and record interface.

Bearing-related disturbances are typically concentrated at low frequencies and may include both periodic components, associated with rotational irregularities, and broadband components arising from stochastic surface interactions. Although such disturbances may be far below audibility when assessed directly, their proximity to the stylus-groove interface makes them relevant as potential sources of force modulation.

Because the stylus operates under a static force bias, even small variations in bearing reaction forces can influence the effective loading of the record-stylus system through the platter and record support structure.

5.2 Motor-Induced Excitation

Drive motors introduce mechanical excitation through torque variation, structural coupling, and dynamic interaction with the platter-bearing assembly. These excitations may arise from cogging effects, electromagnetic torque ripple, belt compliance (in belt-driven systems), or corrective torque applied in closed-loop servo-controlled systems.

In belt-driven architectures, higher-frequency motor artefacts may be attenuated by belt compliance and mechanical filtering. However, low-frequency components associated with torque variation, belt elasticity, and load-dependent slip can remain coupled to the platter. The magnitude and spectral content of this coupling depend on belt stiffness, damping, and the dynamic impedance of the platter-bearing system.

In direct-drive systems employing closed-loop servo control, rotational speed is maintained through continuous torque correction applied directly to the platter. The servo system monitors speed error and applies corrective torque based on loop gain, bandwidth, and control algorithm. While such systems can achieve excellent steady-state speed accuracy and low wow-and-flutter figures, the control process necessarily involves time-varying torque applied to the platter.

Time-varying torque $\tau(t)$produces reaction forces within the platter-bearing assembly according to:

$$\tau(t) = r \cdot F_t(t) \tag{32}$$

where $r$is the effective radius at which the torque is resolved and $F_t(t)$represents tangential force at that radius. These forces are reacted through the bearing structure and ultimately through the mechanical reference plane supporting the record and tonearm.

From a displacement perspective, such forces may produce negligible observable platform motion. However, from a force-domain perspective, they represent dynamic loading of the platter-bearing structure within frequency bands determined by servo bandwidth and system inertia. The relevance of this loading depends on:

- The spectral content of the torque corrections
- The dynamic stiffness and damping of the bearing-platter assembly
- Mechanical coupling paths to the tonearm base
- Coincidence with tonearm-cartridge resonance (typically 5–12 Hz)

The presence of time-varying corrective torque does not imply instability or audible speed error. Rather, it highlights that speed accuracy alone does not fully describe the mechanical forces present within the system. In architectures where multiple feedback loops are active — for example, motor servo control combined with active vibration isolation — corrective forces from independent control systems may coexist within overlapping frequency bands.

The significance of such interactions cannot be inferred solely from steady-state speed specifications or displacement transmissibility plots. A comprehensive evaluation therefore benefits from consideration of both kinematic performance (speed stability, displacement suppression) and dynamic force interactions within the low-frequency regime identified in Section 3.

When combined with active vibration isolation, these corrective forces form part of a multi-loop control system whose interaction is discussed in Section 6.2.

5.3 Rumble Metrics and Their Limitations

Rumble specifications are commonly reported using weighted metrics, such as DIN-B weighting, which emphasize frequency ranges associated with subjective annoyance. While these metrics provide useful comparative information, they intentionally suppress very low-frequency components that may still be mechanically significant.

As a result, a turntable may exhibit excellent weighted rumble figures while still transmitting low-frequency energy capable of modulating stylus force. This distinction is particularly relevant when evaluating systems intended for high-resolution playback, where subtle modulation effects can influence perceived clarity and stability.

Weighted rumble metrics should therefore be interpreted as indicators of audible noise performance rather than comprehensive measures of mechanical neutrality, as detailed in measurements of turntable performance [9].

5.4 Interaction with Vibration Isolation Systems

Mechanical noise sources within the turntable interact with external vibration isolation systems in complex ways. Isolation platforms may reduce the transmission of environmental vibration while simultaneously altering the dynamic boundary conditions experienced by the turntable itself.

For example, compliant isolation can shift resonance frequencies or modify reaction forces transmitted back into the turntable structure. In some cases, this may reduce audible artefacts while increasing low-frequency force modulation at the stylus.

These interactions highlight the importance of evaluating isolation performance in the context of the complete playback system rather than as an isolated component.

5.5 Summary

Mechanical noise sources in turntable systems are not limited to audible artefacts such as rumble or hum. Low-frequency disturbances arising from bearings, motors, and structural coupling can influence the stylus-groove interface through force modulation mechanisms, even when direct audibility is minimal.

It is important to note that the nanometer-scale perturbations discussed here occur well below unity modulation depth and do not imply loss of stylus-groove contact or audible mistracking. Instead, these effects appear as low-frequency velocity modulation within an otherwise linear operating regime, as detailed in the Appendix.

Understanding these interactions is essential for interpreting isolation performance metrics and for designing systems that preserve signal integrity at nanometer-scales.

6. Vibration Isolation Architectures

6.1 Pneumatic Isolation Systems

Pneumatic isolation systems employ compressed air to support a payload on compliant air springs. The effective stiffness of the support is determined by the volume of air and the compliance of the air bladder, resulting in low natural frequencies typically in the range of 1-3 Hz.

Such systems are effective at attenuating higher-frequency vibration and are widely used in laboratory and industrial environments. However, pneumatic isolation exhibits limited damping at very low frequencies and may allow significant motion near the resonance of the supported mass.

Additionally, because air springs store energy elastically, low-frequency excitation can result in phase lag and reactive forces transmitted to the payload. While these effects may be inconsequential for inertial instruments, they are relevant for force-sensitive systems where reaction forces can influence operating conditions at the transducer.

6.2 Active Feedback Isolation Systems

Active isolation systems employ sensors, actuators, and control algorithms to reduce motion of a payload relative to an inertial reference. These systems can achieve very low apparent stiffness and excellent displacement suppression across a broad frequency range.

By design, active systems inject corrective forces into the payload in response to detected motion. The control bandwidth typically spans the low-frequency region where building-borne vibration is most pronounced. As a result, active isolation can substantially reduce platform displacement within the critical 5-10 Hz band.

However, because the system operates by applying forces to the payload, it necessarily introduces force interactions that are not directly visible in displacement-based performance metrics. For applications involving force- or velocity-sensitive transducers, the presence of active force injection within the operating bandwidth may warrant additional consideration.

As with active vibration isolation, closed-loop motor control systems apply corrective forces to the payload, and their influence on force-sensitive interfaces depends on the spectral content and timing of those corrections rather than on steady-state accuracy alone.

When active vibration isolation is employed in conjunction with a turntable incorporating closed-loop motor control, the overall system comprises multiple independent feedback loops acting on the same mechanical structure. Each loop operates based on locally sensed variables and applies corrective forces without awareness of the other.

If the control bandwidths of the isolation system and the motor servo overlap, particularly in the low-frequency region below approximately 10 Hz, uncoordinated force injection may occur. While neither system is unstable in isolation, their combined action can result in time-varying reaction forces that are not directly apparent in displacement measurements or steady-state performance specifications.

From a force-domain perspective, such interaction represents an additional pathway by which low-frequency excitation may be introduced at the stylus-groove interface. The magnitude and significance of this effect depend on control implementation, loop gain, phase margin, and mechanical coupling, and therefore cannot be inferred from the behavior of either subsystem alone.

The distinction is not that passive systems are force-free, but that active systems can introduce control-correlated spectral components of force within the band of interest, whose magnitude depends on loop design rather than passive impedance alone.

6.3 Passive Negative-Stiffness Isolation Systems

Negative-stiffness mechanisms employ geometric stiffness cancellation to achieve low effective vertical stiffness while maintaining structural stability. Unlike active systems, they do not require electronic feedback or continuous energy input, and unlike pneumatic systems, they do not depend on compressed air reservoirs. Their effectiveness depends on mechanical tuning, damping, and load configuration.

Because negative-stiffness systems do not require active control or compressed air, they avoid both feedback-induced force injection and pneumatic energy storage. Their isolation performance is determined by mechanical geometry and load conditions rather than electronic control parameters.

As with all passive systems, negative-stiffness isolation exhibits finite transmissibility at very low frequencies. The distinction is not the absence of elastic energy storage (common to all passive compliant systems), but the absence of active control forces and the avoidance of pneumatic reservoir dynamics within the band of interest, as explored in vibration isolation control literature [10].

6.4 Comparative Considerations

Each isolation architecture represents a different trade-off between displacement reduction, force interaction, complexity, and stability. Displacement transmissibility curves provide useful information about motion suppression but do not fully characterize the forces transmitted through the supported system.

For applications in which the payload includes force-sensitive mechanical interfaces, such as the stylus-groove contact in vinyl playback, the distinction between displacement suppression and force interaction becomes particularly significant.

The following section examines manufacturer-published isolation performance data in light of these considerations.

7. Interpretation of Manufacturer-Published Isolation Data

7.1 Use and Limitations of Isolation Nomograms

Vibration isolation performance is commonly communicated using nomograms or transmissibility plots supplied by manufacturers. These plots typically show the ratio of payload displacement to base displacement as a function of frequency, often normalised to unity at low frequencies.

Such nomograms provide valuable insight into the frequency ranges over which isolation systems attenuate motion and the location of resonant peaks, following principles and guidelines for laboratory measurement of vibro-acoustic transfer properties [11]. They are particularly useful for comparing displacement suppression across different architectures under nominal load conditions.

However, these plots are derived under controlled assumptions and represent only one aspect of system behavior. They characterize relative motion rather than the forces transmitted through the isolation interface or into the supported payload.

7.2 Displacement Transmissibility Versus Force Interaction

Displacement transmissibility describes how much a payload moves relative to its support. It does not, by itself, quantify the forces exchanged between the payload and the isolation system.

In compliant mechanical systems, force transmission depends on acceleration, stiffness, damping, (e.g., $F = ma$ for inertial coupling) and mass distribution, none of which are uniquely determined by displacement alone, but can be accessed via indirect methods for dynamic stiffness [12].

As discussed in Sections 2 and 3, the stylus-cartridge system responds to force and velocity variations rather than absolute displacement. Consequently, an isolation system may exhibit low displacement transmissibility while still transmitting low-frequency force variations that perturb the stylus operating point.

This distinction is particularly relevant near system resonances, where small base accelerations can generate significant inertial forces even when relative displacement remains small.

Displacement transmissibility is a kinematic ratio at two points; force at the stylus depends on accelerations, effective mass, dynamic stiffness, damping, and structural transfer paths (multiple DOF, not 1-DOF).

Even in LTI systems, mapping displacement $T(x) \rightarrow$ force at a remote interface does not uniquely determine force at a remote interface without system impedance information.

7.3 Low-Frequency Behavior in the 5-10 Hz Region

The 5-10 Hz frequency range is of special interest for analog disc playback, as it commonly overlaps both the tonearm-cartridge resonance and dominant building-borne vibration components. Manufacturer nomograms often show this region as one in which displacement transmissibility approaches unity or exhibit modest attenuation.

While such behavior may be acceptable for applications focused on positional stability, it does not preclude the presence of force modulation at the payload. In fact, the dynamic response of isolation systems in this region is often dominated by compliance and damping characteristics that influence reaction forces more than displacement magnitude.

As a result, performance assessments based solely on transmissibility curves may underestimate the influence of low-frequency excitation on force-sensitive components.

7.4 Methodological Considerations

It is important to emphasize that manufacturer-published nomograms are not incorrect or misleading; they are simply limited by the parameters they are designed to represent. These plots are typically generated under steady-state sinusoidal excitation and do not capture transient behavior, nonlinearities, or force-domain interactions.

For analog disc playback systems operating at nanometer-scale signal levels, additional evaluation criteria may be required to fully characterize isolation performance. Such criteria could include force transmissibility, acceleration coupling, or direct measurement of stylus force variation under controlled excitation.

7.5 Summary

Manufacturer-published isolation nomograms provide essential information about displacement suppression but do not fully describe the mechanical forces transmitted to a supported system. For applications involving force-sensitive transducers, reliance on displacement-based metrics alone may lead to incomplete conclusions.

This observation motivates consideration of system-level architectures that address both displacement and force-domain behavior, as discussed in the following section.

8. Architectural Implications for Turntable Design

8.1 Preservation of Stylus Force Integrity

The analyses presented in Sections 2-7 indicate that preserving signal integrity in analog disc playback requires attention not only to displacement suppression but also to the forces transmitted to the stylus-cartridge interface. Because the stylus operates under a large static force bias, even small low-frequency force variations can modulate the recovered signal through amplitude and phase mechanisms.

An effective turntable architecture should therefore seek to minimise low-frequency force modulation acting on the tonearm-cartridge system, particularly within the 5-10 Hz region where resonance and environmental excitation commonly overlap. This objective may not be fully addressed by isolation strategies optimised solely for displacement transmissibility.

8.2 Control of Low-Frequency Reaction Forces

Isolation systems inevitably interact with the supported structure through reaction forces. The magnitude and spectral content of these forces depend on system stiffness, damping, and, where applicable, active control behavior. For force-sensitive applications, it is desirable to avoid mechanisms that store or inject energy within the critical low-frequency band.

Architectural approaches that limit reactive energy storage or active force injection may reduce the potential for stylus force modulation, even if absolute displacement suppression is comparable to other solutions. This consideration highlights the importance of evaluating isolation performance in terms of force-domain behavior rather than displacement alone.

8.3 Reference Plane Stability and Record Support

In addition to isolating the turntable from external vibration, the internal reference plane formed by the platter, record, and tonearm mounting must remain mechanically coherent. Variations in record support stiffness or contact conditions can introduce additional force modulation pathways at the stylus-groove interface.

Vacuum record hold-down provides an additional mechanism for stabilising the mechanical reference plane formed by the record, platter, and tonearm. By coupling the record uniformly to the platter surface, vacuum hold-down reduces record warps and out-of-plane deformation that would otherwise introduce low-frequency vertical displacement during playback. Inclusion of vacuum hold-down as an exemplary mechanism offers a general architectural benefit.

Record warps typically occur at spatial wavelengths corresponding to rotational frequencies in the 0.5-5 Hz range (for 33.33 rpm). [2, 5] When traversed by the stylus, such warps impose cyclic vertical accelerations that modulate stylus force about the static tracking-force bias. This modulation occurs in the same low-frequency band identified in Section 3 as being capable of producing audible amplitude and phase modulation of mid-band program material. Although warps are primarily vertical disturbances, suspension geometry and friction couple vertical motion into lateral force components at the cantilever and vice versa.

By flattening the record and increasing the effective contact stiffness between the record and platter, vacuum hold-down reduces vertical acceleration of the stylus caused by record geometry rather than groove modulation. This reduction is expected to limit low-frequency force variation at the stylus-cartridge interface, complementing vibration isolation strategies that address externally induced disturbances. Importantly, this benefit arises from geometric stabilization and improved force distribution rather than from additional clamping mass, and therefore does not rely on increased inertia or altered resonance tuning.

Architectures that maintain a stable and well-defined reference plane under dynamic loading reduce the likelihood of relative motion between the record and stylus that is not directly related to groove modulation. This stability becomes increasingly important as playback systems approach the practical limits of nanometer-scale resolution.

Record eccentricity constitutes an additional geometric disturbance that introduces low-frequency lateral excitation at the stylus-groove interface. An off-centre record produces once-per-revolution radial displacement of the groove relative to the tonearm pivot, resulting in cyclic lateral acceleration of the stylus at approximately 0.55 Hz for 33.33 rpm playback.[5]

Although such frequencies lie below the conventional audio band, the resulting lateral acceleration modulates stylus force through cantilever compliance and suspension nonlinearity. This modulation acts on the same static tracking-force bias discussed in Sections 2 and 3 and can produce low-frequency frequency and phase modulation of the recovered signal.

Unlike record warps, eccentricity is not mitigated by vertical clamping alone. However, by stabilising the record-platter interface and preventing secondary rocking or slip, vacuum hold-down can reduce the compound interaction between warps and eccentricity-induced lateral motion and vertical stylus force variation.

8.4 System-Level Perspective

The preceding considerations underscore the need for a system-level perspective in turntable design. Vibration isolation, bearing behavior, motor drive, and record support are interdependent and should be evaluated collectively rather than in isolation. Design choices that optimise one aspect of performance may inadvertently compromise another if force-domain interactions are not considered. By explicitly accounting for stylus force integrity and low-frequency modulation pathways, turntable architectures can be better aligned with the physical constraints of analog disc playback.

9. Conclusion

Analog disc playback represents an extreme case of mechanical signal recovery, in which musical information encoded at nanometer-scale must be preserved in the presence of structural disturbances several orders of magnitude larger. Conventional evaluation of vibration isolation systems for turntables has largely relied on displacement transmissibility metrics developed for position-sensitive or inertial instruments.

This paper has shown that such metrics do not fully describe the mechanical conditions experienced by the stylus-cartridge interface. Because the stylus operates under a large static tracking-force bias, low-frequency disturbances can modulate stylus force and velocity even when platform displacement is minimal. These force variations, particularly within the 5-10 Hz region, can produce mid-band modulation products that may not be evident in displacement-only metrics (through established amplitude and phase modulation mechanisms).[7]

Electrical noise, rumble, and masking considerations were examined to demonstrate that nanometer-scale groove modulation remains resolvable within modern playback chains when analysed using appropriate bandwidth and noise-density concepts. The absence of directly audible low-frequency noise does not ensure reduced susceptibility from modulation-induced distortion.

Manufacturer-published isolation nomograms (see Appendix) were discussed in this context to illustrate the distinction between displacement suppression and force-domain behavior. While such data provide valuable insight into isolation performance, they do not fully capture force interactions relevant to force-sensitive transducers.

As summarized in Section 3.5 the analysis suggests that evaluation and design of turntable systems operating near the limits of analog resolution benefit from a force-domain perspective. Architectural approaches that minimise low-frequency force modulation at the stylus-groove interface, in addition to reducing displacement, are better aligned with the physical constraints of vinyl replay.

Disclosure

Two of the authors (Mark Doehmann and George Moraitis) are directors and shareholders of Döhmann Audio Pty Ltd, a manufacturer of high-end analog turntables incorporating vibration isolation and record stabilization technologies. One of the authors (Dallas Clarke) is an independent contractor that works with Döhmann Audio Pty Ltd. The present work is theoretical and analytical in nature, makes no claims regarding the performance of any specific commercial implementation, and references manufacturer-published data solely for illustrative purposes.

Appendix

Appendix A - Small-Signal Scaling, Groove Displacement, and Modulation Depth

A.1 Groove Displacement Corresponding to a Sinusoidal Velocity Signal

The instantaneous groove displacement $d(t)$is the time integral of the groove velocity $v(t)$:

$$d(t) = \int v(t)\, dt$$

For a single-frequency sinusoidal velocity signal at frequency f:

$$v(t) = v_{\mathrm{pk}} \sin(2\pi f t)$$

Integrating with respect to time yields:

$$d(t) = -\frac{v_{\mathrm{pk}}}{2\pi f} \cos(2\pi f t)$$

The peak displacement amplitude is therefore:

$$d_{\mathrm{pk}} = \frac{v_{\mathrm{pk}}}{2\pi f} \tag{A1}$$

This relationship applies strictly to steady-state sinusoidal signals.

If the velocity is specified in RMS form, the relationship between peak and RMS velocity is:

$$v_{\mathrm{pk}} = v_{\mathrm{rms}}\sqrt{2}$$

Substituting into (A1) gives:

$$d_{\mathrm{pk}} = \frac{v_{\mathrm{rms}}\sqrt{2}}{2\pi f} = \frac{v_{\mathrm{rms}}}{\pi f \sqrt{2}} \tag{A2}$$

Using the standard reference velocity of:

$$v_{\mathrm{rms}} = 0.05 \text{ m/s}$$

at:

$$f = 1000 \text{ Hz}$$

yields:

$$d_{\mathrm{pk}} = \frac{0.05}{\pi \times 1000 \times \sqrt{2}} \approx 11.25\ \mu\mathrm{m} \tag{A3}$$

This value (11.25 µm) represents the peak groove displacement corresponding to a 1 kHz reference tone at 5 cm/s RMS.

In the frequency domain, the displacement and velocity amplitudes are related by:

$$|D(\omega)| = \frac{|V(\omega)|}{\omega}$$

where $\omega = 2\pi f$, which yields the same result for single-frequency excitation.

A.2 Modulation Depth from Measured Sideband Levels

Low-frequency disturbances in vinyl replay systems commonly appear as symmetric modulation sidebands about a carrier frequency during standard wow-and-flutter measurements (e.g., a 3150 Hz test tone).

In the small-signal regime, such behaviour may be modelled using classical amplitude modulation (AM) or frequency modulation (FM) theory.[6]

A.2.1 Small-Signal Amplitude Modulation

For amplitude modulation of a sinusoidal carrier:

$$s(t) = A[1 + m\sin(2\pi f_m t)]\sin(2\pi f_c t)$$

where:

- A is the carrier amplitude
- f_c is the carrier frequency
- f_m is the modulation frequency
- m is the modulation index

Expanding this expression yields:

$$s(t) = A\sin(2\pi f_c t) + \frac{Am}{2}[\cos 2\pi(f_c - f_m)t - \cos 2\pi(f_c + f_m)t]$$

Thus, each first-order sideband has amplitude:

$$\frac{Am}{2}$$

relative to carrier amplitude A.

In decibels relative to carrier (dBc), the first-order sideband level is therefore:

$$\Delta L = 20\log_{10}\left(\frac{m}{2}\right) \tag{A4}$$

Rearranging gives:

$$m = 2 \cdot 10^{\Delta L/20} \tag{A5}$$

This relationship is valid provided:

$m \ll 1$ so that higher-order sidebands (proportional to $m^2$and above) remain negligible.

A.2.2 Small-Signal Frequency Modulation (Consistency Check)

For small frequency modulation with modulation index $\beta \ll 1$, the carrier may be written:

$$s(t) = A\sin\left(2\pi f_c t + \beta\sin\left(2\pi f_m t\right)\right)$$

Using the first-order Bessel approximation:

$$J_1(\beta) \approx \frac{\beta}{2}\text{for } \beta \ll 1$$

each first-order sideband has amplitude approximately:

$$\frac{A\beta}{2}$$

Thus, in the small-signal regime, AM and FM produce identical first-order sideband scaling with modulation index.

Accordingly, Eq. (A4) applies for either mechanism provided the modulation index is small.

A.2.3 Definition of Unity Modulation

In this appendix, $m = 1$corresponds to 100% amplitude modulation depth (carrier envelope reaches zero at negative peak).

This use of "unity" is distinct from vibration isolation transmissibility $T = 1$, which represents zero attenuation (0 dB).

A.3 Reference-Equivalent Displacement Scaling

The modulation index m obtained from measured sideband levels (Section A.2) quantifies the fractional magnitude of the modulation relative to the carrier in the small-signal regime. To provide a physically interpretable scale, it is convenient to express this modulation as a reference-equivalent displacement amplitude, defined relative to the peak groove displacement produced by a standard reference velocity at the carrier frequency.

For a sinusoidal groove velocity at carrier frequency f_c, the peak groove displacement at reference level $v_{\text{rms,ref}}$ is (from A.1):

$$d_{\text{ref}}(f_c) = \frac{v_{\text{rms,ref}}}{\pi f_c \sqrt{2}} \tag{A6}$$

Using the conventional reference $v_{\text{rms,ref}} = 0.05$ m/s(5 cm/s RMS), this yields $d_{\text{ref}}(1000\text{ Hz}) \approx 11.25\ \mu\text{m}$, and $d_{\text{ref}}(3150\text{ Hz}) \approx 3.57\ \mu\text{m}$.

We define the reference-equivalent displacement amplitude:

$$d_{\mathrm{eq}}(f_c) = m\, d_{\mathrm{ref}}(f_c) \tag{A7}$$

This mapping does **not** assert that the modulation physically corresponds to additional geometric groove displacement. Rather, it provides a consistent amplitude scale for comparing measured low-frequency modulation (expressed via m) against the nanometer-scale displacements associated with low-level recorded signals at the stylus–groove interface.

Worked Example (Carrier = 3150 Hz)

For a measured first-order sideband level of $\Delta L = -80$ dBc:

$$m = 2 \cdot 10^{\Delta L/20} = 2 \cdot 10^{-80/20} = 2 \cdot 10^{-4} = 0.0002 \tag{A8}$$

For a 3150 Hz carrier:

$$d_{\mathrm{ref}}(3150) = \frac{0.05}{\pi \cdot 3150 \cdot \sqrt{2}} \approx 3.57\ \mu\mathrm{m} \tag{A9}$$

Thus:

$$d_{\mathrm{eq}}(3150) = 0.0002 \times 3.57\ \mu\mathrm{m} \approx 0.714\ \mathrm{nm} \tag{A10}$$

For comparison, if the same modulation index is referenced to a 1 kHz carrier:

$$d_{\mathrm{eq}}(1000) = 0.0002 \times 11.25\ \mu\mathrm{m} \approx 2.25\ \mathrm{nm} \tag{A11}$$

Frequency Dependence

Because $d_{\mathrm{ref}}(f_c) \propto 1/f_c$, the reference-equivalent displacement scales inversely with carrier frequency:

$$d_{\mathrm{eq}}(f_c) \propto \frac{m}{f_c} \tag{A12}$$

Accordingly, identical sideband levels (identical m) correspond to different d_{eq}values depending on carrier frequency. For this reason, $d_{\mathrm{eq}}(f_c)$should always be reported with the carrier frequency f_c.

Interpretation and Limitation

- $d_{\mathrm{eq}}(f_c)$is a reference-equivalent amplitude scale, not literal groove displacement.
- It is intended for comparative interpretation of modulation sidebands in physical units.
- It is most appropriate in the small-signal regime where first-order sidebands scale linearly with modulation index.

A.3a Frequency-Adjusted Example for a 3150 Hz Test Tone

The reference displacement $d_{ref} \approx 11.25\ \mu m$was derived for a 1 kHz tone at 5 cm/s RMS.

However, wow-and-flutter measurements are typically performed using a **3150 Hz test tone** (per IEC/DIN practice).

Because displacement scales inversely with frequency:

$$d_{peak} = \frac{v_{rms}}{\pi f \sqrt{2}} \tag{A13}$$

the equivalent peak displacement at 3150 Hz is:

$$d_{ref,3150} = \frac{0.05}{\pi \cdot 3150 \cdot \sqrt{2}} \tag{A14}$$

$$d_{ref,3150} \approx 3.57\ \mu m \tag{A15}$$

Thus, the same 5 cm/s RMS velocity produces:

- 11.25 µm peak displacement at 1 kHz
- 3.57 µm peak displacement at 3150 Hz

So, for general frequency scaling for any test frequency f_t the formula:

$$d_{ref}(f_t) = \frac{0.05}{\pi f_t \sqrt{2}} \tag{A16}$$

and:

$$d_{eq}(f_t) = d_{ref}(f_t) \cdot m \tag{A17}$$

makes the frequency dependence explicit when comparing measurements performed at different carrier frequencies.

The following graph shows displacement vs frequency at constant velocity (log scale). Peak groove displacement versus frequency for constant RMS velocity of 5 cm/s. Displacement decreases inversely with frequency. Reference values at 1 kHz (11.25 µm) and 3150 Hz (3.57 µm) are indicated. The inverse scaling illustrates that constant velocity encoding does not imply constant displacement amplitude.

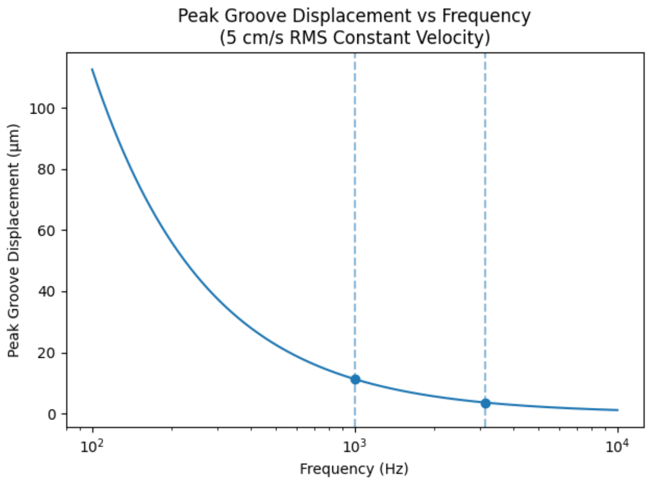

Figure 1.

$$d_{peak} \propto \frac{1}{f} \tag{A18}$$

Equation (A18) illustrates the inverse proportionality:

- 1 kHz ≈ 11.25 µm
- 3.15 kHz ≈ 3.57 µm
- 10 kHz ≈ ~1.1 µm

Here is a second plot showing:

- Equivalent displacement (nm) vs carrier frequency for a fixed measured sideband level of −80 dBc.

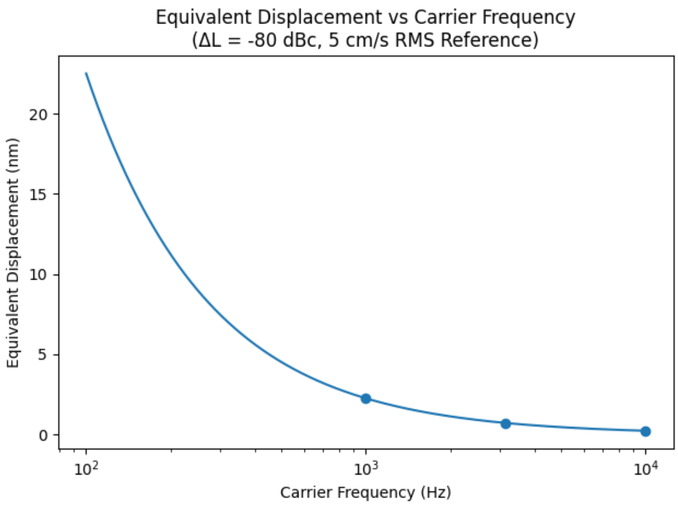

Figure 2.

At the same −80 dBc:

- 1 kHz → ~2.25 nm
- 3150 Hz → ~0.71 nm
- 10 kHz → ~0.23 nm

The equivalent displacement scales strictly as $1/f$.

Table A1 – Equivalent Displacement for −80 dBc Sideband

Carrier Frequency	Reference Displacement (µm)	Modulation Index (m)	Equivalent Displacement (nm)
1 kHz	11.25 µm	0.0002	2.25 nm
3.15 kHz	3.57 µm	0.0002	0.71 nm
10 kHz	1.13 µm	0.0002	0.23 nm

For a fixed measured sideband level, the equivalent displacement decreases inversely with carrier frequency; therefore, identical dBc values do not correspond to identical geometric amplitudes across different test tones.

A.4 Validity of the Linear Approximation

The modulation depths considered here remain well below unity (m ≪ 1), ensuring that stylus-groove contact is maintained and that the system operates within the linear elastic regime. Under these conditions, linear superposition applies and the force-domain analysis employed in this paper remains valid.

The linear regime holds when:

- Modulation index: $m \ll 1$ (typically $m < 0.1$, ensuring higher-order terms < 0.1%)
- Force perturbation: $\Delta F \ll F_0$ (e.g., <2 mN for a nominal 20 mN tracking force)
- Displacement perturbation: ≪ signal displacement (e.g., <1 µm mid-band)

These bounds are typical in high-performance turntables (wow/flutter <0.01%, structural accelerations <1 mg).

Note that although contact stiffness is load-dependent, the *fractional load variation* $\frac{\Delta F}{F_0}$ *is* small; therefore, the incremental stiffness variation is second-order for modulation indices.

We linearize about the DC bias F_0; the incremental response is approximately LTI for $\Delta F \ll F_0$.

A.5 Interpretation and Limitations

- d_{eq} is a heuristic scale, not literal groove displacement.
- It translates velocity/force modulation into an equivalent signal amplitude at reference level.
- Frequency mismatch: Adjust for test tone frequency as shown.
- Modulation type: Primarily FM; AM and PM follow similar scaling.
- Rumble (often AM-dominant), scaling is direct without Bessel approximation.
- No audibility thresholds are claimed (perceptual flutter JND ~0.2-0.5% is outside scope).
- Empirical use: Measure sidebands on test records (e.g., CBS STR-100, Ortofon) to quantify m, following established phono cartridge measurement standards [13].

A.6 Relevance to Vibration Isolation Design

Low-frequency force variations in the 5-10 Hz range produce velocity modulation without large platform motion.

Force-domain isolation architectures are conceptually aligned with reducing such modulation by directly limiting low-frequency force transmission rather than platform displacement alone, reducing equivalent modulation sidebands (and corresponding d_{eq} values) well below the level of low-level musical content - while remaining firmly in the small-signal regime (modulation index $m \ll 1$, far from $m = 1$).

This supports the paper's argument that force-based metrics better predict stylus-groove interface performance.

A.7 Sample Transmissibility Nomograms

For reference the following sample nomograms are provided for exemplary purposes only. These highlight a sample manufacturers published data for a standard commercial off the shelf negative-stiffness passive system. No claims on the sonic attributes or psychoacoustics of analog playback are made. No comparative publicly available active or air isolation transmissibility nomograms are published in this paper.

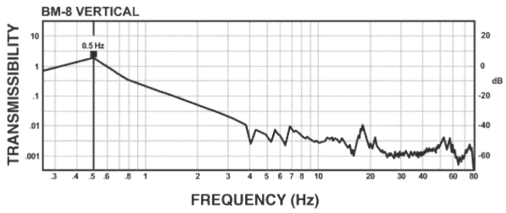

Figure 3

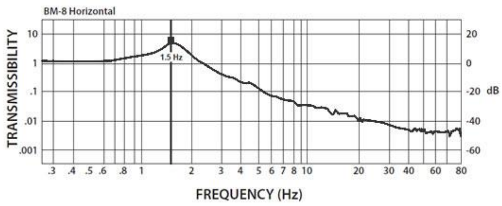

Figure 4

The transmissibility curves or a passive negative stiffness isolator are pictured above (Fig 3 and Fig 4). These were generated using a Stanford Research Instruments SR785 Dynamic Signal Analyzer. The output source of the SR785 is set to generate a swept sine signal. This signal was fed into a Labworks PA-138 power amplifier. The amplifier drives a Labworks ET-126 electrodynamic shaker. The shaker was mounted in a heavily reinforced frame that also supports an isolator. This support isolator has been adjusted to about 2.75 Hz. It has also been modified to work only in the vertical direction. This isolator supports a heavy top plate and whatever ballast weight is needed to bring the total payload including the test isolator up to around 650 pounds (~295Kg).

Transmissibility as it applies to the pictured isolators is a ratio of the output signal on the top plate divided by the input signal that the base of the isolator sees. Two similar accelerometers are used to acquire the input and output signals. The input accelerometer is attached to the heavy top plate on the support isolator. The test isolator rests on the heavy top plate as well. The accelerometer measures the vibrations that are fed to the isolator. The output accelerometer is placed on top of the properly loaded test isolator top plate. Both accelerometers are held in place with a thin layer of seismic wax.

The vertical transmissibility shown in Fig 3 shows vertical performance of a commercial passive negative stiffness isolation system, determined using direct methods for dynamic stiffness of resilient supports [14].

The horizontal transmissibility was acquired in much the same way as the vertical data. Fig 4 shows horizontal performance of a commercial passive negative stiffness isolation system. The SR785 acquires both sets of data, calculates their ratio and displays the ratio as transmissibility. The manufacturer provides these images for illustrative use only.

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Standards and Measurement References

These standards are cited to clarify terminology and measurement conventions rather than to prescribe specific test methods for turntable evaluation.

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