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Vibration Isolation System For Near Field Speakers In Sound Recording Studio

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ABSTRACT

When designing near field speakers position in sound recording studios, vibration transmission are rarely taken into account. Speakers can generate very low frequencies that can affect to microphone measurements even if they are placed far from the sources. This paper presents a prototype design for all the typical speakers range and their supports that cut the vibration transmission.

Keywords: Vibrations, Speakers, Isolators
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1. INTRODUCTION

Vibration transmission of speakers is not always taken into account when designing sound recording studios. Furthermore, classic theories did not make any mention. When talking about far field speakers, they will normally work only in mixing stage, so they will never interfere in proper functioning of the recording measurement chain. This is not the case of near field speakers that are normally playing in real time the recording season in Control Room. Even if a "box in a box" criteria is been followed for isolating Recording Room from the other ones, near field speakers can transmit low frequency structural noise (normally from 20 Hz) causing a high risk of impacting measurement chain via.

2. DEFINITIONS

2.1. Waves

A sound wave is a transfer of energy traveling away from a vibrating source. Sound waves are formed when a vibrating object causes the surrounding medium to vibrate. This medium is a material (sold, liquid or gas). Some concepts are important to be taken into account first of all:

Equation 1.- Period (T): duration of one cycle [s]

$$T = \frac{1}{n}$$

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Equation 2.- Frequency: number of cycles per second [Hz]

$$n = \frac{1}{T} = \frac{\omega}{2\pi}$$

Figure 1.- Amplitude vs Time scheme

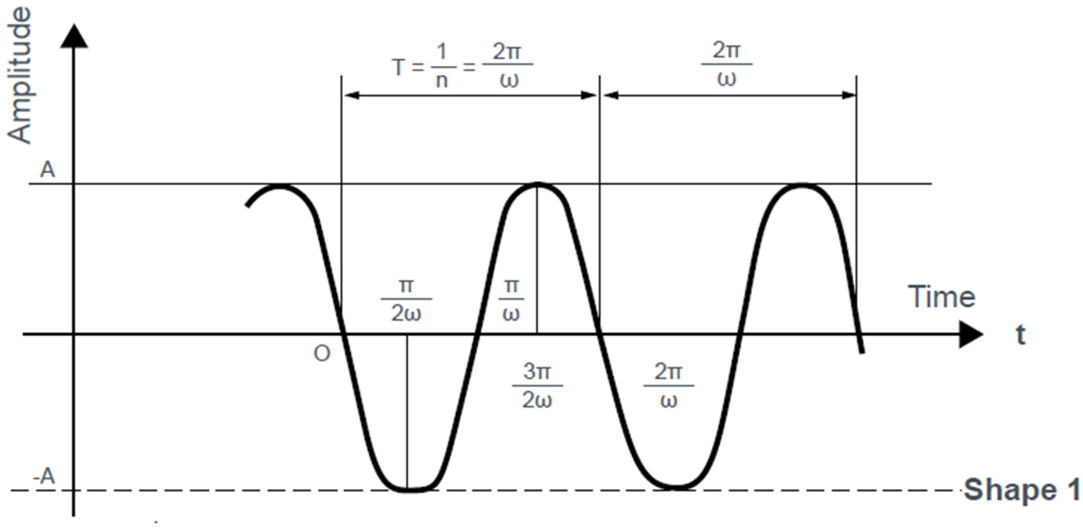
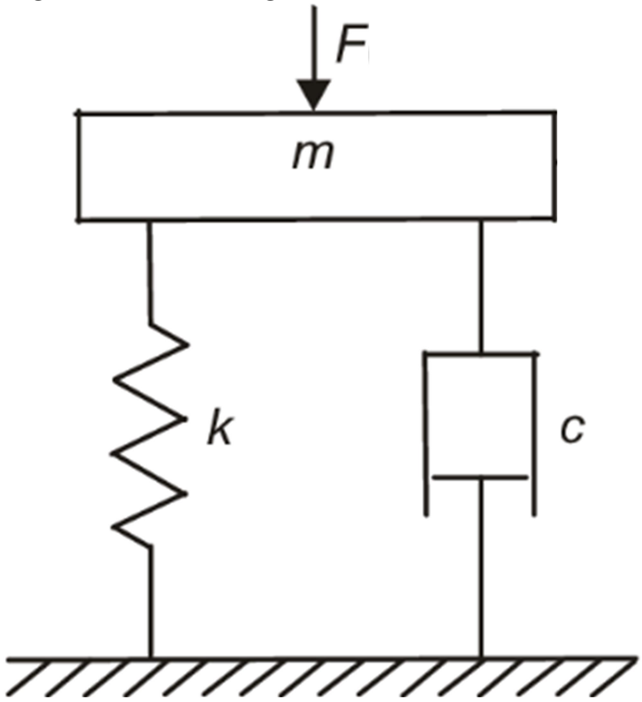


Figure 2.- One degree of freedom figure



F Force [Kg]
K stiffness [N/m]
m Mass [Kg]
C Damping

Equation 3.- Undamped natural frequency [Hz]

$$\omega_0 = \frac{1}{2\pi}\sqrt{\frac{K}{M}}$$

K Isolator stiffness [N/m]
M Mass [Kg]

Equation 4.- Ideal spring deflection [mm]

$$\sigma = \left(\frac{0.5}{\omega_0}\right)^2 \cdot 1000$$

Equation 5.- Undamped vibration attenuation [%]

$$E\,(\%) = 100 \cdot \frac{\left(\frac{\omega}{\omega_0}\right)^2 - 2}{\left(\frac{\omega}{\omega_0}\right)^2 - 1}$$

ω Source forced frequency
ω_0 Isolator working frequency

Equation 6.- Undamped vibration transmissibility [%]

$$T = 1 - E(\%)$$

Equation 7.- Undamped vibration attenuation [dB]

$$E\,(dB) = \log_{10}\left(\frac{100}{100 - E\%}\right)$$

2.2. Springs design theory

Equation 8.- Spring stiffness (Force/ Deflection)

$$K = \frac{G \cdot d^4}{8 \cdot D^3 \cdot N_a}$$

G shear module of the spring wire
D diameter of spring loop
d wire diameter
Na number of active spring loops

Equation 9.- Spring deflection (Hooke's Law)

$$\delta = \frac{\tau}{G}$$

T tension
G shear module of the spring wire

Equation 10.- Tension (Wahl correction factor)

$$\tau = K_w \cdot (8 \cdot F \cdot D / \pi \cdot d^3)$$

F force value applied
D diameter of spring loop
d wire diameter
Kw Wahl correction factor ≈ 1

Equation 11.- Spring deflection

$$\delta = \frac{8 \cdot F \cdot D}{G \cdot \pi \cdot d^3}$$

3. STUDY CASE

3.1. First considerations

After a consultation of 25 different near field speakers models of different brands, we can affirm it weight goes from 6 to 12 Kg for each device.

Regarding speaker supports, after making an analog consultation, resulting weight range is from 10 to 14 Kg

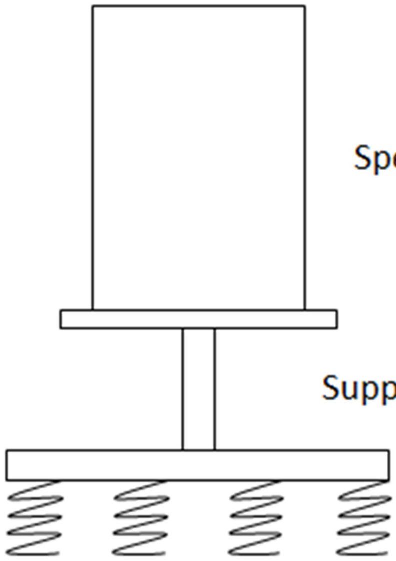
That means total mass of each speaker plus support will be:

- Minimum: 16 Kg
- Maximum: 26 Kg

Normally each assembly speaker + support will be supported for 4 supporting feet. So supposing a centered Center of Gravity, each support will load approximated:

- Minimum: 16/4 = 4 Kg
- Maximum: 26/4 = 6.5 Kg

Figure 3.- System scheme



We can consider lower frequency range that a near field speaker can play is:

- $\omega \geq 20$ Hz

3.2. Isolation target

In order to ensure a minimum vibratory transmission, we fixed a target of 85% isolation against the lowest frequency (20 Hz).

- E (%) = 85

Using equation 5, we can calculate working frequency of isolators should be:

- $\omega_0 \leq 7.224$ Hz

3.3. Isolator design criteria

For these calculations an undamped system was considered. A lineal (ideal) stiffness was also considered. Lateral stiffness was not considered.

As previously mentioned, minimum and maximum weight for each isolator was:

- Minimum: 16/4 = 4 Kg
- Maximum: 26/4 = 6.5 Kg

To reach ω_0, using equation 3, we can fix stiffness (K) target for minimum and maximum weight:

- Min. weight (4 Kg) $\rightarrow$ $K \leq 8240.91$ N/m
- Max. weight (6.5 Kg) $\rightarrow$ $K \leq 13391.48$ N/m

That means target stiffness will be minimum of these values:

- Stiffness target $K \leq 8240.91$ N/m

To reach these targets, a metal spring isolator was designed.

3.4. Isolator design

Using equations 8 we designed a metal spring isolator.

First of all we fixed wire material and diameter according the raw materials that are easy to obtain in the market.

- Wire diameter (d) = 3 mm
- Material: 1080 steel

Then we fixed the other parameters:

- Number of active spring loops (Na) = 2.5
- External spring diameter (D) = 38 mm
- G shear module of the spring wire = 83324148

With this values, we calculated an stiffness (K) = 6150 N/m

Using equations 9, 10 and 11, and taking into account wire diameter, we calculated an optimum free height of 50 mm and a block height of 30 mm.

We considered the next values for minimum, maximum and nominal load:

- Minimum: 30% deflection - 6 mm – 3.7 Kg
- Maximum: 60% deflection - 12 mm – 7.5 Kg
- Nominal: 40% deflection - 8 mm – 5 Kg

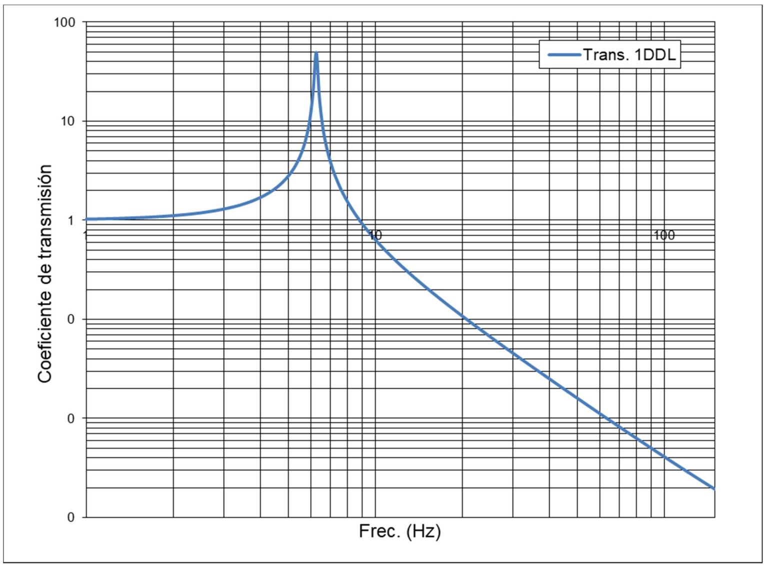
3.5. Calculations results

Below are shown results for minimum and maximum target weight and isolation against minimum excitation frequency (20 Hz) using equations 3, 4, 5, 6 and 7.

3.5.1. Minimum load (4 Kg)

$\omega_0 = 6.241$ Hz
Deflection (σ) = 6.419 mm
Attenuation (E) = 89.21%
Transmissibility (E) = 10.79%
Attenuation (dB) = 19.3 dB

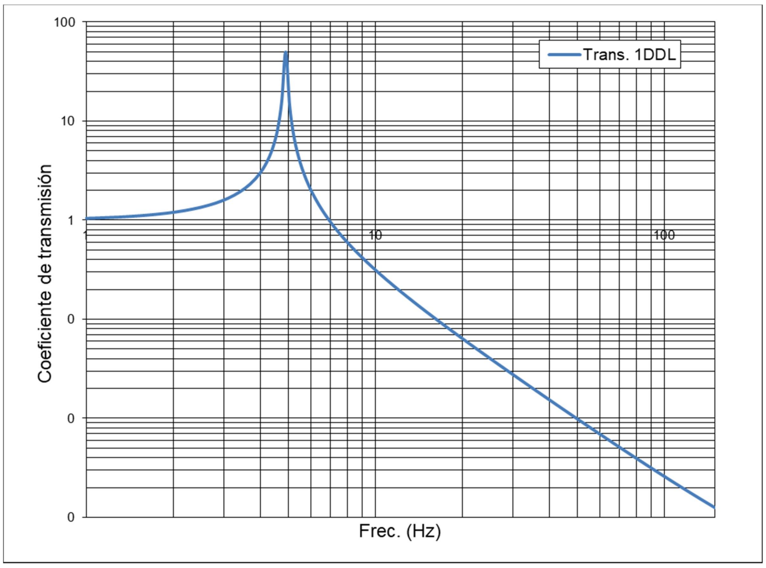
Figure 4.- Transmissibility (1% damping)



3.5.2. Maximum load (6.5 Kg)

ω_0 = 4.896 Hz
Deflection (σ) = 10.431 mm
Attenuation (E) = 93.63%
Transmissibility (E) = 6.37%
Attenuation (dB) = 23.9 dB

Figure 5.- Transmissibility (1% damping)



4. CONCLUSIONS

Designed spring matches with all proposed targets so they are appropriated for this application. Next steps would be to build a prototype and make real tests in a force vs. deflection press machine.

5. REFERENCES

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