STRUCTURE-BORNE SOUND PROPERTIES OF WIRE ROPE ISOLATORS

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ABSTRACT

In presenting an indirect method for the dynamic stiffness of wire rope isolators in the audible frequency range, the stiffness dependence on static preload, vibration amplitude and frequency is resolved. It is shown that although the stiffness is strongly dependent on the frequency, the peaks are rounded; indicating rather high damping – contrary to those of ordinary steel springs – and, thus, is displaying similar behavior as rubber isolators. The main reason for the increased damping is the friction motion between the separate threads of the cable twist. Furthermore, the stiffness is independent on vibration amplitudes typically applied within the audible frequency range, while showing a substantially lower stiffness for significantly higher amplitudes, typically applied for low-frequency applications. Finally, the dynamic stiffness is displaying a strong static preload dependence; the higher the preload, the lower is the stiffness. In conclusion, wire rope isolators – typically applied for shock isolation and low-frequency applications – may also be suitable for noise and vibration applications within the audible frequency range, contrary to ordinary metal springs.

1. INTRODUCTION

Vibrations generated by machines or other sources frequently cause unwanted effects while transmitting to a receiving structure. As a general rule of thumb, structure-borne sound, that is vibrations in the audible frequency domain, radiates sound while low-frequency vibrations may cause structural fatigue and failure. A simple vibration transmission reduction is attained when decoupling the source from a receiving structure by mounting it upon wire rope isolators, see Figure 1. Traditionally, a wire rope isolator provides insulation designed to meet low-frequency requirements, while its structure-borne sound properties remain arbitrary. Moreover, wire rope isolators are frequently used for shock isolation applications. However, increasing interest in noise abatement has heightened the need for effective isolation within the audible frequency domain, requiring structure-borne sound property data from the source, the vibration isolator and the receiving structure. In this proceeding, the wire rope isolator's audible stiffness is focused upon, displaying measurements results.

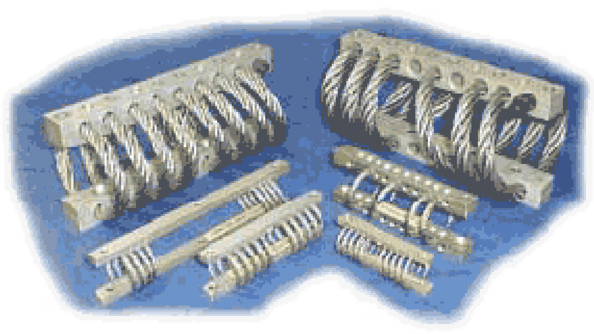

Figure 1. *Wire rope isolators.*

1.1. Test object

The wire rope isolator in Figure 2 consists of a double twisted cables of steel threads AISI 316-T6, wrapped *N* = 8 orbits between the two aluminum bars T6, with chromate finish, of length *L* = 216 mm, width *w* = 25 mm and height *h*= 20 mm. The total length, width and height are *L* = 216 mm, *W* = 133 mm and *H*= 108 mm, respectively. The wire rope isolator is manufactured by Vibratec, belongs to the article series A13 and is denoted A130216-108-II. The tabulated, nominal low-frequency stiffness at 10 Hz is 4.54×10^{5} N/m.

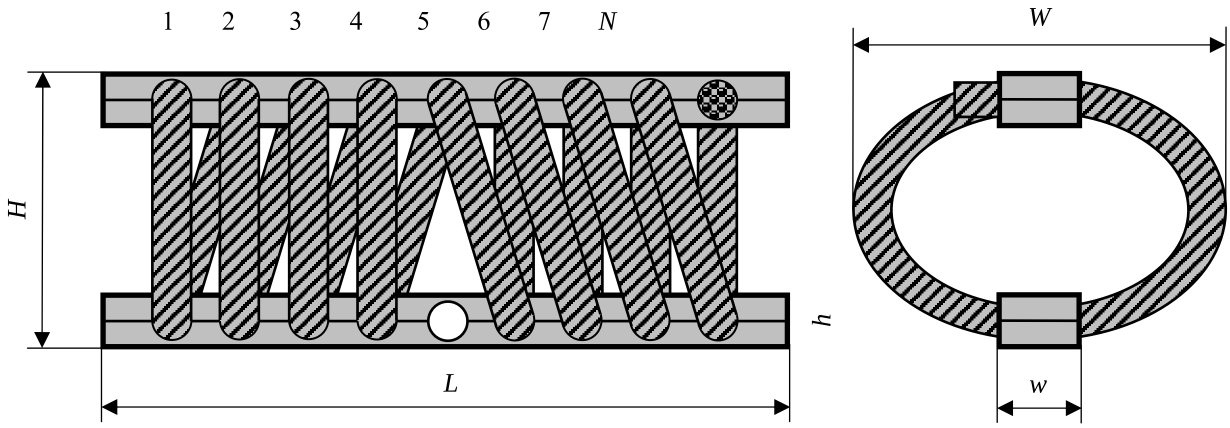

Figure 2. *Wire rope isolator with dimension variables.*

1.2. Method

A specially designed test rig at the Marcus Wallenberg Laboratory for Sound and Vibration Research in Figure 3 is utilized for measuring the dynamic stiffness of the wire rope isolator within the audible frequency range. The reader is referred to Ref. [1] regarding the theoretical details on the measurement procedure. The wire rope isolator is inserted between a moving and blocking mass. The blocking mass is decoupled from the floor by three soft, symmetrically positioned auxiliary isolators. The static preload on the test object is applied by the gravidity force from the moving mass. Various moving masses (with different mass) are applied to generate a range of preloads. An electro-dynamic vibration generator (exciter) excites the moving mass. The exciter is hanging from the test frame by two soft rubber bands. The motion is measured by four piezo-electric accelerometers; two symmetrically positioned, isolator close, on the moving mass bar and two symmetrically positioned, isolator close, on the top surface of the blocking mass. The electrical signals are conditioned in charge amplifiers without explicit time integration to increase the high-frequency accuracy. The measurement is processed by a personal computer using a 4-channel frequency analyzer to collect data and supply the signal to the generators via an amplifier. The acceleration and mass of the blocking mass multiplied, supplies the blocking force needed for the dynamic stiffness estimation. All necessary instrumentation is given in Table 1. A photo of the test set-up is shown in Figure 4.

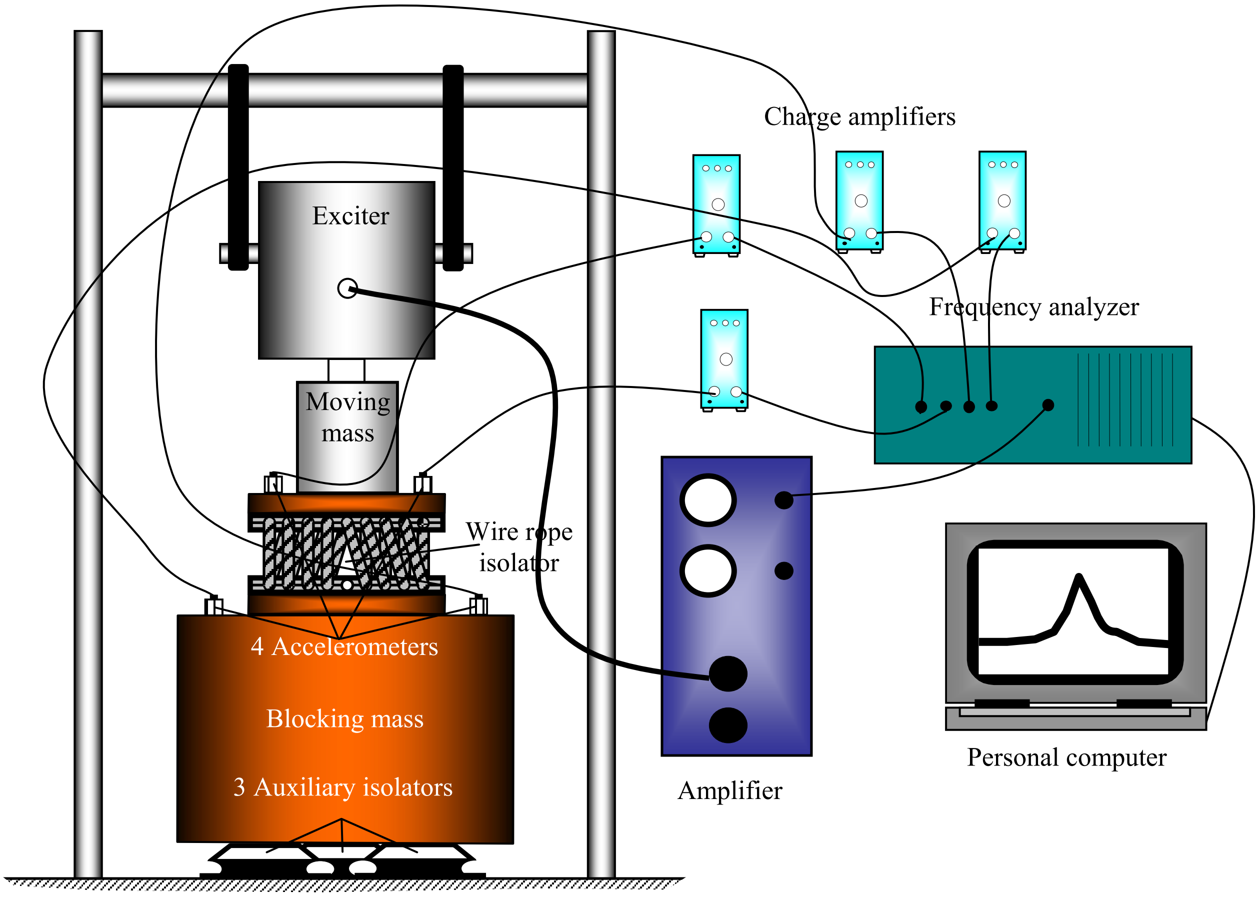

Figure 3. *Test set-up with measurement instruments.*

Instrument	Type	Number
Excitator	LDS V409	1
Amplifier	Labgruppen SS1400	1
Accelerometer	B&K 4367	2
Accelerometer	B&K 4338	1
Accelerometer	B&K 4370	1
Charge amplifier	B&K 2635	4
Frequency analyzer	Tektronix 2630	1
Personal compuiter	Aquila 486	1

Table 1. *Test rig instruments.*

Figure 4. *Test set-up photo*.

For sufficiently high frequencies, Newton's second law provides an estimate of the blocking force on the wire rope isolator

$$\widetilde{F} = \omega^2 M_{\mathrm{bm}} \widetilde{U}_{\mathrm{bm}}, \tag{1}$$

needed for the dynamic stiffness

$$K = \omega^2 M_{\mathrm{bm}} \frac{\widetilde{U}_{\mathrm{bm}}}{\widetilde{U}_{\mathrm{mm}}}, \tag{2}$$

where $\omega = 2\,\pi\, f$ is angular frequency, $\widetilde{U}_{\mathrm{bm}}$ and $\widetilde{U}_{\mathrm{mm}}$ are vertical displacement of the blocking and moving mass, respectively, and $(\tilde{\cdot})$ denotes temporal Fourier transform. The mass of the blocking mass is $M_{\mathrm{bm}} = 473$ kg. Through the frequency response function $\widetilde{H}(\widetilde{X},\widetilde{Y}) = \widetilde{Y}/\widetilde{X}$ the stiffness becomes

$$K = \omega^2 M_{\mathrm{bm}} \frac{\widetilde{H}(\widetilde{A}^1_{\mathrm{mm}},\widetilde{A}^1_{\mathrm{bm}}) + \widetilde{H}(\widetilde{A}^1_{\mathrm{mm}},\widetilde{A}^2_{\mathrm{bm}})}{\widetilde{H}(\widetilde{A}^1_{\mathrm{mm}},\widetilde{A}^1_{\mathrm{mm}}) + \widetilde{H}(\widetilde{A}^1_{\mathrm{mm}},\widetilde{A}^2_{\mathrm{mm}})}, \tag{3}$$

where the vertical accelerations $\widetilde{A}^1_{\mathrm{mm}}$ and $\widetilde{A}^2_{\mathrm{mm}}$ are measured on the moving mass and $\widetilde{A}^1_{\mathrm{bm}}$ and $\widetilde{A}^2_{\mathrm{bm}}$ on the blocking mass.

1.3. Dynamic stiffness at no preoad

The measurements are performed at room temperature (23°C) and within a frequency range of 100 to 2000 Hz. The measurements are reliable up to 1000 Hz. The measurement errors start to notably grow above 1000 Hz due to non-rigid motion of the blocking mass; the higher the frequency, the bigger is the error. In order to increase the signal-to-noise ratio, the excitation signal is a stepped sine signal; starting from 100 Hz, with a constant step of 10 Hz. The signal is recorded within a 10 Hz bandwidth, averaged 5 times and delayed 500 ms between each recording.

The magnitude $|K|$ and phase $\angle K$ of dynamic stiffness K are defined by $K = |K| \exp(\mathrm{i}\angle K)$ where i denotes the complex unit. The dynamic stiffness magnitude and phase versus frequency at vanishing preload are shown in Figures 5 and 6, respectively, using logarithmic scales. Clearly, the magnitude and phase are rather constant up to approximately 700 Hz, where the magnitude increases sharply while the phase decreases 180 degrees. The magnitude peak at 1000 Hz is an anti-resonance. At an anti-resonance for elastic materials (that is, with no material damping) the stiffness shows a magnitude peak and a phase jump of – 180°, while displaying a magnitude trough and a phase jump of + 180° at a resonance. Introducing damping, the resonances and anti resonances are blunted; in general, the magnitude troughs and peaks are rounded, while the sudden phase jumps disappear, showing a ‘slower’ phase shift. Consequently, the peak of the studied wire rope isolator is rounded; indicating rather high damping – contrary to those of ordinary steel springs – and, thus, is displaying similar behavior as rubber isolators. The main reason for the increased damping is the friction motion between the separate threads of the cable twist.

The minor fluctuations of the stiffness magnitude and phase are due to measurement noise, in particle at higher frequencies. In the higher frequency range (above 1000 Hz), the non-rigid motion of the blocking mass influences the results to a great extend.

The rather constant stiffness magnitude in the frequency range 100 to 700 Hz is considerably higher (almost 10 times) than the tabulated stiffness in Section 1.1. A plausible explanation of this behavior is the non-linear properties of wire rope isolators, displaying a strong dynamic amplitude dependence; the higher the amplitude, the lower is the stiffness magnitude. A physical explanation to this is that the separate threads of the cable twist are close and in contact to each other at small vibration amplitudes, thus, displaying a high global stiffness magnitude, while breaking from each other at higher vibration amplitudes and, consequently, displaying a lower global stiffness magnitude. The nominal stiffness magnitude in Section 1.1 is evaluated at considerably higher amplitudes (in mm-range) than in our test (in μm-range), since the amplitudes are in those ranges for low-frequency applications (in mm-range) and high-frequency [audible frequency range] applications (in μm-range), respectively.

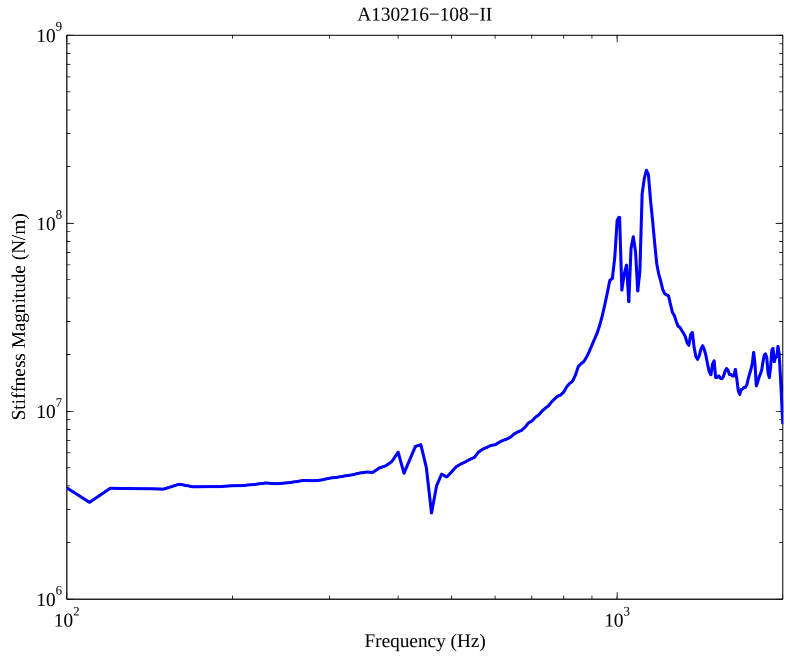

Figure 5. *Stiffness magnitude versus frequency at no preload*.

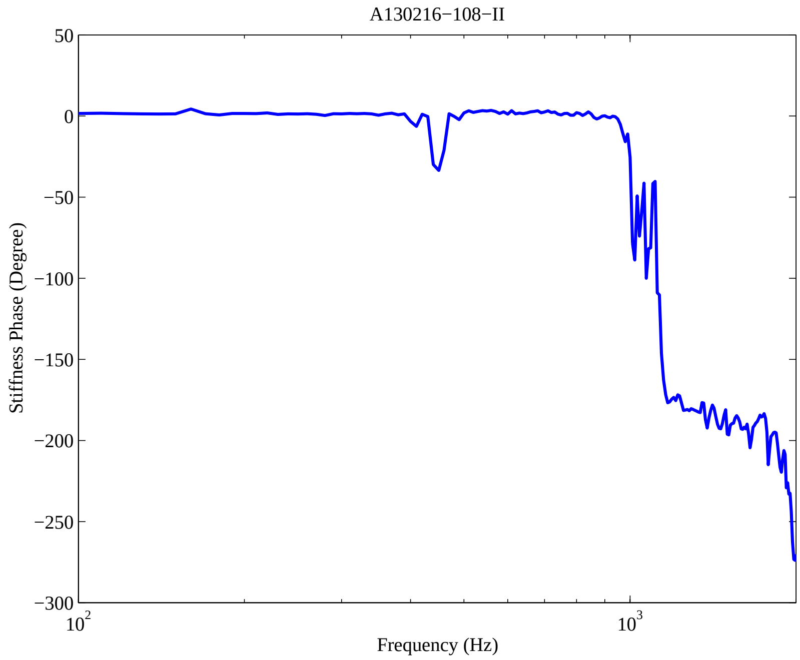

Figure 6. *Stiffness phase versus frequency at no preload*.

1.4. Dynamic amplitude dependence

Three vibration amplitudes within the audible frequency range are applied: `Low´, `Normal´ and `High´, all in the µm-range and where the amplitude ratio between `High´ and `Normal´ and between `Normal´ and `Low´ is approximately 10, and where `Normal´ refers to the amplitude used in the measurement above.

The dynamic stiffness magnitude and phase versus frequency at `Low´, `Normal´ and `High´ vibration amplitudes are shown in Figures 7 and 8, respectively, using logarithmic scales. Clearly, the curves are almost identical. Consequently, the dynamic stiffness is independent on vibration amplitudes typically applied within the audible frequency range, while showing a substantially lower stiffness magnitude for significantly higher amplitudes, typically applied for low-frequency applications.

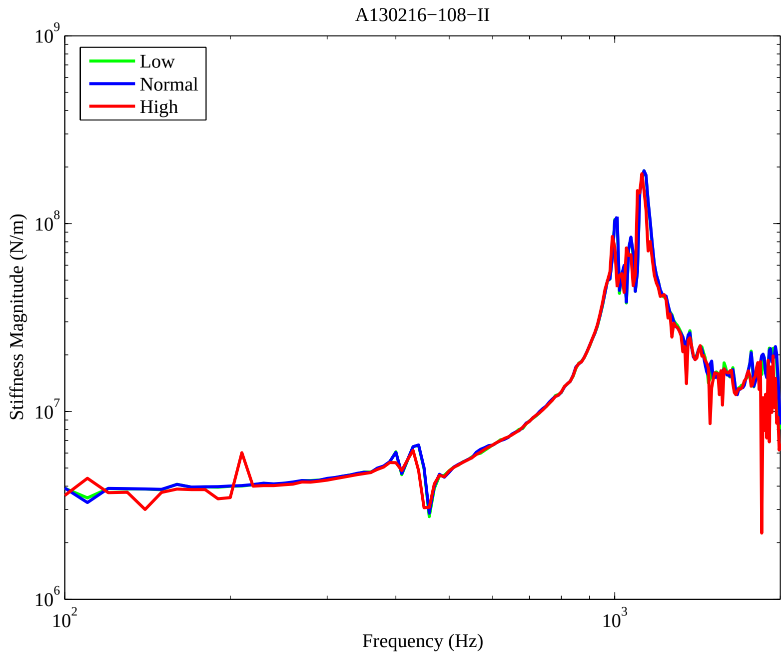

Figure 7. *Stiffness magnitude versus frequency at different vibration amplitudes.*

1.5. Static preload dependence

Four different moving masses within the audible frequency range are applied giving the following static preloads: `0 N´, 332 N, 1200 N (≈ nominal preload according to manufacturer) and 2200 N (≈ max preload according to manufacturer). The case `0 N´ refers to the measurement above in Section 1.3 (using a very light aluminum cylinder as a moving mass).

The dynamic stiffness magnitude and phase versus frequency at different static preloads are shown in Figures 9 and 10, respectively, using logarithmic scales. Clearly, the rather constant stiffness magnitude in the frequency range 100 to 700 Hz decreases with static preload and the anti-resonance magnitude peak frequency decreases with static preload. A plausible explanation of this behavior is the geometrical re-configuration the wire rope isolator undergoes at static preload in Figure 11; the height H decreases to H_{Preload} while the width W increases to W_{Preload}. This makes the wire rope isolator less stiff in the low-frequency range 100 to 700 Hz.

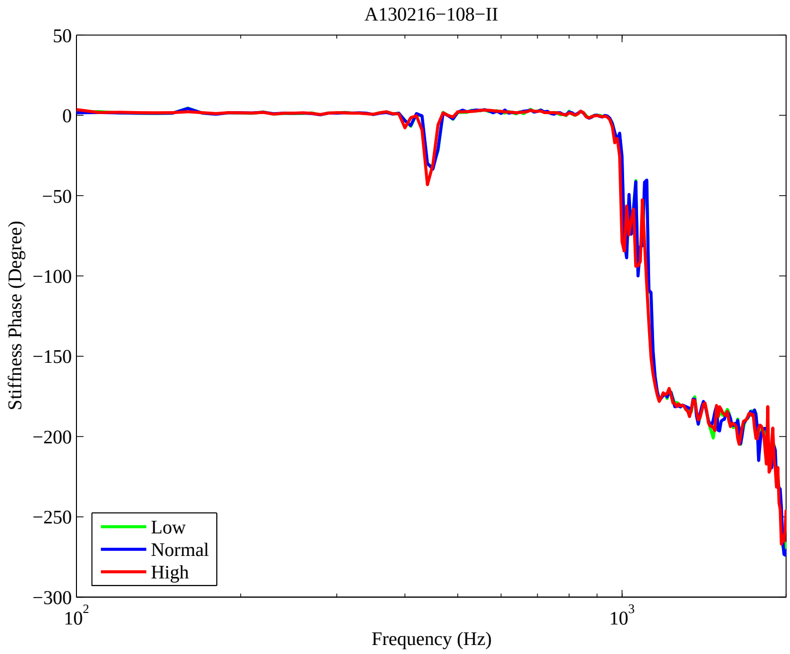

Figure 8. *Stiffness phase versus frequency at different vibration amplitudes.*

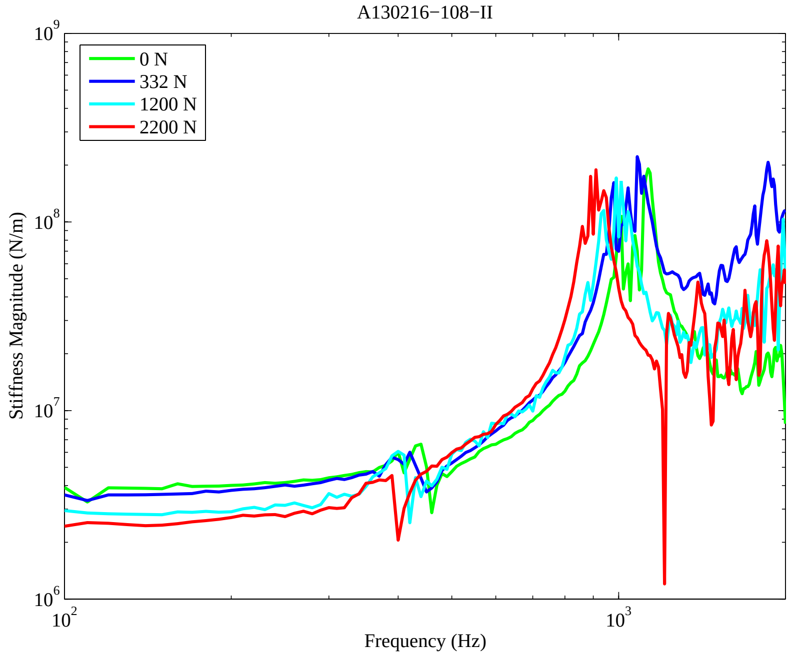

Figure 9. *Stiffness magnitude versus frequency at different static preloads.*

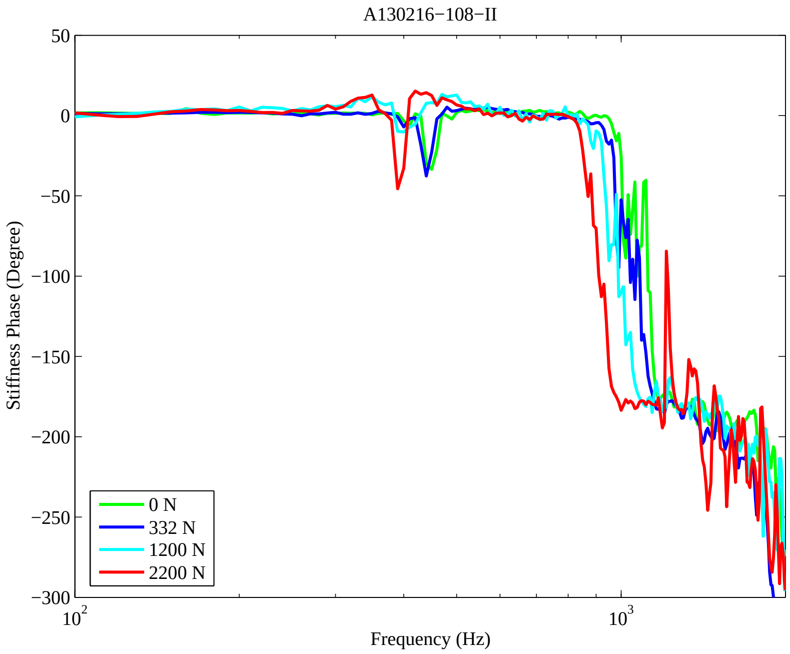

Figure 10. *Stiffness phase versus frequency at different static preloads.*

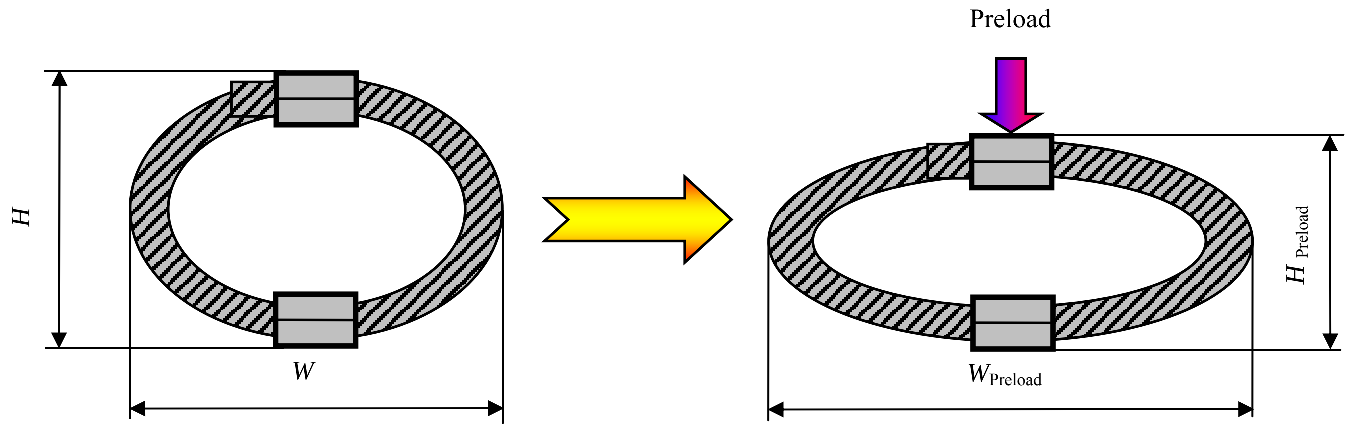

Figure 11. *Geometric re-configuration of wire rope isolator while applying a preload.*

1.6. Acknowledgements

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2. CONCLUSIONS

A specially designed test rig at the Marcus Wallenberg Laboratory for Sound and Vibration Research is utilized for measuring the dynamic stiffness of the wire rope isolator within the audible frequency range. It is shown that the low-frequency stiffness up to 700 Hz is rather frequency independent while is displaying a strong dependence on the frequency above 700 Hz where the anti resonance peak magnitude is rounded;

indicating rather high damping – contrary to those of ordinary steel springs – and, thus, is displaying similar behavior as rubber isolators. The main reason for the increased damping is the friction motion between the separate threads of the cable twist. Furthermore, the stiffness is independent on vibration amplitudes typically applied within the audible frequency range, while showing a substantially lower stiffness for significantly higher amplitudes, typically applied for low-frequency applications. Finally, the dynamic stiffness is displaying a strong static preload dependence; the higher the preload, the lower is the stiffness. In conclusion, wire rope isolators – typically applied for shock isolation and low-frequency applications – may also be suitable for noise and vibration applications within the audible frequency range, contrary to ordinary metal springs.

3. REFERENCES

[1] Kari, L. "Dynamic transfer stiffness measurements of vibration isolators in the audible frequency range," *Noise Control Engineering Journal* 49: 88-102, 2001.