SB - Noise Control Design Rules & Rules of Thumb revisited by numerical Experiments

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Introduction

Noise Control **Design Rules** and **Rules of Thumb** for structure-borne noise ('SBN') are a kind of recipes and valuable tools for the daily noise control by design process as well as for trouble shooting tasks, even so they are rough procedures, practices or guesstimates. A few of these SBN-rules are investigated (e.g. λ/2 -Rule for Mass, see Figure 1) in numerical experiments to show the effect of the applied methods of SBN-reduction with regard to transmission and radiation. Therefore three simple scenarios from literature are numerically revisited and discussed.

Motivation

Bombardier manufactures trains and air-planes and it obvious that for functional vehicles lots of SBN-sources (e.g. compressors, coolers) need to be connected to the car-body without generation, propagation or radiation of excessive sound. **But why are rules still needed if 'anything' can be measured or computed?** Measurements are only possible if a prototype does exist, simulation also takes its preparation and time, and it is certainly not possible to compute every (e.g. mass-, stiffness- or damping-) variation, you need to have a good proposal to start from. In daily engineering life you are still asked for instant as well as long term solutions: it is important to develop both intuitive as well as conceptual-rational thinking and to use a clear communication – based on quantitative values- to the designers. Numerical tools and the vast pool of literature examples seem to be an ideal solution for these demands. The investigated scenarios II & III are examples documented in [3]. Additionally it is possible to investigate and test measurement rules and scenarios.

Scenario I: λ/2-Mass-Rule

Task: Connect a source with a given structure
Structure: 4 mm thick, stiffened steel-plate
Source: Force source (flat spectrum up to 2200 Hz)

Question: What mass in kg needs to be added to reduce the radiated sound above 300 Hz?
Rule: the add-on mass needs to be heavier than a part of the excited structure with the dimensions of ca. half a wavelength (λ/2).

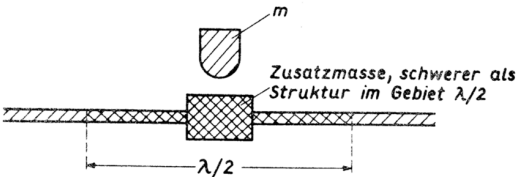

Figure 1: Original drawing from M. Heckl [1])

The procedure is quite simple: 1. get an estimate of the relevant wavelength and compute the needed add-on-mass (the mass chosen here is intentionally equal to the structure with the dimensions of half a wavelength, not bigger as demanded by the rule!), 2.apply the add-on mass once as point mass and once as distributed mass (e.g. by increased thickness), 3.compute the SBN and air-borne noise ('ABN') relevant spectra (e.g. sound power).

The input mobility Y(f) at the mounting point (s. Figure 2) is here close to the input mobility of the infinite plate for higher frequencies. The mobility does already provide a quantitative insight of the influence of the added mass (1.62 kg) for the three considered cases. From Figure 2 it could be concluded that for the medium frequencies the distributed mass works best and for very high frequencies >2 kHz: the point mass seems to be the best solution.

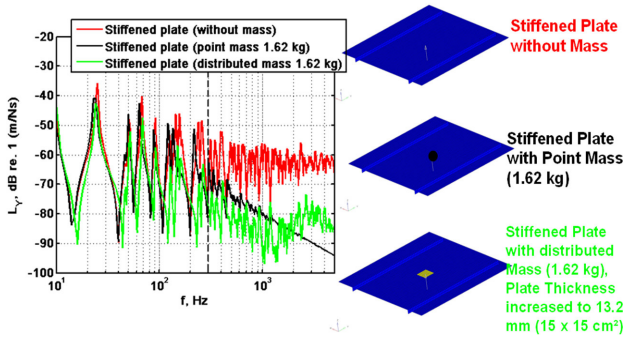

Figure 2: Input mobility L_Y=20*log10(|Y|) at source mounting point for original plate, plate + point mass at mounting point and plate + distributed mass

With regard to the radiated sound power (Figure 3) the point mass configuration provides -for all frequencies above 1.2 kHz- the best performance. The radiation efficiency of the original plate and the point-mass configuration are identical; the radiation for the distributed mass configuration is increased in certain frequency regions.

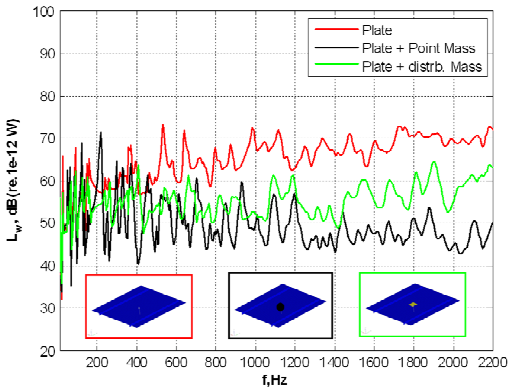

Figure 3: Radiated Sound-Power L_w (ABN)

Scenario II: Spring design & insertion loss

Task: connect a vibration-exciter with a given structure
Structure: 4 mm thick, stiffened steel-plate, keep tuning frequency of elastic suspension system constant.
Source: Force excitation at top of source (flat spectrum up to 1000 Hz)

Question 1: Should the excitation be mounted on a single 'area spring' or on four elastic 'spring' elements?
Question 2: How can you measure the insertion loss for a machine (excitation) and structure (foundation) configuration without removing the elastic elements (resp. springs)?
Rule Set 1: Impedance of structure should not have a similar behaviour as spring; structure under ‚springs' -> high impedance (best ~ mass-character); spring-elements with small impedance compared to foundation-impedance (mount on several elements) see [3]

Rule Set 2: excite once the machine & once the structure; measure in each case the level difference of the velocity between machine & foundation; the smaller values of the two provides an estimate of the insertion loss [4]

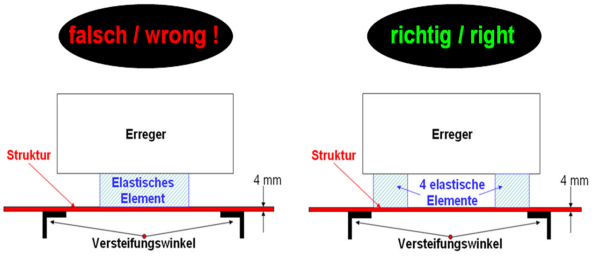

Figure 4: Sketch of Configuration for Scenario II -'spring area' (left) **and 'spring elements'** (right), **based on [3]**

The four 'springs elements' are separately tuned on a stiff foundation to be in agreement with the 'spring area' case. The main interest is the sound power of the structure (plate) only, excluding the structural parts of the excitation ('Erreger', s Figure 4): Figure 5 shows the radiated sound-power L_w(ABN) of structure (plate) only for the three investigated cases, which are in agreement with the mentioned rules.

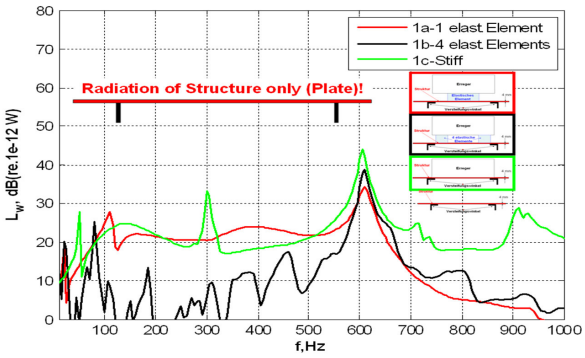

Figure 5: Sound-Power L_w(ABN) radiated of structure only !

(Question2) The level difference for the top excitation fits very well with the real insertion loss (results not shown here), due to the fact, that Z1 (impedance foundation) << Z0 (impedance source ~ excitation); respective considering a mounted 'velocity source',

Scenario III: Source on U-Profile

Task: mount a vibration-exciter on a given structure

Structure: Light-weight, oscillatory framework ('U400 Profil'); partly big, thin plate-components, spring-mounts tuned to 270 Hz (s. Figure 6)

Source: Upper part of the system: low sound-radiation, but high accelerations at mounting points, [3], force excitation at source structure

Question: What mounting position & configuration is best for low noise radiation?

Rule Set : Force excitation at stiff parts - stiff & spring mounted - reduces sound radiation at low frequencies. Force excitation between stiff Parts - stiff & spring mounted - reduces sound radiation at high frequencies (> 1kHz); Using mass is independent of excitation position (for other reasons better placed between stiff parts); Mass increases performance of spring mounting.

For both configurations (X & Y) the radiated sound power (Figure 7) in the frequency region above 2 kHz is clearly reduced by introducing spring and mass. The Y–configuration has in general (there is one exceptional frequency region around 1.5 kHz) lower or at least equal values of radiated sound power compared to the X-configuration. In the end, the best configuration is 'X-Mass' due to the fact that for the Y-Mass configuration the radiated sound power in the frequency region around 1.5 kHz increases. The results are in agreement with the stated rules.

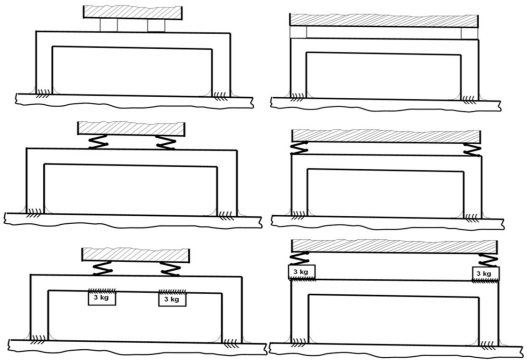

Figure 6: Sketch of Configurations for Scenario III -'X' (left) **and 'Y'** (right);'**Stiff**'- top,'**Spring**' – middle & '**Mass**'-bottom,**[3]**

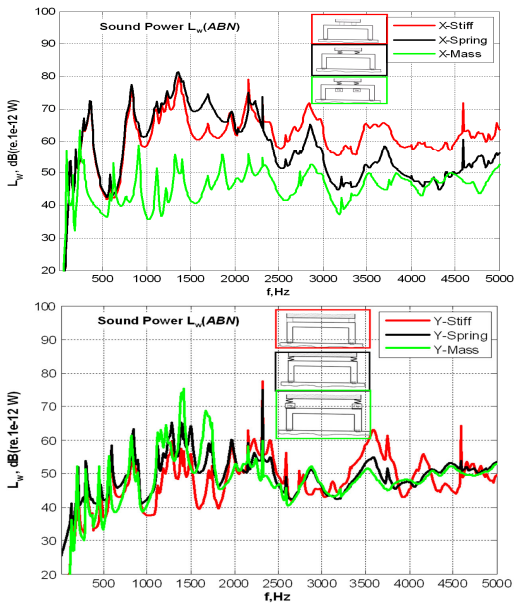

Figure 7: Radiated sound power L_W(ABN) for Scenario III – 'X' (top) and 'Y' (bottom)

Conclusion

Design-rules & rules of thumb are still of value; numerical revisits strengthen the conceptual thinking & intuition; numerical revisits give lots of freedom and opportunities you do not have on a test bench (e.g. consider parts of interest only, which you can not separate in reality); easy check of your own computational performance & lots of examples from literature; simple assessment of your experimental setup & procedure computational effort & time consumption are still high: the choice of computational methods matters. For the daily acoustic engineering work there is a lack of systematic rule collections.

Literature:

[1] Cremer/Heckl – Körperschall, Berlin 1982, Springer-Verlag

[2] Manfred Heckl Die Schwingungs-und Körperschall-isolierung von Maschinen; VDI-Berichte Nr. 291, 1977

[3] K.P.Schmidt: Lärmarm konstruieren -Beispiele für die Praxis-

[4] M. Heckl, lecture notes, "Körperschall" June 1978