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Numerical investigations about Structure-Borne Insertion Loss as applied in the automotive field and a measurement system for its assessment

Claudio Bertolini[1*] **Giulio Ruggeri**[1]

[1] Research and Technology, Autoneum Management AG, Switzerland

ABSTRACT

In the automotive industry, it is common to use Air-Borne Insertion Loss (ABIL) to assess the performance of an acoustical multilayer, while Structure-Borne IL (SBIL) is rarely used, partly because of the lack of a commonly accepted definition and test procedure. This paper first dwells on the definition of SBIL, investigating by means of simple FE simulations some critical and non-obvious aspects related to this same definition, namely the type of SB excitation used and the formula used for the calculation of SBIL. After this, a very short and broad overview of a measurement system suitable for the measurement of SBIL is given.

Keywords: *structure-borne noise, insertion loss, test-methods*

1. INTRODUCTION

Air-Borne Insertion Loss (ABIL) is commonly used in the automotive industry for assessing the performance of acoustic multilayers used as bills-of-materials for the manufacturing of sound package parts. The procedure for measuring the ABIL is well-established: the ABIL of an acoustic multilayer is typically obtained by subtracting from the diffuse-field Air-Borne Transmission Loss (ABTL) of a flat metal plate coated with a flat specimen of the acoustic multilayer the ABTL of the bare metal plate itself. In turn, the ABTL of the metal plate (coated or not with the acoustic multilayer) is commonly measured in a two-chamber facility consisting either of two reverberant chambers (see, e.g., [1]) or of a reverberant chamber and an anechoic chamber (see, e.g., [2]), wherein the two chambers are coupled through an aperture in which the plate is installed.

On the other hand, a quantity analogous to ABIL but for Structure-Borne (SB) excitation is not at all well-established in the automotive sector. Actually, this appears to be true in general and not only in relation to the automotive sector. In fact, even in the technical literature much less attention appears to be dedicated to SBIL compared to ABIL. Classical text books on acoustics discuss the definition of ABTL/ABIL, whereas they provide no mention of analogous quantities for SB excitation. For a definition of SBIL, one can rely on only a few technical papers [3]-[6]. However, at least for what concerns the automotive field, this lack of attention does not seem to be fully justified. In fact, it is known that sound package may have an effect on the interior NVH of a car also in the mid-low frequencies and for SB excitation (see, e.g., [3]). Thus, this paper intends to be a contribution to a reconsideration of the concept of SBIL/SBTL, first elaborating on the (few) definitions/contributions concerning it that may be found in the technical literature and then briefly proposing a measurement tool for its evaluation in an industrial environment.

2. A REVIEW OF SOME CRITICAL ASPECTS ABOUT THE DEFINITION OF SBIL/SBTL

2.1 Definition based on point-mechanical excitation applied directly to the structure

A first definition of SBTL/SBIL is introduced in [4] and recalled, analyzed and used also in [5], in [6] and in [8]. This definition is based on the following experimental set-up: a baffled flat plate clamped along its edge is set in vibration by a mechanical point force applied directly on its surface. The mechanical power Π^{in} input by the force into the plate and the sound power Π^{rad} radiated by the plate are measured. The SBTL is then defined as follows:

$$SBTL = 10 \cdot Log_{10}\left(\frac{\Pi^{in}}{\Pi^{rad}}\right) \qquad (1)$$

Note that this definition, introduced here for the case of a flat plate, can be generalized to the case of a three-dimensional structure in an obvious way. As remarked in [4], Equation (1) represents essentially an "acoustical-

Corresponding author: claudio.bertolini@autoneum.com

mechanical conversion efficiency". Based on this definition, the SBIL of a flat acoustical multilayer is obtained by first evaluating according to (1) the SBTL of the plate with the acoustical multilayer applied on it and then –always according to (1)- the SBTL of the bare plate and eventually taking the difference:

$$SBIL = 10 \cdot Log_{10}\left(\frac{\Pi_{trim}^{inp}}{\Pi_{trim}^{rad}}\right) - 10 \cdot Log_{10}\left(\frac{\Pi_{bare}^{inp}}{\Pi_{bare}^{rad}}\right) \quad (2)$$

This definition of SBIL, while very sensible, presents some aspects that deserve further investigation. One of these aspects is –quite obviously- the possible dependence of the SBIL defined in (2) on the chosen position for the exciting point mechanical force. A second aspect –maybe less obvious- is the fact that the input power Π_{trim}^{inp} in trimmed conditions may be substantially different from the one Π_{bare}^{inp} in bare conditions and –even more- that both may depend on the damping of the structure on which the point force is applied.
It is worth remarking here that, in relation to this latter point, the definition (2) of SBIL is substantially different from the one of ABIL. In the AB case, and using an experimental set-up similar to the one above-described for the case of SBIL, the excitation is an acoustic diffuse-field and the ABIL is defined as:

$$ABIL = 10 \cdot Log_{10}\left(\frac{\Pi_{trim}^{inc}}{\Pi_{trim}^{rad}}\right) - 10 \cdot Log_{10}\left(\frac{\Pi_{bare}^{inc}}{\Pi_{bare}^{rad}}\right) \quad (3)$$

wherein Π_{trim}^{inc} is the *incident* acoustic power of the exciting diffuse field for the trimmed case and Π_{bare}^{inc} is the *incident* acoustic power of the exciting diffuse field for the bare case. Note that the actual acoustic power *input* by the diffuse-field into the plate (which may be different from the *incident* power) does not play any role in (3). Given that, obviously, $\Pi_{trim}^{inc} = \Pi_{bare}^{inc}$, from (3) one has:

$$ABIL = 10 \cdot Log_{10}\left(\frac{\Pi_{bare}^{rad}}{\Pi_{trim}^{rad}}\right) \quad (4)$$

Thus, for AB excitation, the IL is nothing but the (logarithmic) ratio between the radiated acoustic power for the bare plate and the radiated acoustic power for the trimmed case. Which is different from what is done for the SB case according to the definition (2). The substantial difference between (4) and (2) is apparent from the formulas and it is basically due to the impossibility to define in a simple way a concept similar to the "incident acoustic power" for the case of a mechanical excitation.

These aspects of definition (2) for SBIL were investigated by means of some simple numerical simulations. The following test case was considered: a rectangular steel plate having thickness 1mm and dimensions 0.5m x 0.6m coated with a classical "spring-mass system" consisting of a "spring layer" of polyurethane foam with thickness 20mm, on top of which a "mass layer" of an elastomeric material (EVA, Ethylene-Vinyl-Acetate) with an area weight of 3kg/m^2 is placed. This is a very traditional kind of treatment used since decades in the automotive field. Figure 1 shows the FE model used for the investigations, which covered the frequency range between 50Hz and 1kHz. The in-plane size of all the elements is 10mm. The spring foam layer is modeled with 10 elements through its thickness, while the EVA mass layer is modeled with just one element through its thickness. All elements are linear isoparametric. All simulations were conducted using the commercial software Actran [8]. The material parameters used for the various layers are given in Table 1. Furthermore, in order to analyze the dependency of the SBIL according to (2) on the position of the excitation, 10 different excitation points randomly distributed over the surface of the plate were considered, shown in Figure 2.

Figure 3 shows the SBIL for the analyzed spring-mass system, calculated according to the definition (2) for the different excitation points (grey lines), together with the corresponding average (red line). Note that "taking the average" here basically amounts to considering the 10 excitation forces as uncorrelated. In the same Figure 3, also the ABIL for the same spring-mass system calculated using the classical Transfer Matrix Method (see, e.g. [8]) is shown in green. As one can see from Figure 3, the dependency of the SBIL on the excitation point may be rather substantial, in particular in the frequency range around the spring-mass resonance and up to about 500Hz-600Hz. A deeper analysis of the quantities involved in the calculation of SBIL according to (2) shows that this dependence is there both in the input and in the radiated powers. This means that, when applying definition (2), in order to get a smooth curve that may be seen as representative of the vibro-acoustic behavior of the acoustic trim and not dependent on the chosen position for the exciting point force, some kind of averaging over different excitation positions may be necessary.

As far as the comparison between SBIL and ABIL is concerned, results in Figure 3 seem to indicate that the two quantities are rather similar, with the SBIL generally higher than the ABIL, in particular around the spring-mass resonance.

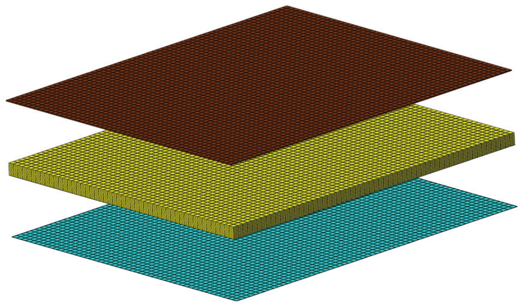

Figure 1. FE model of flat steel plate (cyan) coated with a two-layer acoustic "spring-mass" trim consisting of a foam layer (yellow) and an EVA layer (brown).

Steel	
Young's modulus	206GPa
Density	7800kg/m^3
Poisson's ratio	0.3
Damping Loss Factor	0.03
Polyurethane Foam	
Skeleton Young's modulus	80kPa
Skeleton density	1378kg/m^3
Skeleton Poisson's ratio	0.33
Skeleton Loss Factor	0.18
Air Flow Resistivity	13000Ns/m^4
Porosity	0.979
Tortuosity	1.7
Viscous Length	65.5μm
Thermal Length	114μm
EVA	
Young's modulus	120MPa
Density	1800kg/m^3
Poisson's ratio	0.4
Damping Loss Factor	0.04

Table 2. Material parameters used for Steel, PU foam and EVA

The higher level of Insertion Loss for the case of SB excitation – in particular in the frequency range around the spring-mass resonance- is essentially due to the damping added by the trim to the plate, which is less relevant in ABIL compared to SBIL ([6]). However, as already pointed out in [6], this result must be considered with some care, since the relationship between the ABIL and the SBIL calculated according to (2) may depend quite substantially

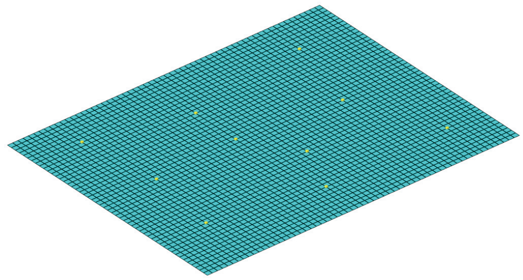

Figure 2. Positions (indicated by yellow dots) of the excitation point mechanical forces

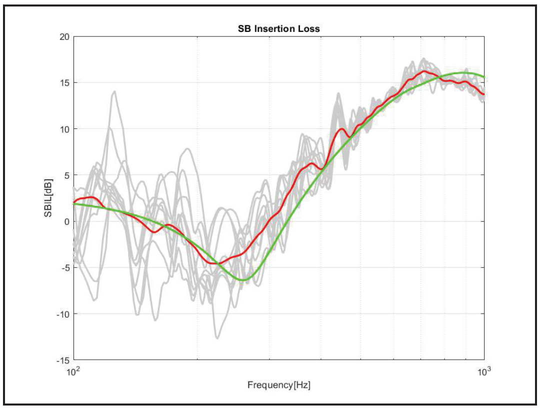

Figure 3. Grey curves: SBIL according to (2) for different excitation points. Red curve: average SBIL. Green curve: ABIL evaluated with Transfer Matrix Method ([8]).

on the damping of the (bare) structure considered, which is the second aspect of the definition (2) that actually deserves some investigation, as previously pointed out. Figure 4 compares the SBIL (averaged over the 10 excitation points) for the case of the plate with 3% damping (red curve, the same of Figure 3) with the one obtained for the case of the plate with a damping set equal to 0.3% (blue curve). In the same chart, also the ABIL is reported in green, similarly to what was done in Figure 3.

Even though the difference in damping considered here (from 3% to 0.3%) may be considered rather extreme, it is evident from the results in Figure 4 that the SBIL calculated according to (2) may be very dependent on the intrinsic damping of the structure or –better- on how relevant the damping added by the trim is, compared to the intrinsic damping of the structure. This is, at least to a large extent, a

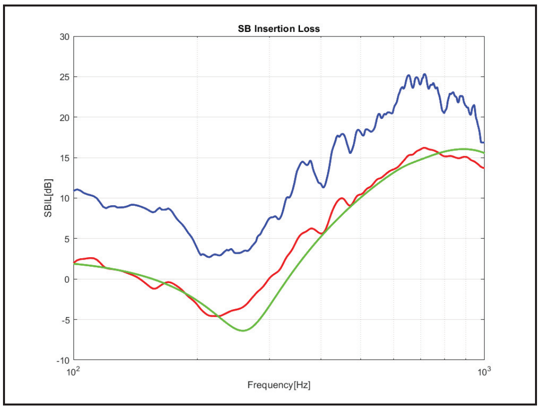

Figure 4. Red curve: SBIL for 3% damping of bare plate. Blue curve: SBIL for 0.3% damping of bare plate. Both curves calculated according to (2). Green curve: ABIL evaluated with Transfer Matrix Method

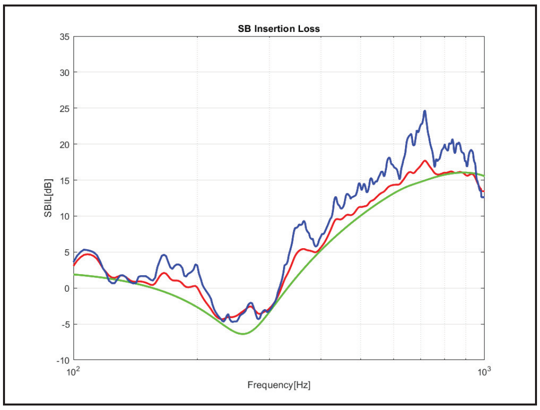

Figure 5. Red curve: SBIL for 3% damping of bare plate. Blue curve: SBIL for 0.3% damping of bare plate. Both curves calculated according to (5). Green curve: ABIL evaluated with Transfer Matrix Method

by-product of the presence of the input power(s) in the definition (2), in particular for the bare case.

Even more than the dependence on the position of the excitation point, this aspect of the definition of SBIL according to (2) may be seen as penalizing for an industrial application of this same definition since, in practice, it may be very difficult to "tune" and keep under control the intrinsic damping of the structure used as a reference for the measurement of the SBIL. As an alternative, one may consider applying to the case of SB excitation a definition formally similar to (4) for the calculation of the SBIL, wherein only the ratio of the output power in the bare case to the output power in the trimmed case is considered, without any reference to input powers:

$$SBIL = 10 \cdot Log_{10}\left(\frac{\Pi_{bare}^{rad}}{\Pi_{trim}^{rad}}\right) \quad (5)$$

By doing so, one simply disregards, when evaluating the performance of the trim vs. SB excitation, the effect that the trim may have on the power input into the plate and looks only at what is the effect on the radiated noise (it is worth noticing that, however, this effect encompasses also the impact that the trim may have on the vibration of the underlying plate). Figure 5 shows a comparison similar to the one shown in Figure 4 when definition (5) is used for SBIL. As one can see when comparing the results in the two Figures, with the definition (5) of SBIL the dependence of this quantity on the damping of the base plate is at least reduced, in particular when one takes into account that -as already mentioned- the difference in damping analyzed here is quite substantial (from 3% to 0.3%). Still, this dependence cannot be considered negligible.

2.2 Definition based on imposed displacement along the boundary of the structure

An alternative proposal for a method aimed at assessing the effectiveness of an acoustic multilayer for SB excitation is – somehow implicitly- given in [3] and in [9]. A similar method is proposed also in [10]. According to the method described in these works, a baffled flat plate is dynamically excited by imposing a vertical and uniform displacement along its boundary, by means of a frame in which the plate is clamped along its boundary. This kind of dynamic excitation will be hereinafter referred to as "boundary excitation". In order for the dynamic excitation to be really uniform along the plate boundary it is necessary that the frame used to transfer the dynamic displacement to the plate boundary behaves, within the frequency range of interest, as a rigid body. In the above-mentioned works, this kind of dynamic excitation is preferred over the direct mechanical excitation of the plate surface by means of point forces as described in previous section, since it is deemed to be more similar to the kind of dynamic excitation automotive panels are normally subjected to.

In order to investigate how to extend to this case the definition of SBIL, the FE model of the plate described in previous section was modified by including a rigid frame all around the plate boundary, as shown in Figure 6 and simulations analogous to those described in previous section, i.e. with bare plate and with plate coated with a spring-mass multilayer, were carried out.

Figure 7 shows the SBIL calculated using definition (2) and a boundary excitation, again for the two different levels of damping of the bare plate already mentioned, namely 3% and 0.3%. From this Figure, two remarks can be made. First of all, the SBIL shows strong "oscillations" in frequency which are obviously due to the modes of the plate (every dip in the SBIL curves appearing in Figure 7 corresponds to a mode of the plate). These oscillations -as one had to expect- are more relevant at low frequency and for low values of the damping of the plate. Secondly, also in the case of boundary excitation, when one calculates the SBIL using (2), this quantity appears to be rather sensitive to the damping of the plate used as a substrate. Similarly to what was done for the case of point force excitation and described in previous section, the SBIL for boundary excitation was eventually calculated using formula (5), both for the case of 3% damping of the plate and for the case of 0.3% damping of the plate. Results are shown in Figure 8.

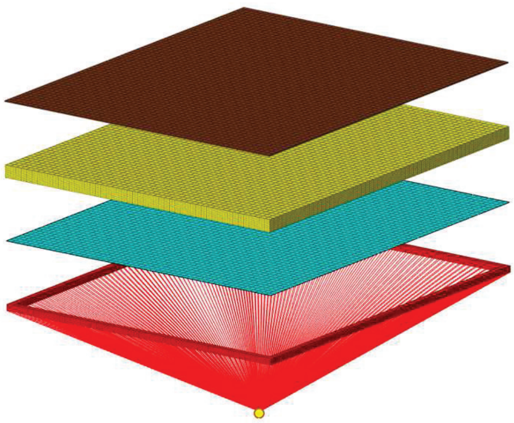

Figure 6. FE model for boundary excitation. Compared to Figure 1, rectangular frame for boundary excitation is added, shown in red. Yellow dot indicates the node where vertical excitation is applied.

Results in Figure 8 can be compared both with those in Figure 7 (in respect of the type of definition used for the calculation of SBIL) and with those in Figure 5 (in respect of the type of SB excitation considered).

These comparisons show that for boundary excitation and when formula (5) is used for the calculation of SBIL, the dependency of this quantity on the damping of the plate used as a substrate is strongly reduced. Actually, one sees from Figure 8 that such dependence is confined to the frequency ranges around the modes of the plate. From Figure 8, furthermore, it appears visually intuitive that,

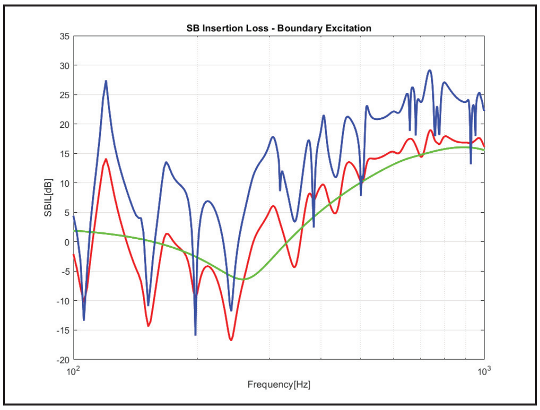

Figure 7. Red curve: SBIL for boundary excutation for 3% damping of bare plate. Blue curve: SBIL for boundary excitation for 0.3% damping of bare plate. Both curves calculated according to (2). Green curve: ABIL evaluated with Transfer Matrix Method

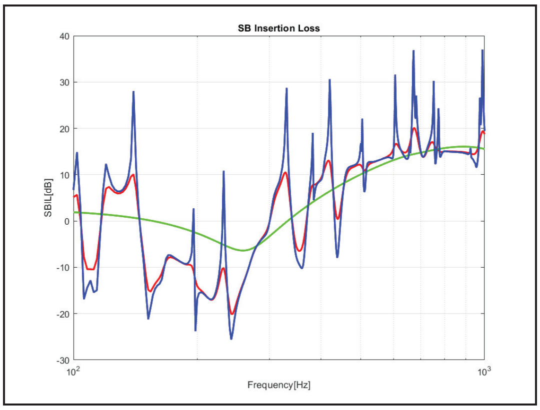

Figure 8. Red curve: SBIL for boundary excitation for 3% damping of bare plate. Blue curve: SBIL for boundary excitation for 0.3% damping of bare plate. Both curves calculated according to (5). Green curve: ABIL evaluated with Transfer Matrix Method

when some kind of frequency averaging is introduced, the two curves may become rather similar. In fact, Figure 9 shows the same comparison of Figure 8 wherein a moving average having a width of +/-30Hz has been introduced. As one can see, already with such a (relatively) small averaging window, the dependency on the damping of the plate underlying the trim is almost eliminated. It is worth noticing that the introduction of some kind of frequency averaging may be seen not just as a way of "combing" the SBIL curves but, at least up to a certain extent, also as a kind of

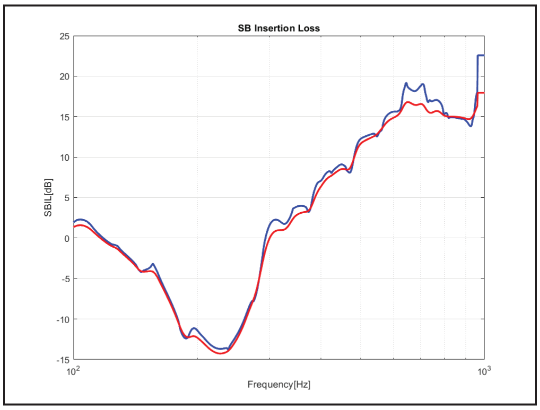

Figure 9. Red curve: SBIL for boundary excitation for 3% damping of bare plate. Blue curve: SBIL for boundary excitation for 0.3% damping of bare plate. Both curves calculated according to (5) and smoothened using a moving average with a window of +/-30Hz

averaging over an ensemble of different plates having, e.g., different thicknesses and/or made with different materials and thus having modes at different frequencies.

2.3 Comparison between SBIL with point force excitation and SBIL with boundary excitation

Eventually, it is interesting to compare the SBIL obtained for the considered spring-mass system using the two definitions (2) and (5) as well as for the two different excitation types described and try to draw some conclusions.

Figure 10 shows the comparison between the SBIL calculated according to (2) for the case of direct point force excitation and for boundary excitation. As one can see, the two SBIL curves are rather similar, with some misalignment in the region of the spring-mass resonance of the trim considered. A similar comparison is shown in Figure 11 for the SBIL calculated according to (5). In this case the two quantities appear to be very similar only above about 300Hz; however, below this frequency the SBIL obtained according to (4) for boundary excitation is much lower than that obtained according to the same formula for direct point force excitation. This is due to the fact that - compared to a random distribution of uncorrelated forces- a *uniform* displacement imposed along the boundary of the plate conforms much better to the vibro-acoustic deformation of a double-wall system such as the one considered here at the spring-mass resonance. As a

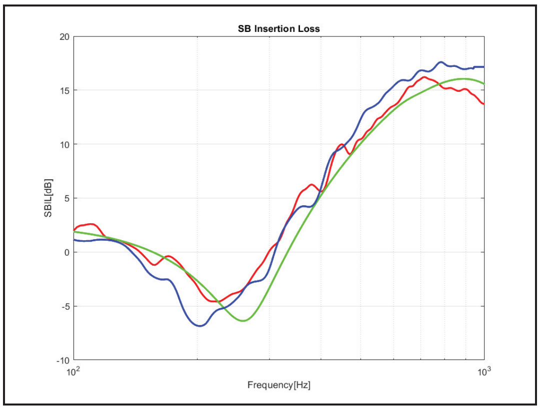

Figure 10. Red curve: SBIL evaluated according to (2) for direct point force excitation. Blue curve: SBIL evaluated according to (2) for boundary excitation (and smoothened with a smoothing window of +/-30Hz). Both curves refer to the case in which the base plate has 3% damping

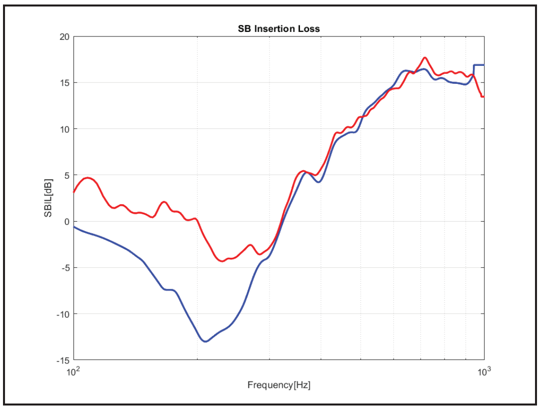

Figure 11. Red curve: SBIL evaluated according to (4) for direct point force excitation. Blue curve: SBIL evaluated according to (5) for boundary excitation (and smoothened with a smoothing window of +/-30Hz). Both curves refer to the case in which the base plate has 3% damping

consequence of this, the spring-mass resonance is excited in a much more efficient way with a boundary excitation compared to the case of a random distribution of point forces over the surface of the plate. This idea was confirmed by the analysis of the sound power radiated by the plate in the bare and trimmed case and for the two different types of excitation, shown in Figure. As one can see, the spring-mass resonance (between 200Hz and 300Hz) is much more strongly excited for the case of boundary excitation (lower part of Figure 12) than for the case of point force excitation (upper part of Figure 12). Thus, the boundary excitation appears to be more capable

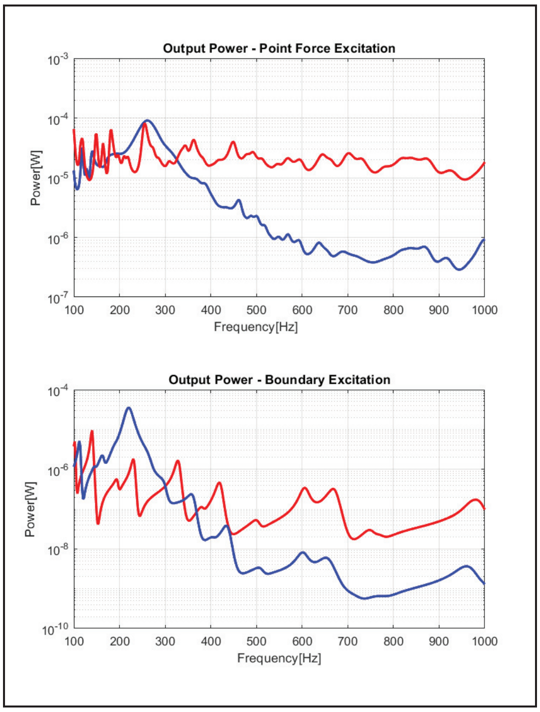

Figure 12. Red curve: sound power radiated by bare plate. Blue curve: sound power radiated by trimmed plate. Upper: point force excitation (average over 10 force positions). Lower: boundary excitation. All curves refer to 3% damping for the bare plate

of highlighting "structural" vibro-acoustical phenomena taking place within the analyzed multilayer, a fact that may be particularly relevant for SB excitation.

2.4 Final considerations

Previous sections have analyzed and compared two different ways of assessing the performance of an acoustic multilayer against SB excitation. All in all, the definition based on formula (5) combined with boundary excitation seems to be the most preferable. This combination is the one that makes the SBIL the least dependent on the damping of the plate used as a substrate. Furthermore, and as previously mentioned, the use of a boundary excitation is generally considered more similar to the typical dynamic excitation vehicle body panels are subjected to in a car. Eventually, it is also clear that from the practical standpoint, this kind of excitation may potentially lead to a faster test procedure, since the SBIL may be measured based only on one dynamic test while, as seen, with the use of direct point force excitation may require several tests (with point forces applied at different points, as in [4]) in order to provide -by averaging- results that do not depend on the chosen position for the applied force.

3. A MEASUREMENT TOOL FOR SBIL

This paragraph provides a very short and broad overview of a measurement tool for the estimation of SBIL, which was designed and prototyped on the basis of all the considerations given in previous section. More details about this measurement tool may be found in a companion paper [11]. The hardware of the tool was developed along the lines described in [10] and it is shown in Figure 14. It comprises a magnesium mounting chassis, which was designed by means of topological optimization in such a way to have its first resonance above 1kHz and thus suitable for tests at least up to 800Hz-900Hz.

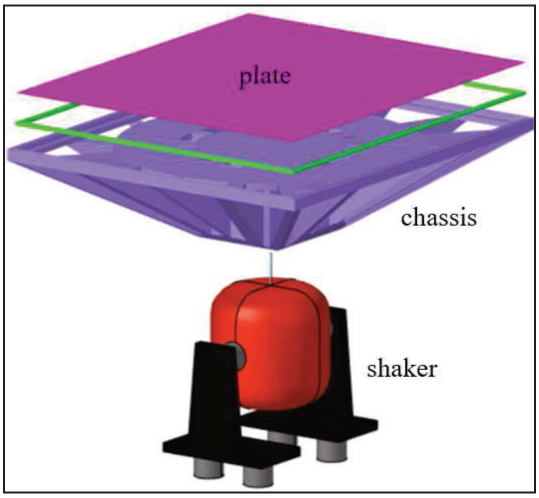

Figure 14. Sketch of measurement system for evaluation of SBIL based on boundary excitation

The mounting chassis is connected at its bottom to an electrodynamic shaker and it comprises in its upper part a rectangular frame in which a plate having approximate dimensions 660mm x 560mm may be clamped. 4 microphones are placed above the plate, at a distance of 175mm from its surface and at positions corresponding to Gauss integration points. From the SPL at these 4 microphones, an estimate of the sound power radiated by the plate may be obtained ([11]). Figure 15 shows an example of IL curves obtained with this tool, for two different types of acoustic trims, one of which is similar in construction to the one described in previous sections. As

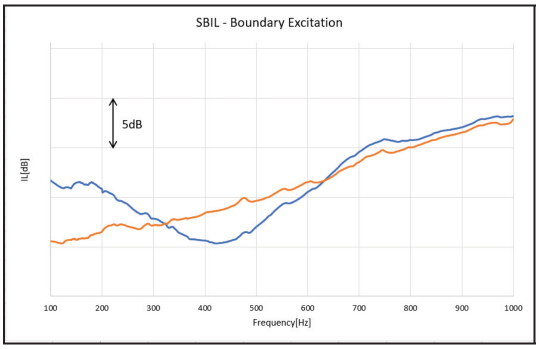

Figure 15. Blue: SBIL for [foam-heavy layer] spring-mass system, thickness 20mm. Orange: SBIL for [felt-film-felt] system, thickness 20mm

mentioned, a detailed description of the measurement tool may be found in [11].

4. CONCLUSIONS

In this paper, a review of the definition of SBIL was first provided, with the purpose of understanding its applicability for the assessment of the performance of flat multilayers normally used for the manufacturing of sound package parts in the automotive field. From the analyses carried out, it appears that a definition based on a boundary excitation and only on the assessment of acoustic powers radiated by the plate (i.e. without any involvement of input powers) is the one that should guarantee the best compromise between some solid scientific background and industrial applicability.

A measurement tool based on this latter definition was also very briefly and broadly presented. More details about this may be found in the companion paper [11].

In perspective, SBIL can be used, in parallel to the already used ABIL, for the development of acoustic multilayers, in such a way to have, in the mid-low frequencies, a better and more comprehensive assessment of their performance.

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