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Sound-transmission measurements on composite and metal fuselage panels for different boundary conditions

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Summary

In the framework of the EU-project FACE, sound transmission measurements have been performed by NLR and KTH on a curved and stiffened composite fuselage panel and an aluminium panel with the same structure. Both panels consist of a part with axial stiffeners and a part, suitable for mounting windows, without these stiffeners. The main goals of the measurements are to provide experimental data for validation of numerical models, and experimental determination of the effect of a.o. panel material on the sound transmission. At NLR, the panels have been suspended on springs, implementing well defined (free-free) boundary conditions, suppressing flanking noise adequately by a special designed panel support structure. At KTH, the panels have been clamped. In spite of the different boundary conditions, the TL data measured by KTH and NLR TL data show a good agreement for 200 Hz and higher frequencies. Due to the curvature and stiffening, the transmission loss of the panels is much lower than the mass law prediction. For frequencies above about 600 Hz, the transmission loss of the composite panel is significantly lower than that of the metal panel, despite its 6% larger mass. For the frequency band of 250 – 2000 Hz, the transmission loss of the window area of the composite panel is much (up to 5 dB) larger than for the stringer area. It seems that the stringers of the composite panel have some bad influence on the sound transmission loss and should be further investigated.

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In the framework of the EU-project FACE, sound transmission measurements have been performed by NLR and KTH on a curved and stiffened composite fuselage panel and an aluminium panel with the same structure. Both panels consist of a part with axial stiffeners and a part, suitable for mounting windows, without these stiffeners. The main goals of the measurements are to provide experimental data for validation of numerical models, and experimental determination of the effect of a.o. panel material on the sound transmission. At NLR, the panels have been suspended on springs, implementing well defined (free-free) boundary conditions, suppressing flanking noise adequately by a special designed panel support structure. At KTH, the panels have been clamped. In spite of the different boundary conditions, the TL data measured by KTH and NLR TL data show a good agreement for 200 Hz and higher frequencies. Due to the curvature and stiffening, the transmission loss of the panels is much lower than the mass law prediction. For frequencies above about 600 Hz, the transmission loss of the composite panel is significantly lower than that of the metal panel, despite its 6% larger mass. For the frequency band of 250 – 2000 Hz, the transmission loss of the window area of the composite panel is much (up to 5 dB) larger than for the stringer area. It seems that the stringers of the composite panel have some bad influence on the sound transmission loss and should be further investigated.

Nomenclature

f_r	=	ring frequency
FRF	=	frequency response function (unit mg/N with g = 9.8 m/s^2 the free fall acceleration)
OB	=	octave band
S	=	test specimen area
S_m	=	measuring surface
SIL_n	=	sound intensity level (in dB re 1 pW/m^2), normal to and averaged over the measuring surface S_m
SPL_{send}	=	space and time averaged sound pressure level in the sending room (in dB re 20 μPa)
TL	=	transmission loss

I. Introduction

The possibility of using composite materials in a fuselage as opposed to traditional aluminium structures has for many years been considered by manufacturers of large commercial aircraft. Not only weight and strength but also acoustic properties of the new materials are of importance. A poor acoustic performance of the material could enforce the addition of sound insulating materials adding weight to the structure thus making the total construction less favourable than a traditional solution. Therefore, knowledge is acquired currently to increase the sound

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insulation of composite fuselage structures, without giving up too much of the reduced mass benefits[1,2,3,4]. In the EU-FP5 project FACE, this issue is one of the research focuses. In these kind of investigations, it is essential to make reliable comparisons between the acoustic properties of "new" and traditional materials. However the comparisons between measurements carried out in different laboratories on different panels can be misleading. The differences between results from two laboratories can be caused by geometrical differences between the rooms and test openings rather than by the test panel itself. The mounting conditions of the test objects are of great importance for the measured results. Due to practical reasons the mounting can vary considerably between various labs. This has been discussed extensively in the literature, Ref. [5] is just one example. Typical aircraft panels are curved and can therefore not readily be mounted in a traditional test opening.

Sound transmission measurements have been performed by NLR and KTH on a composite and an aluminium fuselage panel with the same structure. The main goal of the measurements is to provide experimental data for validation of numerical models, used for sound transmission predictions, presently carried out within the FACE project. Another goal is experimental determination of the effect of panel material and attached damping tape on the sound transmission.

At NLR, the panels have been suspended on springs, free from the surrounding structure. The reason for choosing a free-free set-up is to have well defined boundary conditions, in order to preclude possible difficulties to formulate the boundary conditions correctly in a FEM model. For other boundary conditions, it may be difficult to model the boundary conditions correctly[2]. Flanking noise has been suppressed adequately by a special designed panel support structure. Both TL (Transmission Loss) and modal data have been measured. At KTH, the panels have been clamped.

II. Experimental Methods

Both at NLR and KTH the TL has been measured according to the method described in the ISO standard 15186-1 [6], the TL in dB being determined from

$$TL = SPL_{send} - 6 - SIL_n - 10\log S_m / S \quad [dB] \tag{1}$$

with SPL_{send} the sound pressure level in the sending room (in dB re 20 μPa), measured with a microphone on a rotating boom, SIL_n the sound intensity level (in dB re 1 pW/m^2), normal to and averaged over the measuring surface S_m, and S the area of the test specimen (i.e. the part radiating sound to the receiving room). For the KTH set-up, S and S_m are both equal to the panel surface, and for the NLR set-up these surfaces are both equal to 1 m^2, see section III.

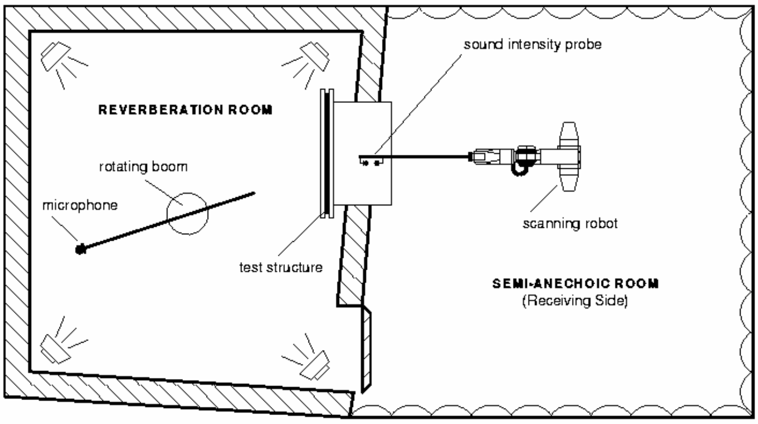

Figure 1. Test set-up for transmission loss measurements on panels.

The modal analysis has been performed with the "Modal Analysis" module of the standard CADA-X software of LMS [7], applying hammer excitation.

III. Experimental Set-up

A. NLR set-up

The NLR test set-up is shown in Fig. 1. The volume of the reverberation room is 33 m^3, resulting in a diffuse sound field for frequencies of about 500 Hz and higher. In order to reduce the measuring error below 500 Hz due to insufficient diffusivity of the sound field, the TL has been determined from successive measurements for three different loudspeaker positions, according to the procedure, described in Annex C of ISO 140-3 [8].

Because the panels have been mounted free from the surrounding structure a special provision has been designed for adequate flanking noise suppression, see Fig. 2. On all four sides of the test opening a U-shaped sound insulating structure is mounted, filled with a sound absorbing foam. The width of the foam (i.e. the green material in Fig. 2 right) is about 21 cm on the upper and lower sides, and about 27 cm on the left and right sides of the test opening. The foam depth (i.e. perpendicular to the panel) is between 75 mm (in the middle) and 140 mm (at the upper and lower sides of the test opening).

Between the panel and the support structure (i.e. the U-shaped structure and the sound absorbing foam, see Fig. 2 left) an air slit is present, which has been minimized to less than 1 mm.

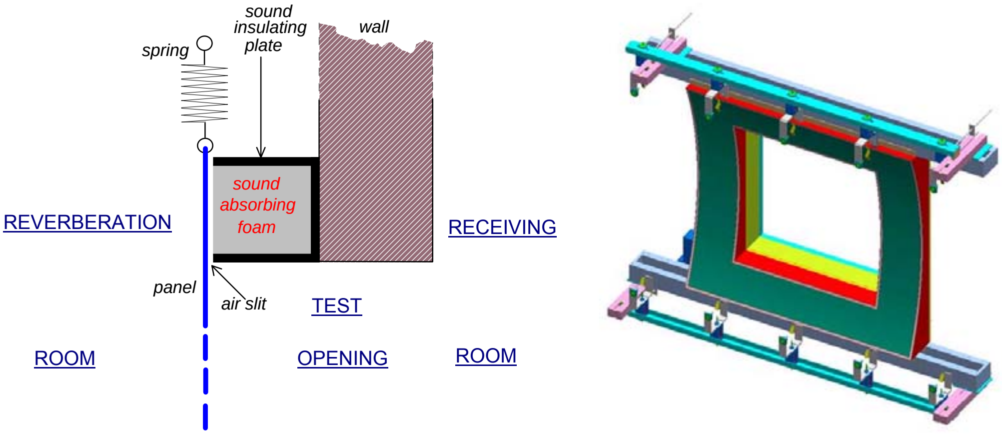

Figure 2. Principle (left) and CATIA picture (right) of the NLR flanking noise suppression structure.

The panel frames have been suspended on springs, see Fig. 3, which have been selected so as to obtain a mass-spring resonance frequency of about 5 Hz.

The 1 m × 1 m test opening (niche) has a depth of about 1 m. The sound power radiated by the panel has been determined from sound intensity measurements over the cross section of the niche, applying a microphone spacing of 12 mm.

The measuring surface of 1 m × 1 m was located 135 mm from the end of the niche (on the side of the receiving room). A (silent) robot has been used for scanning the measuring surface, see Fig. 4. The scanning speed of the robot was approximately 75 mm/s, and the scanning pattern according to the guidelines of ISO 15186-1 [6].

To suppress the effect of reflections on the walls of the semi-anechoic receiving room, having a volume of about 205 m^3, sound absorbing foam has been installed around the test opening, see Fig. 4.

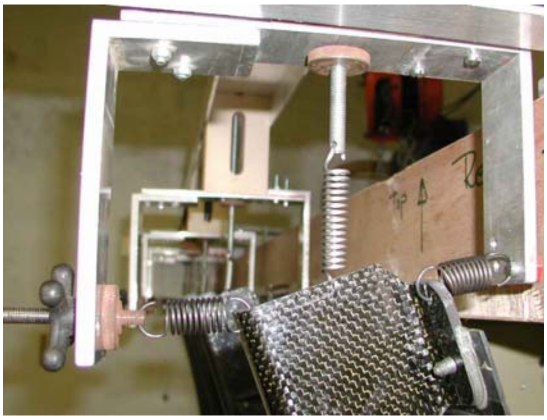

Figure 3. Details of spring suspension.

Figure 4. Scanning robot for intensity probe.

The modal analysis has been performed for the following panel mounting conditions: (A) Set-up on an “isolated panel”, i.e. without the flanking noise suppression structure (shown in Fig. 2). (B) Set-up used for the TL-measurements, i.e. with the flanking noise suppression structure in place.

In both set-ups, hammer excitation has been applied, using the same accelerometer positions.

In set-up A, natural frequencies, modal damping, and mode shapes on the bare panel have been determined from FRF measurements on a grid of 256 excitation points.

In set-up B, and also in set-up A on the panels with damping tape, natural frequencies and modal damping have been determined from FRF measurements, exciting the panel at 12 positions, also applied in set-up A. In this way, the effects of the flanking noise suppression structure and the application of damping tape on the modal damping and possible changes in natural frequencies have been determined.

B. KTH set-up

At KTH the panel under test has been mounted in between a reverberation room (source room, 6.21 m × 7.86 m × 5.05 m) and an anechoic room (receiving room, 7.00 m × 5.95 m × 5.80 m, cut off frequency 80 Hz), see Fig. 5 (pictures at the right). The test set-up is in fully accordance with the ISO standard 15186-1:2000 [6].

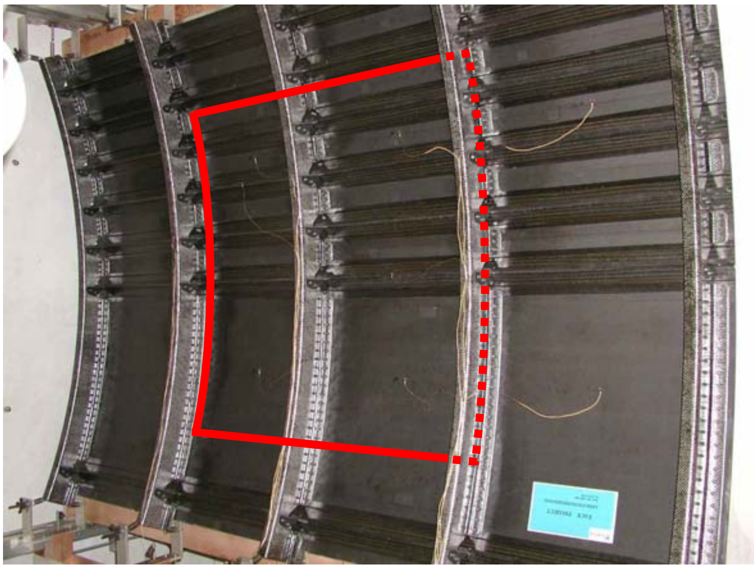

Figure 5. Left: composite panel, mounted in the NLR facility, with the measuring surface indicated in red; Right: composite panel with attached damping tape, mounted in the KTH facility.

C. Reference measurements

Tests on flat panels (1mm aluminium and steel panels) showed very good agreement with the prediction (mass law with the correction for small size, according to Annex A of ISO standard 15186-3 [6]), indicating that the test set-up and the test method is OK for both KTH and NLR. Due to insufficient diffusivity, the NLR results at low frequencies (up to the 160 Hz 1/3 Octave Band, further denoted with OB) showed deviations up to 5 dB from these mass law predictions (in particular the 160 Hz 1/3 OB). Furthermore, intensity measurements near the slit at the receiving side (NLR set-up) showed, that the sound transmission through the slit was negligible, compared with the sound transmission through the panel.

IV. Test Structures

The measurements have been performed on two curved and stiffened fuselage panels with the same structure: a composite and an aluminium panel. Both panels, dimensions about 2.2 m × 1.5 m, having a radius of curvature of 2.0 m, consist of a part with axial stiffeners and a part without these stiffeners, suitable for mounting windows. The mass of the composite panel is about 6% larger than that of the metal panel. Figure 5 shows the composite panel, indicating also the position of the NLR test surface. A constraining layer type damping material has been applied (mass/m^2 about 4% of the panel mass/m^2). At KTH, only the panels with damping tape have been measured. This was due to the fact that it was very difficult to remove the damping tape from the composite panel without damaging the panel. For the composite panel, the skin thickness for the window area is about 50% higher than for the part with axial stiffeners.

V. Experimental Results

A. Modal Analysis

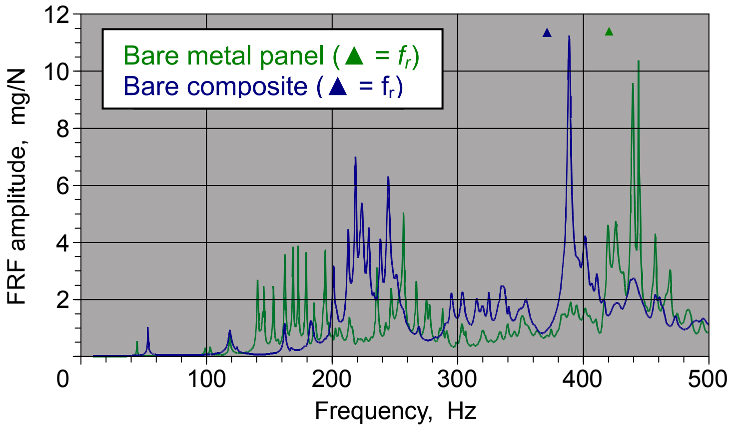

Figure 6. FRF "SUM BLOCKS" of bare panels, for set-up A.

Modal testing was performed on two bare and damped panels, for the set-up A, as described in section IIIA. A large number of natural frequencies, with corresponding mode shapes and modal damping, were identified.

An overview of the modes up to 500 Hz is presented in figure 6. In this figure, the FRF (Frequency Response Function) data is plotted in the form of so called "SUM BLOCKS" of all frequency response function blocks. For the "SUM BLOCKS" picture all selected records are summed and averaged according to their type, see Ref. [7]. Summing the data enhances the resonance phenomena and can help when picking peaks. Note that the FRF is expressed in mg/N, with g = 9.8 m/s^2 the free fall acceleration. Also indicated in Fig. 6 are the ring frequencies f_r = 370 and 420 Hz for the composite and metal panel respectively (adopted from Ref. [9]). Note the presence of strong modes near the ring frequencies.

The mode shapes of the first 4 modes, which are clearly "full panel" modes, are represented in Fig. 7 (see next page). For the first three modes of the composite panel, the natural frequencies are higher than for the corresponding modes of the metal panel by 20%. The higher natural frequencies for the composite panel are probably caused by the (expected) higher stiffness of the composite panel. Assuming that the square of the natural frequency is proportional to the stiffness divided by the panel mass, this stiffness effect can be quantified. From the measured natural frequencies, the stiffness of the composite panel is expected to be about 50% higher than for the metal panel, taking also the extra mass (6%) of the composite panel into account. From Fig. 7 it is not clear whether or not the mode shapes of the last modes (119 and 163 Hz, which differ by a factor 1.4) are identical for both panels. Analysis of higher modes is much more difficult due to the fact that these become more and more local.

In Fig. 8 the FRF amplitude data are plotted for one of the 12 excitation points on the composite panel, applied on both set-ups A and B, for the frequency ranges 10-200 Hz and 200-1000 Hz. Four configurations are plotted: without and with damping tape, both for set-ups A (free hanging suspension, see section IIIA) and B (flanking noise suppression structure in place, see Fig. 2). This figure shows the effect of the panel support structure and the damping tape on the FRF.

For the lower frequencies, shown in the upper plot of Fig. 8, it can be observed that the damping tape has no impact at all. For some frequencies, the damping is even decreasing. A similar decrease, at the same frequencies, was observed for other FRF's. On the other hand, the panel support structure (set-up B) has a considerable effect on the damping. This is caused by the presence of the sound absorbing foam around the edges of the measurement surface, very near to the panel (distance about 1 mm). Moreover, the natural frequencies measured in set-up B are slightly higher than the corresponding frequencies, measured in set-up A. This is attributed to the extra stiffness, induced by the foam, at the panel section adjacent to the measuring surface of 1 m^2 (see Fig. 5 left). This extra stiffness has the same effect as a (very small) change in boundary condition from "free-free" to "simply supported". For the higher frequencies, shown in the lower plot of Fig. 8, in particular for frequencies above 400 Hz, the effect of the panel support structure on the damping is very small. For the higher frequencies, the damping tape causes a significant increase of the modal damping. Assuming no effect of the damping tape on the panel stiffness, a decrease of about 2% is expected in the natural frequencies, due to the extra mass of the damping tape (assuming a proportion between the natural frequencies and $\sqrt{(k/m)}$ with k and m the panel stiffness and mass respectively). A decrease of the natural frequencies in this order of magnitude is indeed observed in the lower plot of Fig. 8, for both set-ups A and B.

For the metal panel, the lower bound of the frequency range, where the damping tape has effect is 150 Hz, where it is 200 Hz for the composite panel. For the rest, the results for the metal panel are similar to those, discussed above for the composite panel.

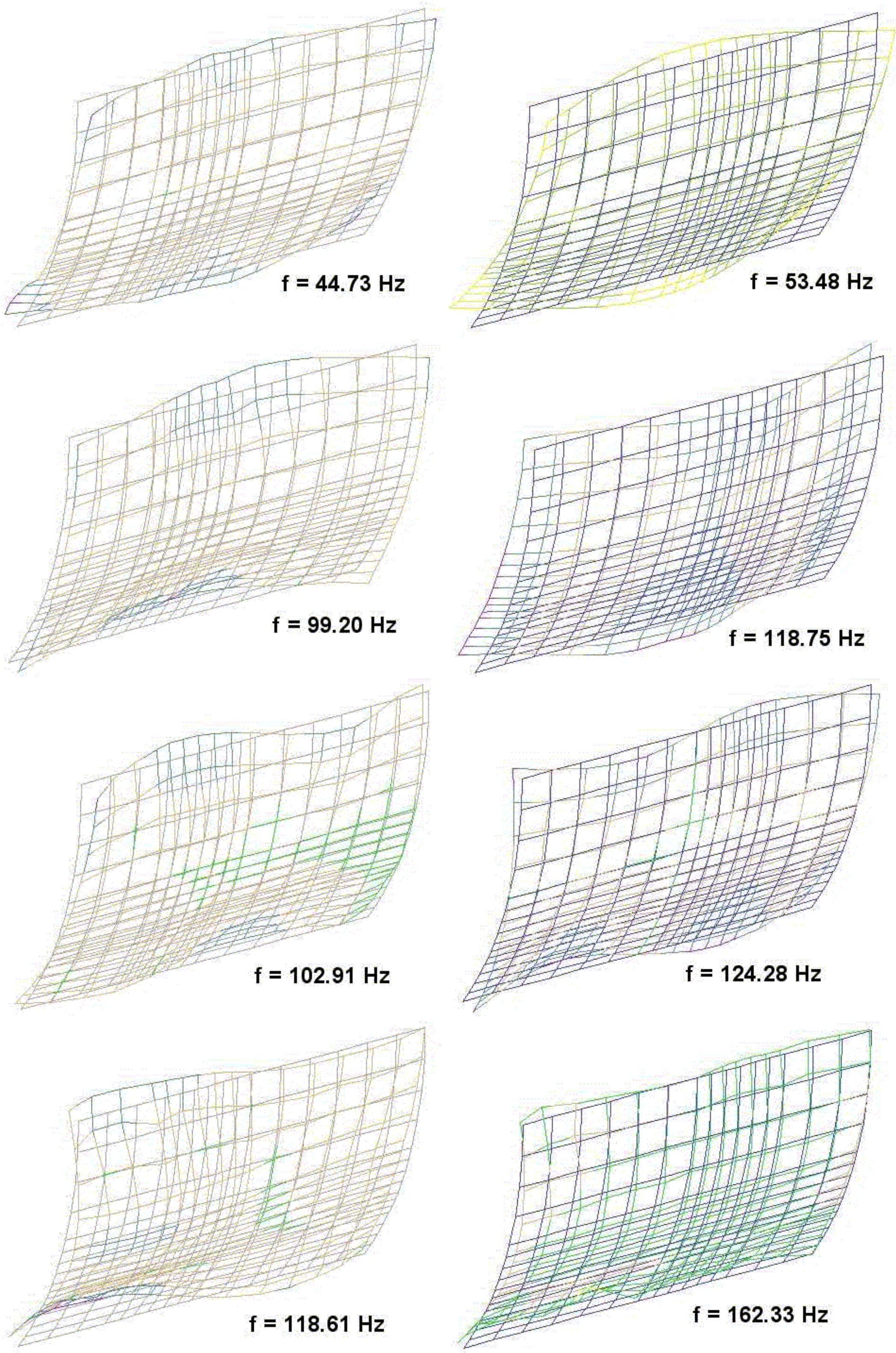

Figure 7. Mode shapes at lower frequencies of bare metal panel (left) and composite panel (right).

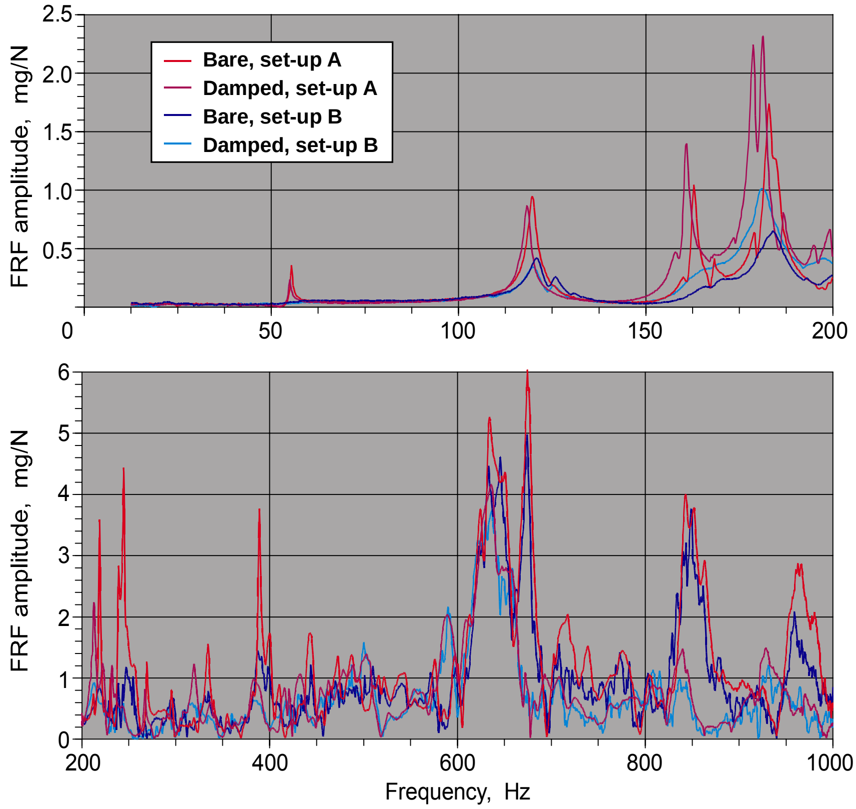

Figure 8. Comparison of effect of panel support structure and application of damping tape on the composite panel, for two frequency ranges.

B. Transmission Loss

Figure 9 shows the TL data, measured by NLR, for both bare panels. For comparison, also the TL, according to the mass law is plotted for the composite panel mass (with included the small size correction, according to Annex A of ISO standard 15186-3 [6]).

For the bare panels, the TL appears to be much lower than the mass law prediction for a limp plate with the same mass per m^2 as the composite panel. For an important part, this is due to the curvature and the stiffening of the panels. In particular a flaw occurs in the TL in the 250 Hz and 200 Hz 1/3 OB (Octave Band), for the composite and the metal panel respectively. The corresponding narrow-band frequency ranges are 210 – 270 Hz and 175 – 235 Hz respectively. This flaw is attributed to the presence of panel modes in the range of 200 to 250 Hz (composite panel) and 140 to 200 Hz (metal panel), see Fig. 6. From the above, it is observed that the frequency band where the modes are situated is somewhat lower than the frequency band where the flaw occurs in the TL, the difference for the metal panel being larger than for the composite panel. It is noted that the natural frequencies of the panel are increased slightly by the presence of the panel support structure (see Fig. 8), but this effect is too small to account for the observed frequency difference between the modes and the TL flaw. Because of the different frequencies where the TL flaw occurs (200 and 250 Hz for the metal and composite panel respectively), and the correlation with the natural frequencies of the panels, it is unlikely that these flaws in the TL are inherent to the

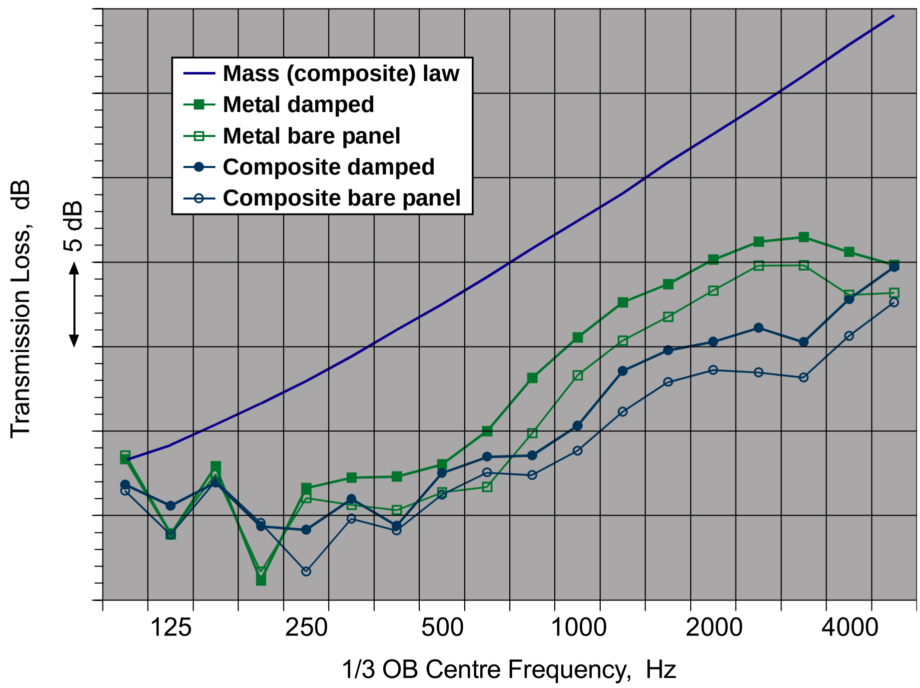

Figure 9. Measured TL data (NLR) of both panels, both bare and damped, in comparison to the mass law.

experimental set-up (e.g. the niche behind the panel, or a non-diffuse sound field in the reverberation room at low frequencies).

Also, modes are present near 400 Hz for both panels, probably connected with the ring frequencies, see Fig. 6, which could be the cause of the (smaller) flaw in the TL at 400 Hz. Starting from the 1/3 OB above the ring frequencies (500 Hz), an increase of the TL data is observed for both bare panels, up to the 1/3 OB's of 2000 Hz (composite panel) and 3150 Hz (metal panel).

Furthermore, for the 1/3 OB's of 800 Hz and higher, the TL of the bare metal panel is larger than the TL of the composite panel, despite its lower mass. The maximum difference is 6.6 dB at the 3.15 kHz 1/3 OB. This difference may be caused by the larger stiffness of the composite panel.

The TL values of the damped configurations are also plotted in Fig. 9. The effect of the damping foil on the TL is about +2 dB on average for both panels, over the 1/3 OB's from 250 – 5000 Hz. Concerning the frequency range, this agrees with the results of the modal analysis, see discussion of Fig. 8, where also an effect of the damping tape is observed for 200 Hz and higher frequencies, for the composite panel. For the lower frequencies (except for the 250 Hz 1/3 OB, where the TL of the bare metal panel is already 4.3 dB higher than the TL of the composite panel), the effect of the damping tape for the metal panel seems slightly larger than for the composite panel. This agrees with the results of the modal analysis, where the damping tape starts to have effect at about 150 Hz for the metal panel, as opposed to 200 Hz for the composite panel. It is noted that the flaw at 250 Hz in the TL of the composite panel disappears (TL increase of 3 dB), after application of the damping tape.

Figure 10 shows the TL data, measured by KTH and NLR on the damped panels. From the KTH results on the metal panel, it is observed that, below the 160 Hz 1/3 OB band, the sound transmission loss decreases when the frequency is increased. After that, the sound transmission loss increases when the frequency is increased. It seems that when the frequency is higher than about 200 Hz (400 Hz for the NLR data), the performance of the panel is similar to a flat plate. Below that frequency, the panel behavior is like a finite shell. For both the KTH and NLR data the coincidence frequency of the metal panel seems to be located around 4 kHz. It seems the composite panel is stiffer at low frequencies. So the low frequency TL is higher (KTH results). However, the coincidence frequency of the composite panel seems also at least one 1/3 octave band lower than that of the aluminum panel. Generally speaking, the acoustic performance of the metallic panel is better.

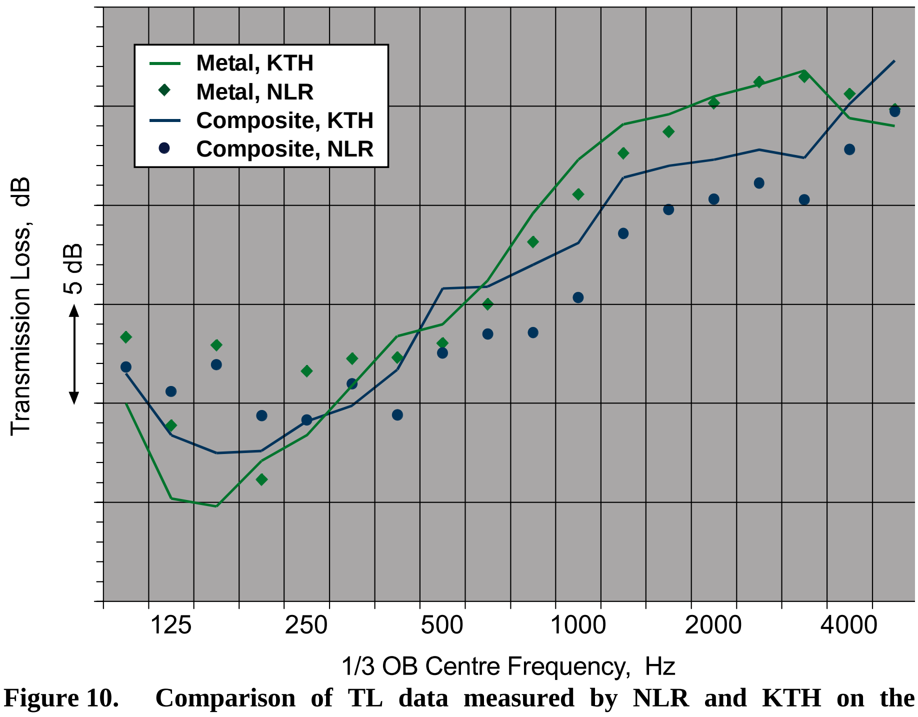

Figure 10. Comparison of TL data measured by NLR and KTH on the damped panels.

Concerning the differences between the KTH and NLR results, for the metal panel these are very small for frequencies at and above 315 Hz. In the low frequency range the discrepancies are due to different mounting conditions of the panels and to different modal densities of the acoustic fields in the source rooms. The thickness of the composite panel varies across its surface. At KTH the sound transmission loss measurements were representative for the entire panel. At NLR the transmission loss of only part of the panel was measured, see Fig. 5 left. The average thickness and thus also the area weight of this part was less than for other parts of the structure. The transmission loss measured at NLR is therefore lower than the transmission loss measured at KTH. For the 1/3 OB of 160 Hz, the niche at the receiving side of the NLR set-up, having a depth of about 1 m, could be the cause of the peak in the NLR TL data, as the niche depth is close to half a wavelength for this 1/3 OB. This peak was also observed in the results on the clamped reference flat plate of 1 mm aluminium, not shown here.

It is also interesting to compare the acoustic performance of the different parts of the panel, which was determined by KTH. Figure 11 compares the sound transmission losses of the area with stringers and the area reserved for windows, for the metal panel. In the frequency region of 160 – 2500 Hz, the window part has a better performance, while the stringer part performs better at the other frequencies, although the difference is not very big. However, the acoustic performance of the stringer area and of the window area shows a big difference for the composite panel (Fig. 12). In some frequency bands, the sound transmission loss of the window area can be over 4 dB higher than that of the stringer area. Besides the higher skin thickness of the window part, vibration of the stringers may be one reason for that.

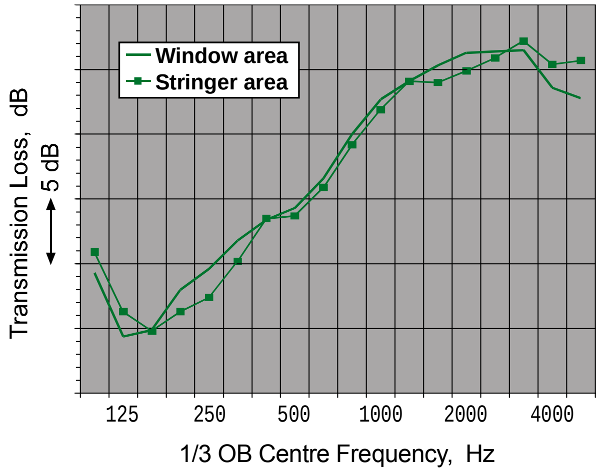

Figure 11. Comparison of sound transmission loss for different parts of the metal panel.

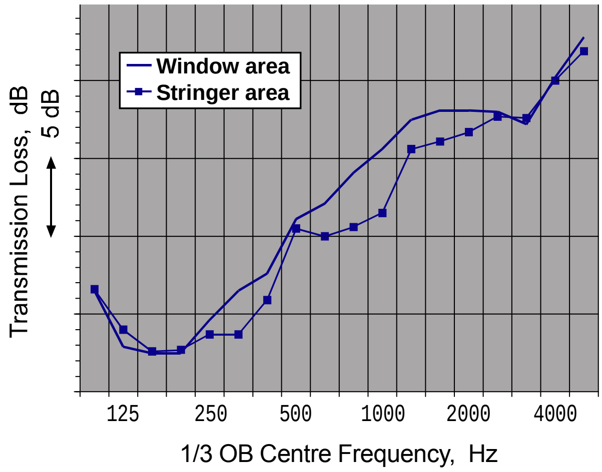

Figure 12. Comparison of sound transmission loss for different parts of the composite panel.

VI. Conclusions

In order to provide experimental data for the validation of numerical models and for assessment of the effect of both the panel material and add-on damping tape on the sound insulation, TL measurements and modal testing on 4 panel configurations have been performed. The most important conclusions are:

- Despite the 6% larger mass, the TL of the composite panel is significantly lower than that of the metal panel for frequencies above about 600 Hz.
- Due to the curvature and stiffening, the TL of the panels is much lower than the mass law prediction.
- The natural frequencies of the lowest modes of the composite panel are 20% higher than the corresponding natural frequencies for the aluminium panel. From this, it is expected that the composite panel is 50% stiffer than metal panel.
- For the composite panel, the transmission loss of the window area and of the stringer area show a big difference: in some frequency bands, the sound transmission loss of the window area can be over 4 dB higher than that of the stringer area. It seems that the stringers of the composite panel have some bad influence on the sound transmission loss and should be further investigated. For the metal panel these differences, being also present, are much smaller.
- On average (over the frequency range of 250 Hz to 5 kHz) the effect of damping tape on the TL is about +2 dB for both panels. For 200 Hz and lower frequencies, the damping tape has no effect. The effect of the damping tape on the TL is confirmed by the results of the modal analysis.
- In spite of the different boundary conditions, the TL data measured by KTH and NLR TL data show a good agreement for the 200 Hz 1/3 OB and higher. Due to variations in the panel skin thickness, the differences for the composite panel are somewhat larger (up to about 3 dB) than for the metal panel. The experimental data are considered to be well suited for the validation of numerical models.
- For low frequencies, up to about 400 Hz, the panel support structure, applied in the NLR set-up for flanking noise suppression, has a large effect on the panel damping. For higher frequencies, this effect is small.

Acknowledgments

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