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**A symmetrical Boundary Element Formulation for
Sound Transmission through Fuselage Walls
- Application -**

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<p>ABSTRACT</p> <p>This paper discusses the results from a numerical analysis of the dynamical behaviour and sound transmission characteristics of a double wall structure, containing the most important components of an aircraft fuselage structure: stiffened skin panel, trim panel connected with vibration isolators to the skin panel, and layers of thermal insulation material in between skin and trim panel. In the presented analysis, which has been performed in the framework of the BRITE/AERO BRAIN (Basic Research on Aircraft Interior Noise) project, a symmetrical acoustical boundary element formulation (ref. 1) was used to model the fluid domains and was coupled to a finite element formulation modelling the structure. This numerical model has been validated by comparing the predicted skin and trim panel responses and transmitted sound intensities with measurements.</p> <p>Two different types of structural damping have been used: a conventional loss factor damping model for the skin panel, while the trim panel damping has been modelled with an AHL (Augmented Hooke's Law) model. The porous thermal insulation material has been modelled by a limp model, in which the stiffness of the fibres of the material is neglected.</p> <p>Four possible configurations with respect to the thermal insulation material were investigated: a cavity completely filled with air, one partly filled with air and partly filled with insulation material, one completely filled with insulation material and one in which the insulation material was compressed. The measurements were carried out by K.U. Leuven and the calculations by NLR. The calculated results show a good agreement with the measurements.</p>			

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A symmetrical boundary element formulation for sound transmission through fuselage walls II: Application

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ABSTRACT

This paper discusses the results from a numerical analysis of the dynamical behaviour and sound transmission characteristics of a double wall structure, containing the most important components of an aircraft fuselage structure: stiffened skin panel, trim panel connected with vibration isolators to the skin panel, and layers of thermal insulation material in between skin and trim panel. In the presented analysis, which has been performed in the framework of the BRITE/AERO BRAIN (Basic Research on Aircraft Interior Noise) project, a symmetrical acoustical boundary element formulation (ref. 1) was used to model the fluid domains and was coupled to a finite element formulation modelling the structure. This numerical model has been validated by comparing the predicted skin and trim panel responses and transmitted sound intensities with measurements.

Two different types of structural damping have been used: a conventional loss factor damping model for the skin panel, while the trim panel damping has been modelled with an AHL (Augmented Hooke's Law) model. The porous thermal insulation material has been modelled by a limp model, in which the stiffness of the fibres of the material is neglected.

Four possible configurations with respect to the thermal insulation material were investigated: a cavity completely filled with air, one partly filled with air and partly filled with insulation material, one completely filled with insulation material and one in which the insulation material was compressed. The measurements were carried out by K.U. Leuven and the calculations by NLR. The calculated results show a good agreement with the measurements.

1. INTRODUCTION

In aircraft industry one of the topics is the reduction of cabin noise. The noise in propeller aircrafts is mainly generated by the propulsion system. They produce a sound field in air that impinges on the outer surface of the fuselage. The vibrations caused by this sound field are transmitted through the fuselage wall and cause interior noise. The fuselage wall consists of a stiffened outer skin connected to frames and a trim panel which is mounted to the frames by means of connectors. The cavity in between the skin and trim is partly filled with thermal insulation and partly filled with air.

The cabin noise can be reduced by changing the double wall structure. For example, by

choosing another type of thermal insulation or by adding a damping layer to the structure. To minimize the time consuming and expensive prototype testing to assess the influence of these modifications, one should have a good insight in the physical phenomena, governing the sound transmission characteristics of the fuselage and a reliable numerical model, capable of predicting the noise levels in the cabin due to an impinging acoustic field on the fuselage. Therefore, the BRITE/AERO BRAIN (Basic Research on Aircraft Interior Noise) project aimed at getting a better understanding of the fundamental mechanisms of noise transmission through double wall fuselage structures. In the first phase of the project, these mechanisms have been identified and quantified in mathematical models, describing the effects of the different mechanisms on the sound transmission. In the second phase, the mathematical models have been transformed into numerical prediction schemes and several solution procedures have been developed. In the final phase, an experimental analysis of the dynamical behaviour and sound transmission characteristics of a validation double wall set-up, containing the most important components of an aircraft fuselage structure, has been performed. In parallel, the prediction schemes and solution procedures have been applied to this double wall set-up and validated by comparing the experimental and numerical results.

This paper compares the experimental results of the validation double wall set-up with the numerical results, obtained from one of the developed solution procedures: a symmetrical boundary element formulation for the fluid domains (ref. 1) coupled to a finite element formulation for the structure. Chapter 2 describes the experimental set-up and motivates the choice of the different components. The numerical model of the experimental set-up is discussed in chapter 3. The results obtained are presented in chapter 4 and compared with the measured results in chapter 5.

2. EXPERIMENTAL SET-UP

2.1 Introduction

The validation double wall set-up contains a stiffened skin and a trim panel, connected to each other by four vibration isolators, and an enclosed cavity between both panels, padded with a certain amount of thermal insulation blankets. Both the vibration isolators and the thermal insulation blankets were provided by an aircraft manufacturer. The whole double wall was mounted in the ground floor of a semi-anechoic room. In this way, sound transmission measurements can be performed by using the rigid-walled enclosure, in the cellar of the room below the double wall, as sending room, while the semi-anechoic room above the double wall serves as receiving room. An inertial shaker, attached to the stiffened skin panel, provide the dynamical excitation of the double wall (a burst random signal with frequency components between 0 and 300 Hz).

In the 150 mm thick, concrete ground floor of the semi-anechoic room, there is an aperture, whose four side walls are covered with a 10 mm thick rigid steel framework. In the cellar of the room, below the aperture (1480 mm x 780 mm x 150 mm), there is an enclosure of 2110 mm x 1400 mm x 1380 mm. All sides of this enclosure are made of concrete. As indicated on the top view of figure 1, the aperture is slightly asymmetrically positioned with respect to the enclosure. The skin panel is a 2.5 mm thick flat aluminium panel (1480 mm x 780 mm), while the trim panel is an 8 mm thick flat plexiglass panel (1380 mm x 680 mm). In the coordinate system, indicated on figure 1, the skin panel is located in the plane $z=0$ and the trim panel is located in the plane $z=100$, while the rigid ground floor of the semi-anechoic room is located in the plane $z=150$. As a result, the cavity between both panels has a height of 100 mm

and is enclosed by rigid side walls of the aperture and by two flexible boundaries: skin and trim panel.

All edges of the skin panel are clamped between two steel bars. The skin panel is stiffened with two L-shaped aluminium beams (see figure 2) in the longest direction. The thickness of the stiffeners (1430 mm long) is 1 mm. The L-shape of the beams is 40 mm x 15 mm. Two appendices (35 mm x 100 mm) are added to both stiffeners to allow the mounting of one vibration isolator on each of the four appendices.

The trim panel is mounted on top of these four vibration isolators, resulting in a mechanical connection of the skin and the trim panel. Due to the smaller dimensions of the trim panel (1380 mm x 680 mm) compared to the cavity (1480 mm x 780 mm), there is a gap of 50 mm between the edges of the trim panel and the cavity. As a result, the trim panel has free boundary conditions along all edges.

Four different double wall configurations have been tested. In the first configuration, the cavity between skin and trim panel was filled with air. In the three other configurations, the cavity was filled with 3, 4 respectively 6 layers of thermal insulation material. As the cavity has a height of 100 mm while each layer of insulation material has an uncompressed thickness of 25 mm, the cavity in the 4-layer-configuration is completely filled without any compression of the layers. In the 6-layer-configuration, the insulation material is compressed to 66% of its uncompressed volume in order to fit in the 100 mm high cavity. In the 3-layer-configuration, the layers are resting on the skin panel, resulting in an air gap of 25 mm between the upper face of the insulation material and the trim panel.

The choice of the different components is motivated by the fuselage structure of a real aircraft. The low frequency (i.e. up to 250 Hz) dynamic response of an aircraft's outer fuselage is mainly dominated by the circumferential modes of the skin - which are determined by the stiffness of the frames - and not by the axial modes of the skin panels in between the frames. Therefore, the skin panel and stiffeners in the experimental set-up were dimensioned to have at least a few bending waves in the stiffeners below 250 Hz, while the number of local skin panel modes in between the frames was kept small. However, due to these requirements, the thickness of the skin panel in the experimental set-up (2.5 mm) is much larger than in a real aircraft (< 1 mm). The modes of the composite inner fuselage of an aircraft (trim panel) have at least twice the number of wavelengths compared to the corresponding skin panel modes and also their modal damping values are substantially higher. Based on these requirements, plexiglass was chosen as trim panel material with free boundary conditions. The thickness of the trim panel (8 mm) was a compromise between a high modal density (the lower the thickness, the higher the modal density) and a pre-loading of the vibration isolators, that should be comparable to the actual loading in an aircraft (the higher the thickness, the higher the pre-loading).

The skin panel displacement was obtained by measuring the acceleration in 45 points on the skin panel. 45 PCB Structcel accelerometers (each weighing 0.0032 kg) were placed on a regular grid of points, distributed over the whole surface of the skin panel.

The displacement of the trim panel was obtained by measuring the velocity with an automatically scanning laser vibrometer. Due to the transparency of the plexiglass trim panel, small pieces of reflective tape were glued on a regular grid of 128 points to obtain reflections of the laser beam on the trim panel.

The radiated sound intensity of the different configurations (zero, three, four and six layers of insulation blankets) was measured by scanning a 1D intensity probe in a horizontal plane at a distance of 300 mm above the trim panel. In order to have measurement data that are directly comparable with the numerical results, obtained from a unit input force modelling, the measured intensity spectrum was divided by the autopowerspectrum of the input force.

Figure 3 shows a detailed view on the double wall test set-up. One can recognize the mounting of the vibration isolators on the trim panel and on the stiffeners of the skin panel, the

pieces of reflective tape on the trim panel to allow laser vibrometer measurements, accelerometers on the skin panel, a pressure microphone in the cavity and three layers of thermal insulation material.

3. NUMERICAL MODEL

3.1 Introduction

In this chapter the numerical model of the experimental set-up is discussed. The structure is modelled with finite elements and the fluid domains with boundary elements. In the next two sections the modelling of both the structure and fluid domains will be discussed. Also a number of remarks are made on the most important aspects concerning the numerical model and experimental set-up, which can cause possible discrepancies.

3.2 Finite element model of the structure

The modelling of the different parts of the structure: skin, stiffeners, connectors and trim panel, are discussed in this section.

Skin panel and stiffeners

The following material properties were used:

		<u>Skin panel</u>	<u>Stiffeners</u>	
<i>Modulus of Elasticity</i>	-	$7.0E+10$	$7.0E+10$	N/m^2
<i>Poisson's ratio</i>	-	0.3	0.3	
<i>Density</i>	-	2850.	2800	kg/m^3
<i>Thickness</i>	-	0.0025	0.001	m
<i>Loss factor</i>	-	0.02	0.02	

The density of the skin panel has been increased by a small amount in order to compensate for the added mass of the accelerometers attached to the panel. Also an extra point mass of 0.025 kg has been added at the position of the shaker to compensate for the mass of the inertial shaker. The mass of the shaker is much higher than this value, but with a two mass - two spring system it can be shown easily that the structure does not feel the mass of the shaker itself. The only mass left, which is felt by the structure, is the mass between the force transducer and the skin panel, which is the 0.025 kg. The force applied on the model consists of a point force of 1 N at the position of the shaker.

The skin panel is clamped along its edges. This condition is very hard to fulfil in reality, which has also been noticed by the KUL in their measurements.

The damping properties of the skin panel are approximated by a mean loss factor based on measured damping values at different eigenfrequencies, which only partly describes the real situation. The damping hereby consists of material damping and damping introduced by the clamping condition.

The skin and stiffeners are modelled with linear quadrilateral shell elements.

Trim panel

The following material properties were used:

<i>Modulus of Elasticity</i>	-	$3.44E+9$	N/m^2
<i>Poisson's ratio</i>	-	0.382	
<i>Density</i>	-	$1202.$	kg/m^3
<i>Thickness</i>	-	0.008	m

AHL parameters:

$\beta_1; \beta_2; \beta_3$	-	$0; 60; 50,000$	rad/s
$\varphi_1; \varphi_2$	-	$5427; 10,025$	N/m^2
$\alpha_{\varphi 1}, \alpha_{\varphi 2}$	-	$1.$	N/m^2
$\mu_1; \mu_2; \mu_3$	-	$17,564; 12,000; 62,665$	N/m^2
$\alpha_{\mu 1}, \alpha_{\mu 2}, \alpha_{\mu 3}$	-	$1.$	N/m^2

The high damping of the plexiglass trim panel is modelled with the AHL (Augmented Hooke's Law) damping model (ref. 2 and 3). The AHL model describing the energy dissipation and the values of its variables strongly influence the response of this panel and should therefore be accurate. The damping model is frequency dependent, which means that in principle for every excitation frequency the model should be evaluated again, thus changing the matrices which describe the trim panel.

The trim panel is modelled with linear quadrilateral shell elements modified such that the AHL damping model is taken into account.

Connectors

The four connectors are modelled by linear beam elements, although their real behaviour is non-linear. The connector properties used are:

<i>Density</i>	-	0.001	kg/m^3
<i>Long.; bend.; tors. stiffness</i>	-	$1000; 4800; 4800$	Nm^2

The connectors have a non-linear behaviour, which means that the increase in deflection due to an increase in force is not a linear relation. This non-linear behaviour would make the model even more "complex", then it already is. However, for small deflections this relation can be approximated quite well by a linearised relation.

The bending stiffness of the connector in both directions is uncertain. This, because the connectors are mounted to the stiffeners by bolts through rubber sealings. The bending stiffness, among others, depends on how well the nut is tightened, which will determine the ability of the connector to move in its rubber sealings. Both mechanisms strongly influence the dynamical behaviour of the trim panel.

The number of elements used to model the different parts of the structure are given in table 1. Figure 4a shows the finite element mesh of the structure. The finite element model consisted of a total of 1172 linear quadrilateral shell elements and 4 linear beam element, which resulted in a total of 5942 degrees of freedom.

Table 1 Number of finite elements used to model the structure.

	Number of elements	Type of element
Skin	30 x 20	Q4
Stiffener + appen.	30 + 2 + 2	Q4
Connector	1	B2
Trim	28 x 18	Q4

Q4 = Linear quadrilateral shell element

B2 = Linear beam element

3.3 Boundary element model of the fluid domains

The fluid domains, consisting of air domains and a thermal insulation domain, are modelled with linear triangular boundary elements.

Semi-anechoic room and enclosure

There is a separate air domain on each side of the double wall structure: the semi-anechoic room and the enclosure. Both are bounded domains. However, to limit the size of the model the side walls and the ceiling of the semi-anechoic room are assumed to be perfectly absorbing, so that the air above the double wall could be regarded as extended to infinity. This has the advantage that only the boundary of the air domain in contact with the structure has to be modelled, instead of the whole boundary, which reduces the number of degrees of freedom considerably. This is a good approximation for the structure, because the volume has dimensions such that the structure only experiences an added mass effect and not an added stiffness effect. The small error introduced, due to the bigger air volume, is a slightly higher added mass effect.

The response of the air domain will be influenced more by this approximation. Especially in the low frequency range, a substantial amount of incident energy is reflected by the walls of the semi-anechoic room. As a consequence, the effect of low frequency standing waves in the semi-anechoic room is not taken into account by the boundary element model.

The influence of the enclosure on the dynamical behaviour of the skin panel has been examined with separate finite element calculations. This analysis lead to the conclusion that the air volume of the enclosure has nearly no influence at all and therefore is not modelled. This can be understood by the fact that the added mass is small compared to the stiffness and mass of the stiffened skin panel.

Also of importance are the eigenfrequencies of the enclosure, which can have an influence on the dynamical behaviour of the structure as well. The first eigenfrequency of the enclosure lies around the 190 Hz and the second around the 236 Hz. Only one enclosure eigenfrequency is located in the frequency range (20-200 Hz) of interest. In some measured response curves a little spike can be seen around the 190 Hz, indicating the unimportance of this acoustic mode on the response of the structure. This also corresponds to results obtained with the finite element calculations. The finite element analyses lead therefore to the conclusion that the air volume of the enclosure nearly has an influence at all and therefore is not modelled.

The following material properties, at room temperature, are used for the air domains:

Speed of sound - 340 m/s *Air density* - 1.2 kg/m³

Thermal insulation

The thermal insulation has strong energy dissipation capabilities. Therefore this material has a strong influence on the dynamical behaviour of the structure. The modelling of the thermal insulation, especially its energy dissipation, thus is very important. As in the case of the AHL damping model used for the trim panel this leads to a model which is frequency dependent.

In the limp model used here the stiffness of the fibre matrix is neglected (ref. 1). Due to the very low stiffness of the fibre matrix compared to the stiffness of the structure, this seems a reasonable assumption.

On the boundaries where the thermal insulation is in contact with the structure, it is assumed that there is no relative motion between the structure and the thermal insulation, which may not be the case.

On the boundaries where the thermal insulation is in contact with the air volume it is assumed that there is a force equilibrium and a continuity of the mass flow over the interface. Deviations from these assumptions will give rise to errors.

The following material properties are used for the thermal insulation domain:

		<u><i>uncompressed</i></u>	<u><i>compressed</i></u>	
<i>Speed of sound</i>	-	290	290	m/s
<i>Air density</i>	-	1.2	1.2	kg/m ³
<i>Fibre density</i>	-	2267.	2267.	kg/m ³
<i>Porosity</i>	-	0.9955	0.9933	kg/m ³
<i>Viscous flow resistance</i>	-	2.3E+4	4.3E+4	kg/(s m ²)

The values for the uncompressed thermal insulation properties have been obtained by measurements.

Cavity

The cavity between the skin and trim panel is filled with air and/or thermal insulation depending on the considered case (see chapter 4). This domain is also fully modelled with boundary elements. The side walls of this domain are in contact with the concrete walls which are rigid. Therefore, the normal derivative of the pressure is zero for nodes in contact with these walls.

The boundary element model can be seen as two separate meshes. One modelling the semi-anechoic room and the other modelling the cavity (the last one, however, can also consist of two separate, but directly coupled meshes, for the air and thermal insulation domain, respectively). The pressure degrees of freedom at nodes lying in the air gap between the trim panel and rigid wall, have to be the same.

The number of elements used to model the different parts of the fluid domain are given in table 2. A distinction is made between the cavity filled with only one fluid (air or thermal insulation) and filled with two fluids (air and thermal insulation). The total number of BEM degrees of freedom is 2153 respectively 3355.

Figure 4b shows the boundary element mesh in the case the cavity is partly filled with thermal insulation and partly with air.

Table 2 Number of boundary elements used to model the fluid domains.

	Number of elements	Type of element
Room	1200	T3
Cavity (one medium)	3200	T3
Cavity (two media)	5600	T3

T3 = Linear triangular boundary element

3.4 Frequency dependency

The thermal insulation model and AHL damping model for the trim panel are frequency dependent. This means that the matrices describing the system are frequency dependent too and should be regenerated for every excitation frequency. Due to computation time, it would be impossible to regenerate the matrices all over again for every excitation frequency. Therefore, the strategy is chosen to divide the frequency range under consideration into a number of intervals and recalculate these matrices again for each interval, using its centre-frequency. In this way a reasonable approximation is obtained regarding the frequency dependency of the problem. In this case the frequency range (20-200 Hz) is divided in 20 intervals.

The thermal insulation model (limp) is more frequency dependent than the trim panel damping model (AHL). This is caused by the fact that the limp model is implemented in the boundary element method. This method introduces exponential functions (by the Green's function) which depend on the frequency, making it strongly frequency dependent. The AHL damping model is implemented in the finite element method and therefore does not suffer from this. When the frequency dependent matrices are obtained with a frequency differing from the excitation frequency, the error will, therefore, be larger for the BEM matrices than for the FEM matrices. Hence, subdividing the frequency range in 20 intervals is accurate enough for the AHL model, but still too coarse for the limp model. This last will result in curves which are not completely continuous, but the results are accurate enough for comparison with the measurements. This can be seen in, for example, figure 9. In this figure little jumps in the response curve can be seen at the boundaries of the different frequency intervals.

4. NUMERICAL RESULTS

Four possible configurations with respect to the thermal insulation material have been investigated: a cavity completely filled with air (zero layers), one partly filled with air and partly filled with insulation material (three layers), one completely filled with insulation material (four layers) and one in which the insulation material was compressed (six layers).

Figures 5, 7, 9 and 10 depict the numerical (solid lines) and experimental (dashed lines) skin and trim panel mean responses for the four double wall configurations.

Figures 6 and 8 depict the numerical (solid lines) and experimental (dashed lines) response for a point on the skin (mid point) and trim panel (corner point), respectively, for the zero and three layered configuration.

The skin results are presented in terms of acceleration (measured with accelerometers) and the trim panel results in terms of velocity (measured with a laser doppler vibrometer). The frequency range is 20 to 200 Hz. The frequency resolution is 0.36 Hz for the numerical results. For the measured results the frequency resolution is lower than 0.293 (depending on the different cases).

The mean response for the measurements is obtained by taking the sum of the square of the amplitude of all the measuring points divided by the number of points. The same is done for the calculated values, taking only the nodal values of the nodes lying closest to the measuring points.

All the results are shown on a linear instead of a logarithmic scale (dB), in order to make a better comparison between the measurements and calculations.

For the three cases containing thermal insulation a sound intensity calculation was performed too (fig. 11). The sound intensities have been calculated at 20 x 20 equidistant points lying in a plane, with the same dimensions as the trim panel, at a distance of 0.3 m above the trim panel. All these results are shown on a logarithmic scale (dB), due to the wide range of values.

All the calculations have been done on a SGI power challenge with four R8000 processors (the program only used one, because it is not paralised yet). To calculate the response in one frequency interval 2.3 cpu-hours are needed; to solve the whole problem, including the sound intensity calculation, 48 cpu-hours are needed. The whole problem, consisting of 7976 elements and 9297 degrees of freedom, could be handled in-core, due to the large amount of internal memory (256 Mb). This shows the large amount of cpu-time needed, even on a very fast large-scale computer, to solve such kind of problem.

5. DISCUSSION

In this chapter the numerical and experimental results for the four different double wall configurations (zero, three, four and six layers of thermal insulation material in the cavity) are compared and discussed.

5.1 Zero layer case

In figure 5a the mean response of the skin panel is presented. In this figure we see that the calculated response at the eigenfrequencies is in general higher than the measured one. This indicates that there is too little skin panel damping present in the model. The calculated response in between the eigenfrequencies coincides very well with the measured one.

Some substantial differences in the predicted eigenfrequencies of the double wall system, with respect to the measurements are observed. Several causes for these differences can be stated. First of all, the inertial shaker induces not only a force perpendicular to the skin panel, but induces also a substantial force momentum due to its rotational inertial (shaker mass is 1.5 kg), although only a point force excitation of the skin panel is taken into account in the numerical model. Secondly, an error is induced by the modelling of the boundary conditions of the skin panel. Although the skin panel boundaries in the model are assumed to be perfectly clamped, it is very hard to establish a perfect clamping condition in an experimental set-up.

The trim panel has more frequencies lying within the interval from 0-200 Hz, than the skin panel. This can be seen from figure 5b, where the response curve for the trim panel shows more

eigenfrequencies than the one for the skin panel (figure 5a). This agrees with the fact that the skin panel has a higher stiffness than the trim panel, somewhat compensated by its higher mass. There is a discrepancy between the measured and calculated position of the eigenfrequencies. The most likely explanation is an error in the stiffnesses of the connectors. As explained in section 3.2 these were uncertain due to the mounting, but also due to its non-linear behaviour. This error has a strong influence on the eigenfrequencies of the trim panel. This also corresponds with results found by calculations in which the stiffnesses of the connectors have been varied.

In figure 6 the responses are presented for a skin point and a trim point, respectively. Similar conclusions can be drawn for the separate points as for the mean responses. These last figures give also the phase information. The phase curve for the skin panel point has a similar shape as the measured one, which is less the case for the trim panel point.

5.2 Three layer case

In figure 7 the mean response of the skin and trim panel are presented. The shape of the calculated skin panel response coincides very well with the measured response. This is true for the resonance frequencies as well as for the response in between it. For some resonance frequencies again the response is somewhat over-predicted, suggesting again too little damping. The trim panel response deviates a little more from the measured results. Especially around the 200 Hz where the response is somewhat too low.

Due to the thermal insulation the overall response of the structure is much lower, as expected, compared to the cavity filled with air. This is correctly represented by the numerical model.

In figure 8 the responses are presented for the same skin and trim point, respectively, as for the zero layer case. From figure 8 we can see that for the skin point the phase is calculated very accurate too. This is less accurate for the trim point, due to the difference in measured and calculated eigenfrequencies.

5.3 Four and six layer case

Similar conclusions as for the three layer case can be drawn here. Again, the shape of the calculated response (fig. 9 and 10) coincides very well with the measured response. The response magnitude and number of eigenfrequencies excited is even less, as expected, due to the larger amount of thermal insulation. The location of the peaks in the measurements shift towards higher frequencies when the amount of insulation material increases, which is not the case for the calculated results. An explanation for this phenomenon might be the fact that the gap between trim panel and side wall of the cavity is closed by the insulation material. Then the behaviour of the cavity will be somewhere in between an open and a closed room, which will have a stiffening effect on the modes with a symmetric shape, shifting the eigenfrequencies upwards. This effect will be stronger for the compressed insulation material, resulting in a larger frequency shift for symmetric skin panel modes.

From the figures 5, 7, 9 and 10 also can be seen that the calculated amplitudes at the resonance frequencies have the same level (except for the cavity completely filled with air) as the measured ones, indicating the correctness of the damping models especially the thermal insulation model, having the largest influence.

Another fact pointing towards the correct modelling is the sharp peak seen around the 30 Hz in the response of the trim panel, regardless of the amount of thermal insulation in the cavity.

The sharp peak is seen in both the measured and calculated results.

From the figures we might conclude that the neglect of the stiffness of the fibre matrix, the limp model, appears to be a good simplification.

5.4 Sound intensity

In figure 11 results are presented for a sound intensity calculation respectively measurement in a plane at a distance of 0.3 m above the trim panel. The measured results are reliable from around 80 Hz. The curves in figure 11b are, from top to bottom: zero, three, four and six-layers, respectively.

From the measured sound intensity spectra (fig. 11b), the following conclusions can be drawn. Although porous materials of the glasswool type, as is the thermal insulation material, are generally assumed to yield only a substantial acoustical damping at frequencies where the acoustical wavelength is smaller or at least of the same order of magnitude as the thickness of the porous layer, the experimental results on the validation double wall show, however, that even below 200 Hz some substantial sound reduction is obtained by the thermal insulation material. A comparison of the three, four and six layer case indicates that filling the cavity completely with insulation material (four layer case) is beneficial in terms of sound reduction compared to an only partly filled cavity (three layer case). On the other hand, compressing the insulation material doesn't yield a substantial extra amount of damping (six layers versus four layers), which is also correctly predicted by the numerical model.

The calculated and measured results do not match. The level of the calculated results is 20 dB or more lower than the measured results. The calculated intensity curve of the three-layered case is even lower than the other two curves for the lower frequency range, resulting in a higher transmission loss than the four and six-layered case, which seems not correct. Also the calculated sound intensity is a fluctuating curve which decreases by increasing frequency, while the measured results are more constant.

Some investigations are still going on to find out whether this large errors are due to some programming errors in the implementation of the solution method or due to physical phenomena which are not taken into account in the model.

One cause of inaccuracy is the way the semi-anechoic room is modelled. As mentioned before, in the frequency range of interest of these investigations, the insulation material of the semi-anechoic room is not perfectly absorbing, although this is assumed in the boundary element formulation. Hence, the radiation damping of the double wall is overestimated by the numerical model. This effect is certainly one of the reasons why the predicted sound intensities have lower levels than then the measured levels, but this can not be the only reason for the large differences.

Another cause of inaccuracy is the difference in area in which the power flow has been evaluated. The measured sound intensities are the averaged values of the intensity, flowing through a surface, 300 mm above the trim panel, covering the trim panel as well as the 50 mm gap between the edges of the trim panel and the rigid side walls of the double wall. The predicted intensities, however, are the averaged values of the power flow through a surface, covering only the trim panel and not the 50 mm gaps around it. As the acoustical power flow through the gaps along the trim panel is not negligible at all, the predicted intensities are much smaller than the actual intensities, as indicated in figure 11.

6. CONCLUSIONS

In the first part of this paper (ref. 1) a method to solve the sound transmission through a double wall structure has been discussed. The finite element method has been used to describe the structure and the boundary element method to describe the air and thermal insulation domains.

In this second part of the paper the results from a numerical analysis on the dynamical behaviour and sound transmission characteristics of a double wall structure, containing the most important components of an aircraft fuselage structure: stiffened skin panel, trim panel connected with vibration isolators to the skin panel, and layers of thermal insulation material in between skin and trim panel, are discussed. The presented analysis has been performed in the framework of the BRITE/AERO BRAIN project.

A symmetrical acoustical boundary element formulation (ref. 1), coupled to a finite element formulation has been validated by comparing the predicted skin and trim panel responses and transmitted sound intensities with measurements.

Due to the complex and frequency dependent matrices, special solution schemes and a lot of cpu-time is needed to solve this type of problems. In order to keep the computational effort within reasons, the system matrices were not rebuilt for every single frequency, but the frequency range of interest (20 - 200 Hz) was split up into several frequency intervals, in which the system matrices were evaluated at the centre-frequency and used for every frequency of the interval. This simplification induces some errors, especially for the boundary element model, as the exponential Green's functions are much more frequency dependent than the finite element model.

Four possible configurations with respect to the thermal insulation material have been investigated: a cavity completely filled with air, one partly filled with air and partly filled with insulation material, one completely filled with insulation material and one in which the insulation material was compressed.

The main conclusions are:

- The limp thermal insulation model, neglecting the stiffness of the fibre matrix, describes the dynamical behaviour of the thermal insulation sufficiently.
- The augmented Hooke's Law (AHL) damping model is accurate enough in modelling the damping characteristics of the structure.
- An accurate modelling of the connectors is a key item in obtaining a correct description of the dynamical behaviour of the trim panel.
- The numerical results compared to the measured data for the skin panel are very accurate in the cases involving the thermal insulation. In all these cases, the magnitude of the response is accurate and even the location of the peaks. Also the phase information is accurate. For the case that the cavity is completely filled with air, the magnitude of the response is over-predicted at some eigenfrequencies, suggesting too little damping present in the model for the skin panel.
- The numerical results compared to the measured data for the trim panel are somewhat less accurate than these for the skin panel. The magnitude of the response is accurate, but the

location of the peaks (and therefore also the phase information) does not coincide with the measured results, probably caused by an incorrect modelling of the connector stiffnesses.

- Although porous materials of the glasswool type, as is the thermal insulation material, are generally assumed to yield only a substantial acoustical damping at frequencies where the acoustical wavelength is smaller or at least of the same order of magnitude as the thickness of the porous layer, the experimental results on the validation double wall show, however, that even below 200 Hz some substantial sound reduction is obtained by the thermal insulation material.
- A comparison of the three, four and six layer case indicates that filling the cavity completely with insulation material (four layer case) is beneficial in terms of sound reduction compared to an only partly filled cavity (three layer case). On the other hand, compressing the insulation material doesn't yield a substantial extra amount of damping (six layers versus four layers).
- The numerical sound intensities do not match the measured results at all. The absolute levels are substantially underestimated. Also the calculated sound intensity is a fluctuating curve which decreases by increasing frequency, while the measured results are more constant. Some investigations are still going on to find out whether this large errors are due to some programming errors in the implementation of the solution method or due to physical phenomena which are not taken into account in the model.

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