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FINITE ELEMENT MODEL UPDATE OF AN AIRCRAFT INTERIOR LINING PANEL

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ABSTRACT

In aviation, new propulsion concepts and aircraft configurations are explored to reduce the carbon footprint. Turboprop engines experience a comeback in hydrogen powered aircraft concepts. Compared to conventional aircraft with jet engines, there are changes in noise sources and vibro-acoustic transfer paths. The calculation of these transfer paths should therefore be carried out at the preliminary design phase of a new aircraft to assess the noise exposure of the passengers.

DLR developed a process to automatically generate finite element (FE) models of an aircraft fuselage based on preliminary design data. These models enable the calculation of noise transmission from the source to the passenger's ears. For the enhancement and validation of model parameters, parts with a major influence on the noise transmission are analyzed. This article presents such a process for a commercial aircraft lining panel. The component is subjected to a modal testing with free-free boundary conditions. Experimental data of several tests are composed and modal data are extracted. Based on a 3D scan of the lining panel a FE model is set up. A model update process with ANSYS® OPTISLANG improves numerous geometric and material parameters.

Keywords: *experimental modal analysis, model update, multi-objective optimization*

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Figure 1. Lining panel; left: condition as supplied; right: test condition.

1. INTRODUCTION

Finite element models are powerful and widely used tools to study and predict the static and dynamic behavior of structures. Nevertheless, the models often require simplifications and their predictions are only as accurate as their parameters. Particularly when modeling real structures, there is often a lack of knowledge of the exact parameter values or geometric details. To address this challenge, model updating has emerged as a valuable approach. Integrating experimental data into the work flow of FE model generation refines and improves the quality of simulations. The subject of model updating has become one of the most demanding applications for modal testing in the past decades [1–3]. Numerous methods exist to update FE models which can basically be grouped into two major types. First, the direct matrix methods which directly adjust the system matrices. Secondly, there are indirect methods that manipulate the physical parameters of the model. In this article, the indirect method for updating the material and geometric parameters is the method of choice. This approach not only minimizes the number of parameters required but also ensures that each parameter has a clear and direct physical interpretation.

The goal of this research is to improve the accuracy and precision of noise transmission calculations by refining sub-models. The subject under test is a sidewall lining panel of a commercial aircraft with dimensions 1.05 m×1.22 m (W x H). The panel is supplied as shown in Fig. 1. The glass fiber insulation and the windows, including the shades, are removed for the experiments. The right part of Fig. 1 shows the lining in test condition.

Experimental modal analysis is used to study the dynamic behavior and extract the modal parameters in the target bandwidth from 0 to 200 Hz. In this range the lining panel exhibits a low modal density and shows a dominant modal vibration behavior. The FE model to be updated is created with a reverse engineering approach. Since neither geometric data nor material parameters are available, the model has to be setup from the scratch. In a subsequent optimization process the modal data from experiment and simulation are matched in order to adjust selected parameters.

2. EXPERIMENTAL MODAL ANALYSIS

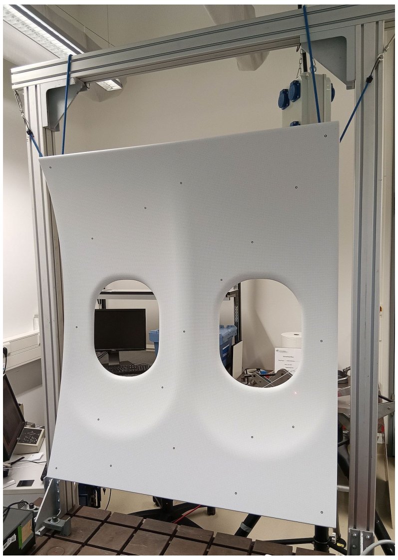

Figure 2. Lining panel in the test rig.

Experimental modal analysis provides a detailed description of the response of a structure. It consists of two main steps, the structural testing and the subsequent modal analysis. Within the structural testing frequency response functions (FRF) from a driving point to selected

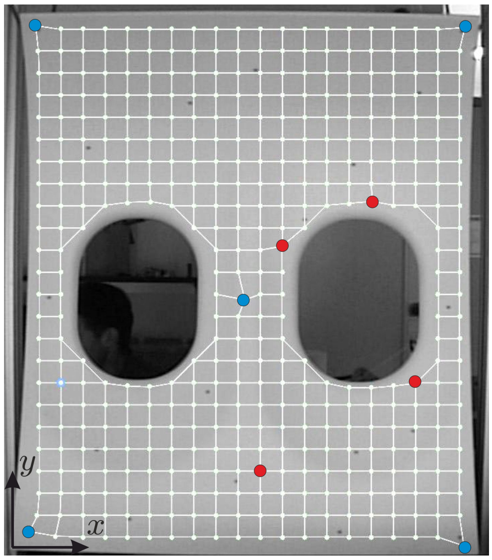

Figure 3. Grid for LSV measurements; white dots: scan points; blue dots: reference points; red dots: shaker excitation points.

degrees of freedom (DOF) are recorded. The modal analysis extracts the modal parameters from the observed FRF.

To study the dynamics of the lining panel without the influence of any supports, a so-called free-free boundary condition is established. For this purpose, a frame of aluminum beams is mounted on a vibration-isolated seismic mass. In this test rig the lining panel is suspended by using two elastic cords connected to the two upper hooks on the backside, see Fig. 2.

The panel is excited on the backside of the panel by an electrodynamic shaker of type LDS® V201. The force transducer of type PCB® 208B01 is wax-mounted and connected to the shaker by an aluminum stinger. To ensure that all modes of interest are excited and observed during test, the shaker is sequentially placed at four different locations. The excitation is a pseudo random signal with a bandwidth from 0 to 500 Hz.

The structural response is determined with a laser scanning vibrometer (LSV) POLYTEC® PSV 200. A grid of 402 scan points with a horizontal and vertical spacing of approx. 50 mm is applied to the surface while the window openings are omitted. Figure 3 shows the scan points and the shaker locations. Additionally, five scan points are used as reference points to fit the different meshes in the following work flow. At each scan point the normal sur-

Table 1. Modal data; undamped eigenfrequencies f_{exp} and damping ratios δ.

f_{exp} in Hz	δ in %	f_{exp} in Hz	δ in %
9.7	3.93	106.6	0.46
26.4	0.51	124.4	0.43
30.4	0.84	139.9	1.19
40.2	0.35	144.1	0.38
60.1	0.64	151.2	0.52
62.4	0.80	161.6	0.51
83.7	0.38	173.1	0.50
85.5	0.46	178.0	0.63
97.5	0.41	196.3	0.56

face velocity is captured and the FRF from the excitation force is calculated. Finally, a mobility matrix with four inputs and 402 outputs is available for modal analysis.

The modal analysis is performed with XMODAL III, a software from the Structural Dynamics Research Laboratory of the University of Cincinnati. Within a bandwidth from 8 to 200 Hz a number of 18 modes is identified using the complex mode indicator function (CMIF). Table 1 summarizes the identified poles.

3. MODEL SETUP

3.1 Geometry

Since a geometric model of the lining panel was not available, a reverse engineering approach had to be carried out. A ZEISS® ATOS 5 optical 3D scanning system provided a detailed triangulated surface model with 500k nodes. Post-processing with CATIA® reduced the complexity and transformed it into a shell model which can easily be imported into ANSYS® Workbench. Additionally, the coordinates of the five reference points of the LSV grid are determined for alignment of the different meshes.

3.2 Materials and layup

The lining panel is made of a flexible honeycomb core sandwiched between two composite laminates. Neither layup nor material data are provided. A sanding of small

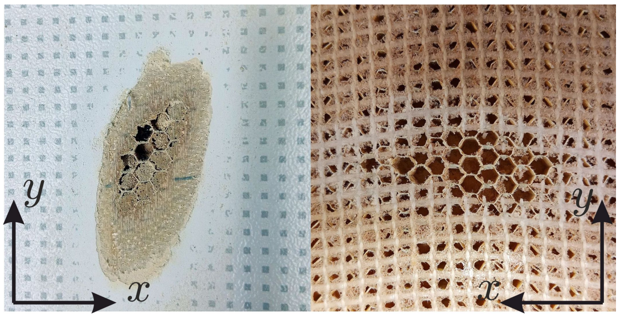

Figure 4. Sanded surface of the lining panel; left: front side; right: back side.

Table 2. Layups from front to back.

Section I	Section II	Initial thickness in mm
Decorative foil	Decorative foil	0.2
Glass weave	Glass weave	0.5
Honeycomb	Honeycomb	5.0
Glass open w.	Glass open w.	0.55
	Glass weave	0.5

areas on the front and the back side of the panel reveals the layers and the orientation of the honeycomb core, s. Fig. 4. A distinction is made between two main layup sections in the model generation. Figure 5 shows these sections. In section I the layup from front to back consists of a decorative foil, a glass fiber weave, a honeycomb core and a glass fiber open weave. The layup in section II has an additional glass fiber weave for stiffening the edges on top of the open weave in the back. The thicknesses of each are approximated by measurements with a micrometer gauge. The 0°-direction of the layups corresponds to the global x-direction. For honeycombs, a distinction is made between ribbon direction (L) and the perpendicular (W) direction. Figure 4 reveals that L is parallel to the 0° direction. The orientations of all weave layers are set to 0°. The decorative foil is defined as an isotropic material. Table 2 summarizes the layups and the initial thicknesses.

Since no material data are provided, realistic assumptions for initial values must be made. The initial material properties utilized in this article are drawn from a dataset

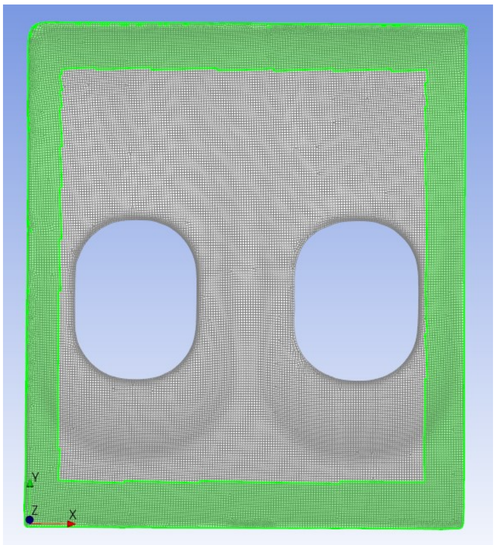

Figure 5. Layup sections; gray: section I; green: section II.

originating from a series of previously conducted experiments involving honeycomb panels from similar applications.

3.3 Masses

Especially in acoustic FE simulations the correct mass of structural parts is necessary for transmission loss calculation. The mass of the lining panel under test conditions, shown in Fig. 1, has been measured at 2.7 kg. In addition to the mass of the laminate, it is essential to consider the relevant point masses that contribute to the overall weight distribution of the assembly. Specifically, these include the masses of the two upper hooks and the two lower fasteners. The removable parts of the hooks are weighed while the masses of the not removable parts of the hook and the fasteners are estimated. A single hook therefore has a mass of 27 g and a single fastener 15 g.

3.4 FE model

The surface model created in Section 3.1 is imported in ANSYS® Mechanical and meshed with approx. 33k shell elements of type SHELL181. The layups are realized with the ANSYS® Composite PrepPost (ACP) module. As in the previous experimental modal analysis, a free-free boundary condition is implemented. Damping is intentionally neglected in this model, as nearly all identified modes exhibit low damping ratios that fall below 1 %, compare Tab. 1.

4. MODEL UPDATE

The objective of the following model update procedure is to enhance the initial material and thickness parameters of the lining panel. Several approaches exist to achieve this goal. In this article the update based on the comparison of modal parameters of the experiment and the FE model is chosen. Within a modal analysis of the FE model the undamped eigenfrequencies and the corresponding mode shapes are calculated neglecting the rigid body modes. A total of three criteria is used for the comparison and the subsequent optimization:

1. Accuracy of mode shapes using the modal assurance criterion (MAC) [1,4]
2. Accuracy of eigenfrequencies
3. Accuracy of lining mass

The first and the second criterion are determined within the ANSYS® modal analysis using the noise, vibration and harshness (NHV) toolkit [5]. After an automatic mode pairing the following combined criterion C_{12} is calculated for N modes:

$$C_{12} = (1-\alpha)\sum_{n=1}^{N}|M_{nn}-1| + \alpha\sum_{n=1}^{N}\frac{|f_{\mathrm{sim},n}-f_{\mathrm{exp},n}|}{f_{\mathrm{exp},n}} \tag{1}$$

The first term benchmarks the MAC criterion by calculating the distance of all diagonal entries of the MAC matrix $\underline{\underline{M}}$ to the maximum achievable value of 1. The second term assesses the accuracy of the paired eigenfrequencies f_{sim} and f_{exp}. Both terms are combined with a weighting factor α which is set to 0.5 in this study.

The mass m of the FE lining panel is evaluated with a quadratic criterion against the real mass determined in Section 3.3:

$$C_3 = (m - 2.7\,\mathrm{kg})^2 \tag{2}$$

The model update or optimization process is divided into two parts. First, a sensitivity analysis is performed to evaluate the impact of material parameters and thickness variations on the criteria C_{12} and C_3. This assessment aims to identify the most sensitive parameters and determine their relative importance. As a second step, parameters of low importance are excluded from the process. Optimization starts with a precise set of selected parameters that have the most significant influence on the behavior of the system .

4.1 Initial parameter set

The initial parameter set for the first step of the optimization process consists of selected material parameters and thicknesses of the layers from Tab. 2 is:

- Decorative foil

  P_1 Young's modulus E
  P_2 Density ρ
  P_3 Layer thickness t

- Glass weave

  P_4 Young's modulus E_x
  P_5 Young's modulus E_y
  P_6 Poisson's ratio ν_{xy}
  P_7 Density ρ
  P_8 Layer thickness t

- Honeycomb

  P_9 Young's modulus E_z
  P_{10} Density ρ
  P_{11} Layer thickness t

- Glass open weave

  P_{12} Young's modulus E_x
  P_{13} Young's modulus E_y
  P_{14} Poisson's ratio ν_{xy}
  P_{15} Density ρ
  P_{16} Layer thickness t

4.2 Sensitivity analysis

The first step of the model update is the sensitivity analysis performed by OPTISLANG. For this purpose, a metamodel of optimal prognosis (MOP) is identified by calculating the criteria C_{12} and C_3 at numerous data points in the parameter space of $P_1 - P_{16}$. The purpose of the approximated MOP is to predict the criteria as a function of the parameters. The convergence criterion for the MOP is the coefficient of prognosis (CoP) [6]. The CoP is a measure of the model's prediction error which is estimated using in cross validation procedures. The CoP that could be achieved for the two criteria within this study are $\text{CoP}_{C12} = 72\,\%$ and $\text{CoP}_{C3} = 99\,\%$. The nearly optimal value of 99 % for criterion C_3 shows that all parameters which affect the mass of the lining are involved.

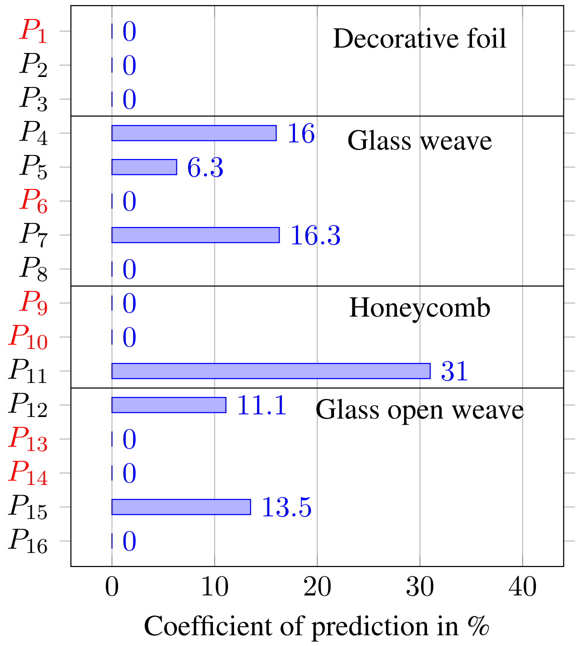

Figure 6. CoP for criterion C_{12}.

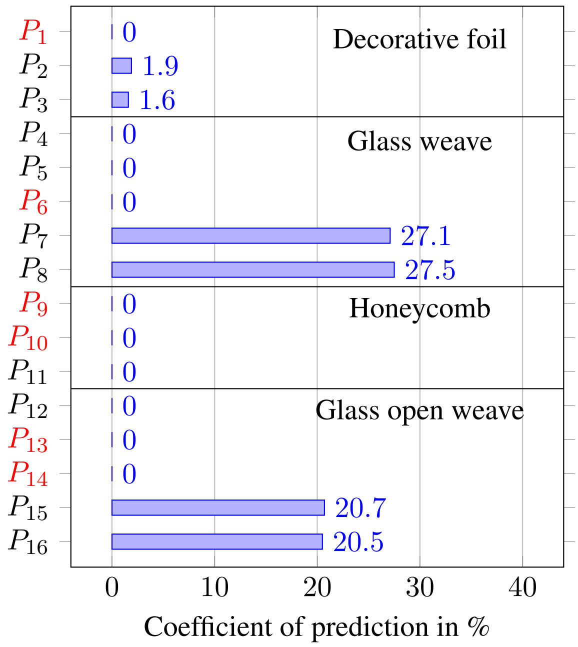

Figure 7. CoP for criterion C_3.

This seems obvious since the densities and the thicknesses of all layers are included. In contrast, the CoP of criterion C_{12} drops significantly with a value of 72 %. This leads to the conclusion that not all relevant parameters to predict C_{12} are covered within the initial parameter set. C_{12} is a mixed criterion of the MAC criterion and the accuracy of eigenfrequencies. In addition to the material parameters and thicknesses, the mode shapes are highly dependent on the geometry of the lining. Since the shape of the lining's surface is fixed, the remaining initial parameters influence the mode shapes and the eigenfrequencies but cannot cause ground-breaking changes.

As mentioned above, the aim of this section is the identification of relevant parameters to achieve the given criteria. Therefore, the CoP values for C_{12} and C_3 are determined for each parameter. Relevant parameters achieve a CoP of more than 1 % for at least one of the criteria. In Fig. 6 and 7 the results are summarized. CoP values below 1 % are set to zero. It is obvious that six parameters (P_1, P_6, P_9, P_{10}, P_{13}, P_{14}) highlighted in red are neither relevant for C_{12} nor for C_3. They can be safely disregarded and excluded from the following optimization. Their values are set to initial values.

4.3 Optimization

After identifying the six non-relevant parameters, attention turns to the remaining 10 out of 16 parameters that can be optimized for improved model accuracy. The multi-objective optimization is conducted with the One-Click Optimization (OCO) of OPTISLANG. It uses the parametrization of the MOP to speed up the optimization process without incorporating any FE solver of ANSYS® Mechanical. As in the sensitivity analysis, C_{12} and C_3 are the optimization criteria. After a maximum number of 4k design evaluations a Pareto front is extracted. Selected designs from this front are validated by modal analyses of the FE model. Finally, the parameter set in Table 3 is chosen.

In the following, the results for the three criteria are presented in detail. The first term of C_{12}, the accuracy of the mode shapes, is visualized with the MAC matrix in Fig. 8. All of the 18 modes identified within the experimental modal analysis are paired with modes of the simulation. The MAC matrix is diagonal dominant. It represents a consistent set of mode shapes with the majority of values exceeding 0.7. A negative outlier is the MAC value of 0.37 at experimental frequency 60.1 Hz. It is a result of a non-symmetric mode shape that cannot ap-

Table 3. Final values for parameters; red: excluded (constant) parameters.

Decorative foil		
P_1	E	5.0 GPa
P_2	ρ	1395.5 kg m^{-3}
P_3	t	0.18 mm
Glass weave		
P_4	E_x	25.9 GPa
P_5	E_y	23.5 GPa
P_6	ν_{xy}	0.139
P_7	ρ	1664.8 kg m^{-3}
P_8	t	0.45 mm
Honeycomb		
P_9	E_z	60.0 MPa
P_{10}	ρ	29.0 kg m^{-3}
P_{11}	t	4.90 mm
Glass open weave		
P_{12}	E_x	24.5 GPa
P_{13}	E_y	25.4 GPa
P_{14}	ν_{xy}	0.126
P_{15}	ρ	1892.4 kg m^{-3}
P_{16}	t	0.50 mm

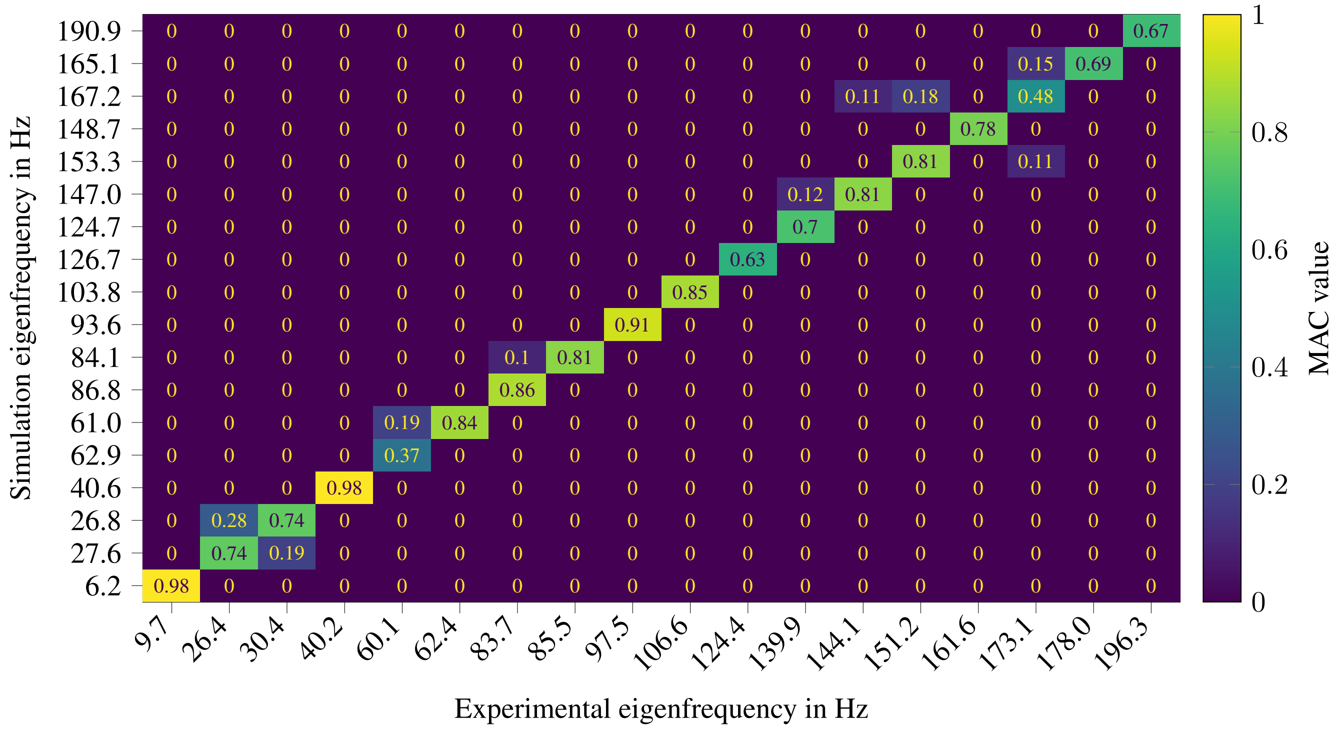

Figure 8. MAC matrix.

propriately be synthesized with the given FE model. Only two eigenfrequencies of modal analysis of the FE model at 58.8 Hz and 114.4 Hz could not be paired with data from experiments. It is possible that these two modes are not observable in experiments due to high damping or small vibration amplitudes.

The second term of C_{12} which ranks the accuracy of eigenfrequencies is validated in Table 4. Here, the paired eigenfrequencies f_{sim} of the simulation and f_{exp} of the experiment are compared and their absolute difference is calculated. Within the 18 eigenfrequencies only three show a frequency error above 10 Hz. All other could be matched quite accurately.

Criterion C_3 considers the difference of the FE lining mass and the real mass. The selected design with its densities and layer thicknesses leads to a total mass of 2.84 kg. The difference to the real mass is therefore 0.14 kg or 5.2 %, which is an acceptable value.

Table 4. Accuracy (Δ) of eigenfrequencies; all values in Hz.

f_{sim}	f_{exp}	Δ	f_{sim}	f_{exp}	Δ
6.2	9.7	−3.5	103.8	106.6	−2.8
27.6	26.4	1.2	126.7	124.4	2.3
26.8	30.4	−3.6	124.7	139.9	−15.2
40.6	40.2	0.4	147.0	144.1	2.9
62.9	60.1	2.8	153.3	151.2	2.1
61.0	62.4	−1.4	148.7	161.6	−12.9
86.8	83.7	3.1	167.2	173.1	−5.9
84.1	85.5	−1.4	165.1	178.0	−12.9
93.6	97.5	−3.9	190.9	196.3	−5.4

5. CONCLUSION & OUTLOOK

In this article the entire process of a FE model update based on experimental modal analysis data is described in detail. The item under test is a commercial aircraft lining panel. An experimental modal analysis revealed 18 modes below 200 Hz with low damping ratios. Since no geometric model was available an optical scan and several post-processing steps generated a surface model of the lining panel. Partial sanding of the surface and measurements with a micrometer gauge led to initial data for layups and thicknesses. Initial material data are drawn from previous projects where honeycomb panels were experimentally tested. As preparation for the model update process three criteria and 16 parameters for optimization were identified. Within the first part of the process, the sensitivity analysis, important parameters could be separated from unimportant ones. The remaining ten parameters are optimized in the second part of the model update with respect to the three given criteria. The result of the optimization is the final parameter set which is afterwards validated against the three criteria. In all criteria the parameters lead to results with acceptable accuracy. The multi-objective optimization created a stable data base for the FE model of the lining panel.

In future work, the parameter set will be employed to conduct acoustic simulations of the lining panel. Removed parts like the shades will be reinstalled gradually to incorporate them into the FE model. Furthermore, it is intended to investigate the impact of geometric details on the mode shapes, exploring how variations in surface topology, curvature, or other design elements affect the vibration behavior. It is the aim to gain a deeper understanding of the relationships between the system's parameters and its acoustic properties, leading to more accurate predictions and improved designs.

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