Study of Noise Characteristics of Vibro Acoustic Materials

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Abstract- *The present thesis deals with the fully anaerobic chamber architecture. The main ambition of thesis is to architecture aerobic chamber according to acoustics codes and requirements. The fully aerobic chamber will be used for checking sound and vibration capacity. The work continued in binary segments. Early segment deals with the broad aerobic chamber approach and all its related architecture expression. Further segment deals with core; the segment is focused on particular architecture according to fundamentals. The architecture of the chamber was implemented using the advanced architecture segments. The chamber is modelled by Finite Element Method (FEM). And analysed using ACT On the other side, testis performed using the KU Leuven facilities to Validate the FEM model. Numerical vs. experimental comparisons were done and acceptable agreement was obtained. On the other side, it was found that sound inside the box reduces due to the smaller sound radiation generated by the treated panel. The absorption coefficients calculated from the small chamber measurements were compared to those taken in a full size reverberation room and an impedance tube. This segment was shown to give absorption coefficients close to those obtained from either the reverberation room testing or impedance tube tests over the frequency range of 200-4000Hz.*

Keywords- Sound, vibration, vibro-acoustic, aerobic chamber, architecture, ACT analysis.

I. INTRODUCTION

The fully anechoic chamber is discussed and the objective of this master thesis is presented. The requirements for measuring sound and vibration are nowadays higher than ever. Customers demand more silent devices, whether it is a car, computer, vacuum cleaner, washing machine or refrigerator. The customer's needs are followed by the ISO standards, which provides requirements, specifications, guidelines or characteristics that can be used consistently to ensure that products are fit for their purpose. Many acoustic ISO standards require the special room, fully anechoic or semi anechoic chamber, where the acoustical measurement has to be performed. To design and implement such a chamber, lot of funds and time Have to be spent. Not all companies can afford it, Even though they need it. In the present paper a noise barrier has been constructed by coupling together several WPC boards. The sound insulation provided by this panel has been tested in a reverberation room coupled with a semi-anechoic chamber. This method allows a much more accurate measurement with respect to the impedance tube method, as it considers a diffuse incident sound field, and more realistic boundary conditions. The experimental transmission loss is used to validate the numerical model based on the TMM framework. Finally, thanks to the possibility of easily varying the profile cross-section by changing the extrusion die, an optimized shape of the cross section has also been simulated by numerical computation using the methods described In response to these needs, The knowledge of the dynamic aspects of excitation and the behaviour of structures, components and/or acoustic enclosures are crucial to have controlled and performing space systems. Virtual prototyping supported by vibro-acoustic solutions is a significant tool to be used on space system developments, due to the proven cost benefits reflected into projects. Low-frequency deterministic methods such as Finite Element Methods design a fully anechoic chamber, which fulfil strict background sound pressure level requirements. The background sound pressure level is closely associated with the transmission loss of the chamber's walls, absorber design inside the chamber and vibration insulation of the whole chamber.

II. ANAEROBIC CHAMBER ARCHITECTURE (DESIGN)

Design Principles In order to achieve the required acoustical properties within the time and budgetary constraints, the following design requirements were set:

- Since most noise rating schemes use data from 250Hz up, the small chamber should be capable of reading from this frequency limit.
- Dispersion of the measured value should be equivalent to that of a reverberation room.
- The sound pressure level should be diffuse, with an absence of standing waves.

- The chamber should be compact and less than 2m wide x 2m deep x 2m high.
- The chamber should be cheap in construction materials and component parts, and easily constructed.
- The chamber should be able to be disassembled for transport, and easily accessible for setting up and experimentation.

III. MATERIAL AND PROPERTIES

Aluminium 5754 alloyis an alloy in the wrought aluminium -magnesium family (5000 or 5xxx series). It is closely related to the alloys 5154 and 5454 (Aluminium designations that only differ in the second digit are variations on the same alloy). Of the three 5x54 alloys, 5754 is the least alloyed (highest composition % of aluminium), but only by a small amount. It is used in similar applications. As a wrought alloy, it can be formed by rolling, extrusion, and forging, but not casting. It can be cold worked to produce tempers with a higher strength but a lower ductility.[14]

Typical material properties for 5754 aluminium alloy include

- Density : 2.67 g/cm^3, or 167 lb/ft^3.
- Young's modulus : 69 GPa, or 10 Msi.
- Electrical conductivity: 33% IACS.
- Ultimate tensile strength: 220 to 330 MPa, or 32 to 48 ksi.
- Thermal Conductivity : 130 W/m-K.
- Thermal Expansion: 23.7 µm/m-K

IV. CHAMBER DESIGN

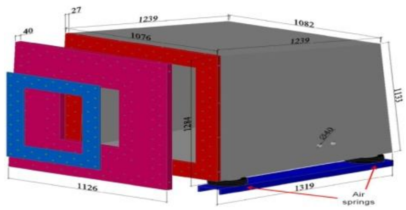


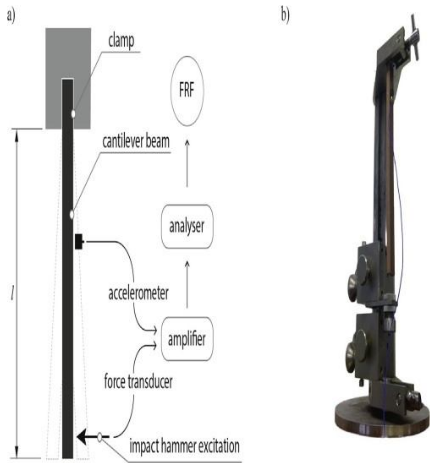
Fig. A sketch of the experimental set-up geometry (dimensions in mm)

Table 1

Point	Coord x	Coord y	Coord z
1	0	0	0
2	0	0.815	0
3	1.15	0.815	0
4	1.15	0	0
5	0	0.001	0.982
6	0	0.778	0.981
7	1.082	0.783	0.848
8	1.082	0	0.849

Table 2Instrument suspending design

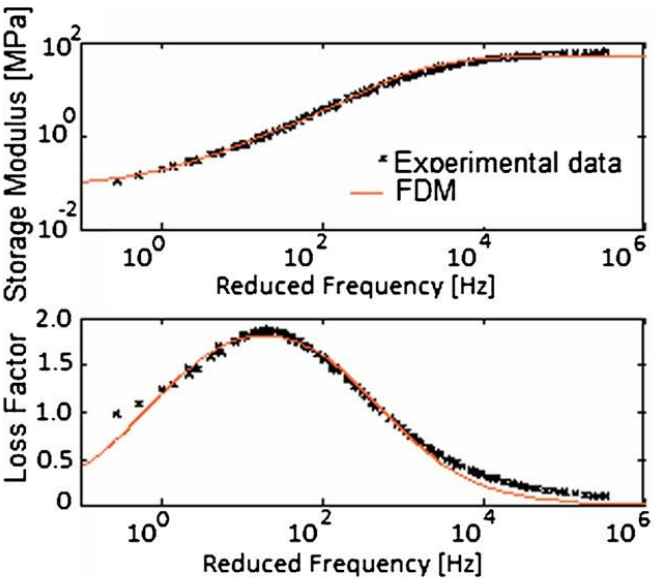
Aluminium characteristics	
Density	2700 Kg/m^3
Young's Modulus	70 e^{-9} N/M^2
Poisson's ratio	0.33



Suspending instrument

V. MODELLING THE VIBRO-ACOUSTIC SYSTEMS (PANELS+SOUNDBOX)

Two coupled models are built:



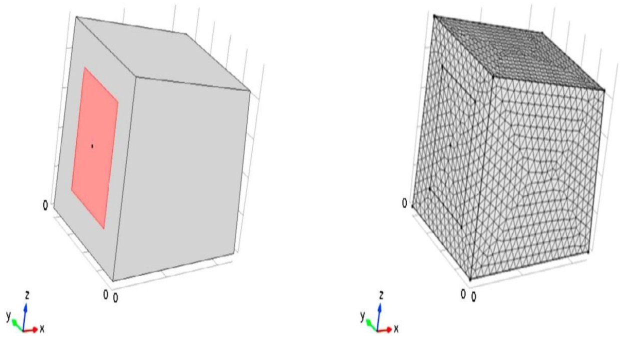
(i)bare panel sound box cavity (a cabin-like testing environment built to exploit both for noise transmission and noise insulation characterization of light weight materials) and (ii)VEM coated panel sound box cavity. University excitation was applied at the centre of the panel and transfer functions (acceleration/force and sound pressure level/force)are calculated upto 1,000Hz. Shows the dimensions of the sound box as well as the set-up of the structural assemblage on the box.

Bare panel + sound box

A vibro-acoustic system consisting of aluminium homogeneous plate dimensions 0.433*0.617*0.003m and sound box acoustic cavity is modelled in COMSOL metaphysics version, using FEM the geometrical co-ordinations of the box is described in the table 1.more detailed description about the KU Leuven-PMA sound box can be seen in.table2 describes the main aluminium characteristics, assigned as structural material.

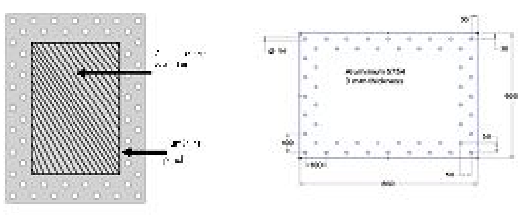
While for the acoustic part one assigned the air at 20 ◦C as material.Describes the main aluminium characteristics, assigned as structural materials. While for the at least 10 elements by wavelength were considered to build the Fem vibro-acoustic meshing. The finer mesh option was chosen, which yielding 53,759 free tetrahedral elements (min &max elements sizes of 0.0046 & 0.0633m, respectively, min and max analysis frequencies of 5000Hz and 12000Hz, approximately)

VEM COATED PANEL



(a)vibro-acoustic system (panel +cavity)
(b)Coupled vibro-acoustic mesh

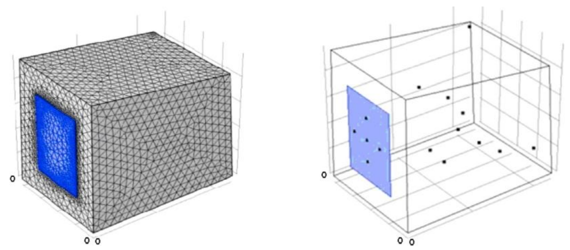
Geometry of the sandwichpanel



Modelling of the sandwich plate

VI. EXPERIMENTAL VALIDATION

In order to validate vibro-acoustic FEM models, an experimental campaign was conducted, where two set-up were built to simulate the numerical models in section 3.1 and **3**.2.The bare and sandwich panels were attached to the front wall of the KU Leuven-PMA Sound Box. Afterwards, the attachment was done by positioning the structural set (front wall panel frame) close to the open part of the Sound Box. Notice that the panel geometry in contact with the inner cavity is nominally identical to a width of 0.420 m and a length of 0.594 m, for an A2 footprint area. Accelerometers and microphone were installed at the same positions as those observations points generated in the numerical models. The system excitation was provided by an impact hammer. This hammer is equipped with the force transducer, which measured the transient input signal. Fig (a) shows the base plate and coated A2 footprint area, while fig (b) shows the referred sandwich panel attached to the KUL euven-PMA Sound Box.



(a)coupled vibroacoustic meshing (b)Acoustic structure

VII. ANALYSES AND RESULTS

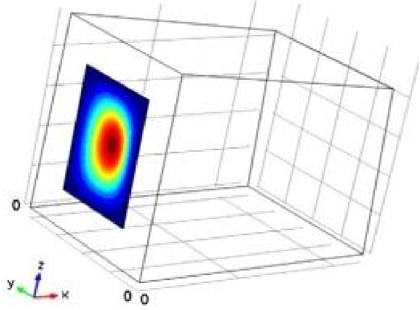
In the section below the structural and acoustic responses for numerical and experimental models are discussed. Transfer function (g/n) for structural part and(dB/N)for acoustic part) are presented and the damping insertion effect is assessed by comparing the structural and acoustics responses of the bare and coated panel configurations. Model validation are also done by numerical vs. experimental comparisons

Bare panel and sound box

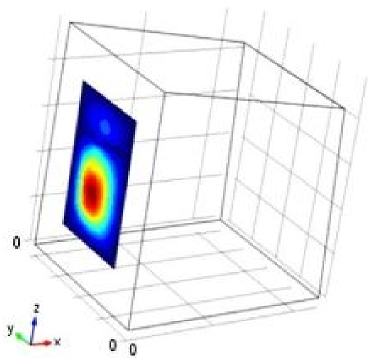
Uncoupled and coupled numerical modal analyses were done to investigate the vibro-acoustic behaviour of the set bare panel sound box. The coupled modal analysis shows that additional modes corresponding to the coupling of the elastic part (panel) are present in this model. The simple frequency comparisons with pure structural and acoustic modal analyses show the influence of the coupling in the panel as well as the cavity responses. The eight first frequencies calculated for purely structural purely acoustic and coupled modal analyses. It is worthy to highlight that differences in the structural natural frequencies of the coupled vibro-acoustic model are noticed ,in comparison with a simple structural model, due to the coupling effect. As an example, see that the first structural bending (a)coupled vibroacoustic meshing (b)Acoustic structure

Table of Analysis result for MODEL, STRUCTURAL & ACT analysis

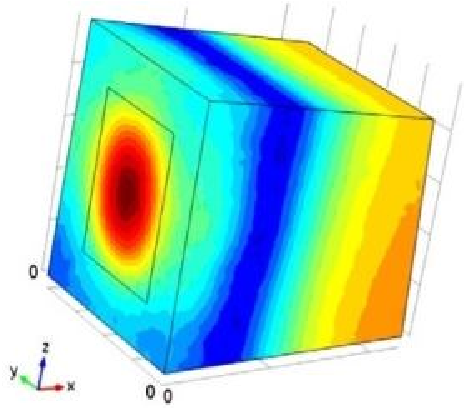
Analyzed Frequency (Hz)			
SL No.	Structural	modal	ACT
1	106.03	1.32e–5	2.15e–5
2	172.43	152.29	103.55
3	253.36	186.95	152.85
4	282.70	213.74	167.97
5	314.70	242.22	185.72
6	418.61	262.41	212.27
7	433.77	284.08	243.15
8	476.60	307.42	245.84



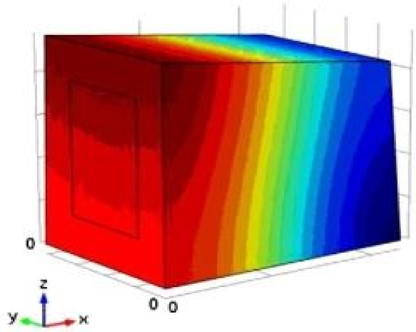
(a)first bending mode at103.5 Hz



(b) structural mode at 152.8Hz

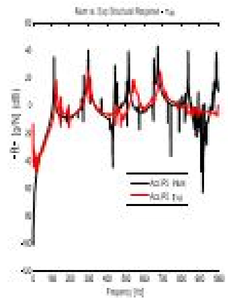


(c)acoustic mode at 103.4Hz)

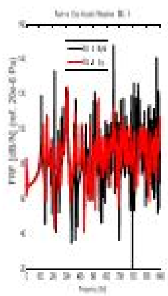


(d) acoustic mode at 152.8 Hz

Structural and acoustic modes

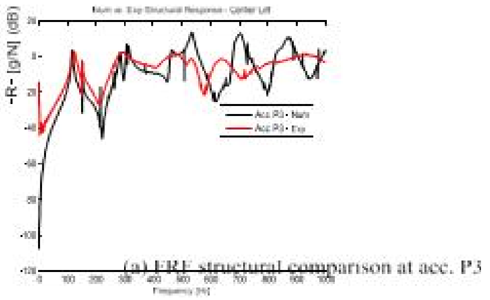


(a)FRF structural comparison

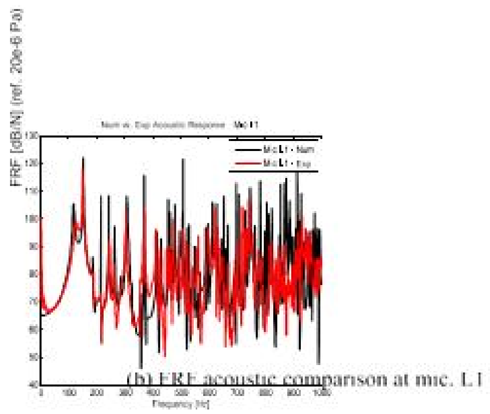


(b) FRF Acoustic comparison

Concerning the acoustic responses, the comparisons are pretty acceptable upto 500Hz, with high modal density above this frequency range, as seen in fig.b. Notice in this figure that the acoustic modes, at frequencies 152.8Hz, 186.95Hz and 307.42Hz, present good agreement. On the other side, at 213.74Hz, one notices a big difference in the pressure magnitude. A better approximation of these numerical vs .experimental comparisons could be obtained by updating the vibro- acoustic model, rather than a purely structural model updating ,as described in section 3.2.



(a)FRF Structural comparision



(b) FRF Acoustic comparision

VIII. CONCLUSIONS

Theoretical vs. Experimental comparisons were done and acceptable agreement was observed for the acoustic part. On the other side, the structural part presented differences, showing that the FD Munder -predicts the added damping, mainly in the higher order modes. Excellent vibration control of the constrained VEM panel was noticed ,in higher frequencies (above300Hz), consequently generating lower sound radiation reduction. For the acoustic part, one observes that the cavity responses were attenuated, mainly due to the portion reduction concerning the structural vibration.

Coupled vibro-acoustic numerical models have been built to assess the effect of structural damping insertion with VEM on acoustic responses. The VEM was firstly characterized and the FDM was adopted to model the shear behaviour of the constrained VEM applied on an aluminium homogeneous plate. The sandwich panel was attached to the acoustic model of the sound box. Structural and acoustic responses were calculated upto1,000Hz. In a second step, vibro-acoustic tests were done, with two experimental setups ,in which the light weight panels (bare and sandwich) were connected to the KUL euven–PMA Sound Box facility.These experimental set-ups allowed us to couple the vibro-acoustics systems and structural displacements and acoustic cavity responses up to 1,000 Hz were measured .

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