Design of Multifunctional Body Panels for Conflicting Structural and Acoustic Requirements in Automotive Applications

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Abstract

Over the past century, the automobile has become an integral part of society, with vast increases in safety, refinement, and complexity, but most unfortunately in mass. The trend of increasing mass cannot be maintained in the face of increasingly stringent regulations on fuel consumption and emissions.

The body of work within this thesis exists to help the vehicle industry to take a step forward in producing vehicles for the future in a sustainable manner in terms of both economic and ecological costs. In particular, the fundamentally conflicting requirements of low weight and high stiffness in a structure which should have good acoustic performance is addressed.

An iterative five step design method based on the concepts of multifunctionality and multidisciplinary engineering is proposed to address the problem, and explained with a case study.

In the first step of the process, the necessary functional requirements of the system are evaluated. Focus is placed on the overall system behavior and diverted from sub-problems. For the case study presented, the functional requirements included: structural stiffness for various loading scenarios, mass efficiency, acoustic absorption, vibrational damping, protecting from the elements, durability of the external surfaces, and elements of styling.

In the second step of the process, the performance requirements of the system were established. This involved a thorough literature survey to establish the state of the art, a rigorous testing program, and an assessment of numerical models and tools to evaluate the performance metrics.

In the third step of the process, a concept to fulfil requirements is proposed. Here, a multi-layered, multi-functional panel using composite materials, and polymer foams with varying structural and acoustic properties was proposed.

In the fourth step of the process, a method of refinement of the concept is proposed. Numerical tools and parameterized models were used to optimize the three dimensional topology of the panel, material properties, and dimensions of the layers in a stepwise manner to simultaneously address the structural and acoustic performance.

In the fifth and final step of the process, the final result and effectiveness of the method used to achieve it is examined. Both the tools used and the final result in itself should be examined. In the case study the process is repeated several times with increasing degrees of complexity and success in achieving the overall design objectives.

In addition to the design method, the concept of a multifunctional body panel is defined and developed and a considerable body of knowledge and understanding is presented. Variations in core topology, materials used, stacking sequence of layers, effects of perforations, and air gaps within the structure are examined and their effects on performance are explored and discussed. The concept shows promise in reducing vehicle weight while maintaining the structural and acoustic performance necessary in the context of sustainable vehicle development.
Acknowledgements

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To my family who has always encouraged and supported me, and constantly wondered when I would be finished with "school", I want to thank you all immensely. You’ve all been a source of inspiration and strength at some point.

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Stockholm, February 2011
Dissertation

This doctoral thesis is based on an introduction to the area of research and the following appended papers:

Paper I


Paper II


Paper III


Paper IV

Paper V


Portions of this thesis have also been presented as follows:


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Part I

Introduction
1 Background and Context

1.1 Historical Vehicle Development Trends

The automobile has come a long way since 1886 when Karl Benz filed a patent for a vehicle with gas powered drive. Henry Ford brought the car to the general public with the model T and the concept of mass production. While different manufacturers would likely disagree as to what is the single most important quality in producing a vehicle, it is likely they would all agree that a policy of continuous development of ones products is an absolute necessity. Continuous improvement of areas such as safety, comfort, handling, performance, and even convenience, have not been achieved without a certain cost. Perhaps the most tangible cost is the growing mass of the vehicles produced. Table 1.1 shows some historical data on vehicle curb weights from a single manufacturer, Saab Automobile, obtained from various unofficial sources, [1, 2, 3]. This trend of constantly increasing vehicle mass and size is by no means limited to a single manufacturer, but is in fact a global trend. While the oil crisis of the 1970’s had a short lived impact on vehicle weight, it’s increase has continued unabated since the 1980’s, at approximately 1.2 % per annum since 1985, [4, 5].

Today’s automobile manufacturers are keenly aware of the challenges facing the automotive industry in the near future. In order to try and achieve some of the mass reductions that will be necessary, manufacturers have already been trying to find areas of obvious mass inefficiency.

Standard grades of low-carbon steel were for many years the norm in vehicle production because they were relatively cheap, formable, weldable, and show a reasonably predictable deformation behavior in a crash scenario. Replacing low-carbon steel with high strength steel, which has a much higher strain to failure and thus can be made thinner, is one method of saving weight which has been well documented within the industry, and has the weight of the steel producing industry behind it, [6, 7, 8].
Table 1: Unofficial Vehicle Curb Weights of Saab cars 1950-2008

<table>
<thead>
<tr>
<th>Model Year</th>
<th>Model Name</th>
<th>Curb Weight</th>
</tr>
</thead>
<tbody>
<tr>
<td>1950-52</td>
<td>Saab 93</td>
<td>805 kg</td>
</tr>
<tr>
<td>1956-58</td>
<td>Saab 93</td>
<td>810 kg</td>
</tr>
<tr>
<td>1959-66</td>
<td>Saab 95</td>
<td>905 kg</td>
</tr>
<tr>
<td>1966</td>
<td>Saab 96v4</td>
<td>946 kg</td>
</tr>
<tr>
<td>1969-84</td>
<td>Saab 99</td>
<td>955 kg</td>
</tr>
<tr>
<td>1976</td>
<td>Saab 99EMS</td>
<td>1161 kg</td>
</tr>
<tr>
<td>1979-93</td>
<td>Saab 900</td>
<td>1174 kg</td>
</tr>
<tr>
<td>1984</td>
<td>Saab 900Turbo</td>
<td>1340 kg</td>
</tr>
<tr>
<td>1986</td>
<td>Saab 9000i</td>
<td>1280 kg</td>
</tr>
<tr>
<td>1985-98</td>
<td>Saab 9000</td>
<td>1302 kg</td>
</tr>
<tr>
<td>1988</td>
<td>Saab 900i</td>
<td>1285 kg</td>
</tr>
<tr>
<td>1999</td>
<td>Saab 9-3</td>
<td>1305 kg</td>
</tr>
<tr>
<td>2001</td>
<td>Saab 9-3Viggen</td>
<td>1438 kg</td>
</tr>
<tr>
<td>2004</td>
<td>Saab 9-3 1.8i</td>
<td>1440 kg</td>
</tr>
<tr>
<td>2008</td>
<td>Saab 9-3</td>
<td>1400-1600 kg</td>
</tr>
<tr>
<td>2010</td>
<td>Saab 9-3</td>
<td>1530-1680 kg</td>
</tr>
</tbody>
</table>

A Larger Step to Conceptual Change

That reduced emissions and fuel consumption are directly coupled to the mass of the vehicle is a well understood fact, however the exact magnitudes of such reductions are only more recently being understood, [9]. There exist significant opportunities to reduce unnecessary mass in existing vehicles and those of the near future, [10], and thus reduce their burden in terms of emissions and fiscal resources. The future of the automotive industry, after all the obvious changes are made, is however much more uncertain. Assuming that regulations regarding emissions, fuel consumption, safety, etc will continue to increase in stringency as they have without exception historically (if in some cases rather slowly such as in fuel economy in the U.S., [4, 5]) it is rather clear that the automotive industry will have to do something more than swap out mild steel for high strength steel and aluminium, should it survive and turn a profit.

The body of work within this thesis has one primary objective: to help the vehicle industry to take a larger step forward in creating the vehicles we will need within the not so distant future. That the passenger car is an integrated part of our lives and of growing importance to developing economies, [11], is understandable as is the simple fact that it is here to stay. Knowing that, we need to be able to produce vehicles in a sustainable manner, in terms of both economic and ecological costs. This concept of achieving sustainable development is the driving philosophical argument for performing the research presented in this thesis, and the foundation of the Centre for ECO² Vehicle Design. By combining
the collective knowledge and experience of vehicle manufacturers in different segments of the transportation industry with the unconstrained ability of academia to explore unproven methods and non-conventional solutions, new synergies and solution strategies combining areas of conventional wisdom traditionally seen as separate entities can be obtained. It is in this spirit that the work presented in this thesis has been performed, and which the author has attempted to convey in the pages which follow.

To understand the concepts and methods proposed herein, it is necessary that the reader understand the two primary areas of vehicle engineering involved; namely vehicle structures and vehicle acoustics. In addition, some familiarity with certain materials and tools which are referred to, i.e. sandwich structures, composite materials, and numerical optimization is also required. A brief introduction to these topics is presented herein. To avoid excessive repetition, the introduction will be kept rather short, and the reader is instead encouraged to examine the accompanying articles or the vast body of external literature on the topics discussed.
2 Vehicle Form and Functionality

2.1 Vehicle Structures

That a modern passenger car looks the way it does is not merely the result of coincidence, or purely an exercise in styling. Certain functionalities are required to achieve an efficient form of transportation, that is among other things; safe, comfortable, reliable, and aesthetically pleasing. The drive train, which propels the vehicle forward, needs to be coupled to the suspension and steering systems, which promote the transfer of tractive forces between the wheels and the road and control its direction. The perceived driving behavior, i.e. turning response, sportiness, etc, is directly effected by how these systems are coupled. Other forces, such as inertial forces encountered during crash, need to be accommodated in a manner that protects the occupants. Seats, doors, and trim details need a support to which they can be attached, and passengers should be protected from the elements. In the context of the present work, these functions are realised through the vehicle’s body structure.

The historical praxis within the automotive industry to accomplish such a structure has been to spot weld together a large number of pressed steel or aluminium components. This assembly, having been coated with a primer prior to being painted, is commonly known as the body in white (BIW). Within the BIW, a significant amount of functional specialization is present. Large sheet metal panels, which in some cases do contribute somewhat to the stiffness of the BIW, are primarily for exterior styling. Underneath these exterior panels are various open and close formed sheet metal profiles to support normal driving loads or to protect the occupants during a crash. Requirements are placed on characteristics such as bending stiffness and torsional stiffness of the BIW to ensure favourable performance and handling. Crash safety, i.e. controlling the modes and degree to which a BIW deforms during various forms of impact, is perhaps the single most important factor involved in the detailed design of the BIW, which often leads to conflicts with other vehicle functions.

While the alarming trends mentioned in the previous chapter may lead one to draw other conclusions, structural mass is in fact a very high priority and vehicle manufacturers spend significant time, effort, and money on reducing it as much as possible. Herein however, lies the fundamentally antagonistic requirements which this work sets out to address; structures which are required to be lightweight and stiff will, by their very nature, vibrate when
subjected to stochastic or frequency dependent forces such as a cobblestone road, or an engine running at varying RPM. This phenomena has in fact led to the development of an entire subcategory of vehicle engineering.

2.2 Noise Vibration and Harshness

Noise, Vibration and Harshness, more commonly known as NVH, is an all encompassing engineering discipline that deals with the objective and subjective structural dynamic and acoustic aspects of automobile design. For reasons of comfort, quality, safety, and reliability among others, NVH engineers are focused on predicting, and controlling or eliminating sound and vibration phenomena within the vehicle.

NVH problems are dynamic by nature, and thus are frequency related. One generally refers to noise as being audible sounds in the frequency range from 30-4000 Hz, and vibration as the mechanical vibration in the frequency range 30-200 Hz, [12]. The three main sources of interior noise are the engine and accessories, tyre/road interaction, and airflow over the external bodywork (wind noise), [13]. Further, one can make the distinction between airborne sound, which is of higher frequencies, and structure borne sound, which is in the lower frequencies. From the perspective of a typical occupant, this distinction might not be so clear, however from the perspective of the NVH engineer, the methods and tools used to deal with them differ significantly.

Airborne Sound

Airborne sound often originates from external sources and propagates into the vehicle interior via holes in the bodywork, door seals, weld seams, etc, [12]. Its contribution to interior sound pressure levels is predominantly in the higher frequency ranges. Controlling airborne sound can be done by eliminating the source of the sound (if possible), or eliminating the transfer paths into the vehicle. Sound pressure waves propagate along the path of least resistance, and commonplace solutions such as acoustic baffles, urethane filler foams, rubber plugs and grommets, and adhesive seam sealants are used to impede an airborne sound propagating into the passenger compartment, [14]. The implementation of these sorts of solutions rely heavily on testing, and experience of the engineers involved. Controlling sound levels inside the compartment is often done with different types of acoustic trim treatments that can offer varying levels of insulation or absorption.

Structure Borne Sound

Structure borne sound is the result of mechanical vibrations propagating through the vehicle structure and eventually causing localised displacements of air. Sound levels due to vibration can be directly related to the volume of air which is displaced. A $1\text{ cm}^3$ volume
displacement, which could be achieved by a 1 m² area of roof vibrating with a displacement amplitude of 1 µm, can cause a sound pressure of 75 dB in a vehicle interior, [15]. Much of the noise in the frequency range up to 500 Hz is caused by cavity resonances which may be excited by vibrations of the BIW caused by suspension or drivetrain inputs. The term booming noise is often used to describe such acoustic phenomena in the frequency range below 250 Hz. Body vibration levels are directly related to the road roughness, vehicle speed, suspension characteristics etc. For a more complete explanation of the phenomena of structure borne noise in general, the interested reader is directed to the literature, [16].

2.3 Alternative Materials

While sheet metal structures offer excellent performance in terms of, among other things, cost, predictable deformation under impact, and recycleability, they do have negative aspects such as a tendency to corrode, and to be a veritable bonanza of vibratory and acoustic phenomena. Sheet metal is however not the only potential candidate for structural design in modern vehicles. Such material concepts as fibre-reinforced plastics and sandwich structures have long been discussed in the periphery of vehicle design. For various reasons, they have not made a significant contribution to the structural composition of mass produced modern vehicles. Nevertheless, as the need for a conceptual change surmounts the challenges of implementing such concepts in production, they are likely to become more common in future generations of vehicles. For the purposes of informing the unfamiliar reader, a brief introduction to these two material system concepts is given.

The Basic Sandwich Structure

Figure 1 shows the most basic form of a structural sandwich. Two thin face sheets of stiff and strong material are attached to a softer and weaker core material to achieve a sum greater than its parts. By separating the two face sheets with a lightweight material, one can significantly increase the bending stiffness, or flexural rigidity, without significantly affecting the weight. This phenomena, commonly referred to as the sandwich effect, is only valid assuming that the face sheets are much stiffer, thinner, and denser than the core material. Mechanically, the face sheet layers take up the applied bending loads and moments as tensile and compressive stresses while the core material carries transverse loading predominantly as shear. For an in depth explanation of sandwich structure theory and technology, the reader is referred to the literature, [17].

Metals or fibre reinforced composites are by far the most common materials used for face sheets. A vast array of materials can be used in the core, but perhaps the most common are expanded polymer foams, honeycombs of aluminium or paper, or balsa wood, all of which can be seen in figure 2. While rather robust, and exceptionally weight effective at carrying loads in bending, sandwich structures are not suited to all types of applications. Concentrated out of plane loading should generally be avoided as it can lead to core or
face material failure and potentially failure of the entire structure. Phenomena such as face sheet buckling, global buckling, core shear failure, face sheet delamination, and adhesive layer failure must be understood to successfully implement a sandwich structure effectively. Special design consideration must be taken when introducing loads into a sandwich structure or when creating a joint between two sandwich panels due to stress concentrations and other effects, [18].

Despite these potential drawbacks, sandwich structures offer a great deal of flexibility in design for minimum weight by merely altering the combination of materials used.
Composite Materials

Composite materials are obtained by mixing two different materials in such a way as to obtain superior properties in the final product than in either of the individual components. Most often in engineering terms, it refers to the combination of some fibrous material, of high stiffness and high strength, with a low stiffness, low strength material which can bind the fibre bundles together and create a structural material. Both components are of significantly lower density than the vast majority of engineering metals.

Depending upon the application, numerous different kinds of fibres exist and are employed offering different properties in terms of stiffness, strength, toughness, damping, mass and fire-resistance to name a few. The same holds true regarding the plastic material, otherwise known as the matrix, and its chemical composition. Figure 3 shows a selection of typical fibres, a two part epoxy resin system used for composite manufacture, and an example of a composite component. For the sake of brevity, the interested reader is referred to the literature for a more in depth discussion on the materials, chemistry, and manufacturing processes of polymer composites [19].

Figure 3: Top Left-A selection of fibres for reinforcement. From top down Hemp, Kevlar/Carbon Fibre (Tiger-weave), Glass fibre, Carbon Fibre. Top Right- A two component Epoxy matrix system. Bottom-A section of a Carbon Fibre composite wheel profile
One of the primary advantages of composite materials from an engineering perspective is their ability to be tuned to a specific application. For an isotropic material such as steel, only the geometry of the material (i.e. profile shape or thickness) can be altered to change its stiffness, strength, or damping. For a fibre reinforced composite, controlling the amount of fibre versus matrix material, fibre direction, mixture of fibres, chemicals used in the matrix, etc, can have a great influence on the materials final properties. In practice, composite materials are often created by stacking layers of uni-directional fibres on top of each other paying specific attention to the direction of individual fibre layers. By adjusting the thickness and fibre direction of each layer, known as a lamina, within the final complete stack of layers, called a laminate, stiffness, strength, etc, can be controlled in multiple directions within the component. Again, for the sake of brevity, the interested reader is directed to the literature for a more in depth discussion of the mechanics of composite materials, [19, 20, 21]. In addition, while the manufacturing processes for most composites are slower than those for pressing sheet metal parts, they offer a high degree of formability using tooling which costs a fraction of that of sheet metal stamps and presses.

**Barriers to the Automotive Industry**

As mentioned previously composite materials and sandwich structures have seen limited use in the automotive industry. This is due to a number of factors. Cost for such materials and the production techniques to create them have historically been rather high. Recent research does however suggest that the time is approaching where such materials may indeed be cost competitive [22]. The concept of multi-material design which can include such materials is considered by some as the future of the automotive industry [6, 10, 23, 24]. This will not, and cannot however, happen overnight. Experimental methods are still necessary to asses the crash worthiness of composite materials and sandwich structures and are still rather simple, for example see [25]. The field of crash analysis and dynamic failure prediction in such materials is only in its infancy compared to the same body of knowledge which exists for metallic material models. This is understandably unacceptable in terms of time and cost within the automotive industry which has become, with the exception of final validation, almost completely computer based in its design and development work [26, 27, 28]. As the cost of such materials continues to decrease, and the modelling techniques continue to improve, the amount of these materials used in load bearing structural components is likely to increase drastically. The first intentions of such mass-production implementation have already been made explicitly clear by BMW [4]. Whether this is the snowball that starts the avalanche, or merely a clever marketing tactic by an automotive producer associated with high-performance, luxury vehicles, remains to be seen.
3 Tools for Vehicle Design

3.1 Structural Design

Nearly all structural design of modern vehicles is done using computer aided engineering tools. Finite element analysis (FEA), in various different formulations, constitutes a significant portion of the numerical analysis within the automotive industry. It also forms the basic foundation of the design methods discussed within this thesis. It is assumed that the reader has a working knowledge of such methods, which have been elegantly and thoroughly derived and explained in numerous textbooks on the subject, for example [29, 30, 31, 32] and will therefore be excluded from the current text.

Such numerical tools are advantageous to the structural design process because they are accurate, consistent, repeatable, and save time and money by eliminating some of the need for testing and prototypes. Perhaps most importantly for the context of this thesis, these methods lend themselves well to the concept of optimization. The ability to quickly and accurately gain an understanding of how a small change to a structural component can change its behavior is critical to achieving a successful design with minimal time and resources.

3.2 NVH tools and methods

Historically, the prediction and control of sound and vibration has been a manpower intensive process of "test-analyse-fix" [33] as vehicle refinement often takes place in the final stages of production. This policy precludes the possibility for NVH engineers to actively avoid designs which are prone to problems, and rather forces them to take on the mantle of the crisis management team.

A significant amount of NVH testing goes towards predicting which portions of the BIW are going to vibrate the most, and thus potentially be the largest source of unwanted sound. Proposals for improvements of such procedures and test methods are numerous in the literature, [34, 35, 36, 37]. By far, the most common method of addressing such vibratory problems, is with the use of viscoelastic damping layers, [38], also known as deadeners,
which are effective, but rather weight inefficient, and sometimes costly. Developing a predictive tool to eliminate the need for such a high level of testing has been a goal on the horizon for many years and has led to many proposed solutions, [39, 40, 41, 42, 43, 44]. For a more detailed survey of the methods explored, the reader is directed to the literature, [13].

In addition to panel damping treatments, a great deal of work goes into selecting interior trim to provide acoustic functionality. While this has also historically been a test and experience driven exercise, in recent years the numerical tools for studying trimmed body behavior have grown tremendously. When the work in this thesis began, very little was available in the published literature, however now the list abounds with researchers exploring methods like finite element analysis of porous materials and statistical energy analysis (SEA) or hybrid methods of the two, [45, 46, 47, 48, 49, 50, 51, 52]. These methods are effective, however they often require large amounts of computational time. This is partially due to the necessary use of a fluid cavity in addition to the actual models of the trim components within the finite element model, which must be coupled to the structure and to each other and greatly increases the number of degrees of freedom to be solved for, and the complexity of the solution type. For a more detailed discussion of such topics, the reader is again directed to the literature, [15, 53].

Perhaps the largest problem in developing effective and consistent NVH solutions, is the variability within the problem itself. Due to the huge number of processes involved in assembly, and the large number of options available within a manufacturer's model family, no two individual vehicles will have the exact same NVH behavior in the frequency range of interest, and thus any "ideal" solution achieved will in fact only be "ideal" for the given vehicle in question [54, 55, 56]. This does not however make the process in identifying and quantifying the dominating mechanisms for response an exercise in futility. On the contrary, appropriate estimates of behavior are highly valuable, whereas extremely precise results obtained at great expense of time and money may be superfluous or even un-useable. This is obviously a question of balance which is still in its infancy. The key lies in incorporating the "necessary" physical phenomena, rendering further analysis and subsequent conclusions viable.

3.3 Optimization

Optimization, in an engineering design context, refers to an iterative process in which an equation or set of equations is solved to minimise(or maximise) a certain quantity which is dependant upon one or more variables chosen by the user. The basis for successfully solving an optimization problem is some form of mathematical expression of the quantity to be optimized, commonly known as the objective function, which may or may not be known explicitly from the outset. The unknowns, or design variables, are changed by the optimization algorithm in order to achieve a best possible value of the objective function. A certain degree of control over the problem can be obtained by placing constraints on
certain parameters, such as the region of validity of the design variables, or the numerical value of output from a portion of the system of equations. Equation 3.3 shows the most basic form of optimization problem where $f(x)$ is the objective function, $x_1$ and $x_2$ are the design variables, and the inequalities for $G(x)$ represent the constraints.

$$
\text{minimize } f(x) = x_1 + x_2 \\
\text{subject to:}
\begin{align*}
G(x_1) & \geq a \\
G(x_2) & \geq b
\end{align*}
$$

(1)

Methods and algorithms for solving such problems are numerous, and the body of theory and knowledge in this subject is immense. Depending on the size and complexity of the problem, it might be possible to solve with pen and paper, or require extended computational time on high performance computer clusters. Furthermore, for any given optimal solution, the question of robustness, i.e. certainty in that the solution is in fact a global optima and not merely a local optima, must be properly addressed. Within this work, two distinctly different methods of optimization have been employed, namely a gradient based optimization method known as the method of moving asymptotes (MMA), [57, 58], and the topology optimization algorithm known as bi-directional evolutionary structural optimization (BESO), [59, 60]. For a more detailed discussion of general optimization theory the reader is again directed to the literature, [61].

**Gradient Based Optimization**

Gradient based optimization tools, of which MMA is only one of many, are based on the concept that while the explicit definition of the objective function at hand might be unknown, i.e. implicit, by evaluating the problem multiple times using small perturbations of the design variables an approximation of the function can be obtained. This approximation can be used then to choose the direction of change of the design variables to achieve an improvement in the objective function.

Such an algorithm is highly effective in certain cases, like for example mass optimization of a truss structure using thickness of its members as design variables. In such a case, constraints might be placed upon the minimum thickness of certain members, or the maximum displacement of the truss for a given loading condition. These algorithms are almost always implemented together with some form of numerical tool to solve the objective function rapidly. In the context of this thesis, FEA is used exclusively.

Gradient based methods are highly effective, however they have limitations in terms of the number of permissible design variables and size of the model to be solved. As each variable will require at least one perturbation analysis per iteration, excessive numbers of variables or excessively long solve times for the model can effectively prohibit the use of such al-
algorithms. This is a fact known within the automotive industry in terms of optimization for crash computations. Here, statistical based methods may be more appropriate, [62]. In addition, for the present work which focusses in part on the dynamics of the systems behavior, the influence of resonances in various sub-systems adds further complication.

As previously mentioned, MMA, [57], including recent improvements in the algorithm regarding global convergence, [58], have been the only gradient based algorithm used in the optimization performed in this work. This algorithm was chosen as it is well known and regarded within the literature, efficient, effective, and implemented in available computer code in several different forms.

**Topology Optimization**

Topology optimization is the process of altering the placement of material within a structure to achieve the most mass effective structure possible. The numerical origins of the concept can be attributed to Bendsøe and Kikuchi, [63]. Topology optimization is perhaps most easily understood by, and most often explained with, the example of a 2D cantilever beam, like the visualisation shown in Figure 4. For a load applied at the end of the beam, and some form of constraint placed on the displacement, the algorithm will create an elegant truss like structure where unnecessary material is removed. A thorough review of the vast body of knowledge on this topic is again, outside the scope of this thesis, and the interested reader is directed to the literature, [64].

The methods described by Bendsøe and Sigmund,[64], use scaling factors for the material properties, usually the density, of the material involved. In practice, this requires the final result of the optimization to be interpreted and re-modelled in some form. As mentioned previously, the BESO algorithm was chosen for use within this work. This algorithm was seen as simpler to implement in the finite element analysis framework used, and perhaps most importantly, the final geometry of the optimization did not require further alterations to be used as input for later work. This by no means excludes validity of other such topology algorithms to address the problem discussed herein, the choice was merely seen as the best possible use of available resources at the time.

![Figure 4: A typical topology optimization problem](image-url)
4 The Multifunctional and Multidisciplinary Design Paradigm

The previous chapters within this thesis have given the reader a background of information necessary to understand the central topic of the work, i.e. multidisciplinary and multifunctional design.

The research work leading to this thesis forms a synthesis between design of a structurally viable component and its acoustic performance in an automotive setting. The paradigm used is multifunctionality, i.e. designing an integrated component with desired acoustic and structural performance at a low weight. The central focus of the work has been finding methods to balance the strongly conflicting requirements of a high stiffness, low weight structure with a comfortable, low noise acoustic environment within the vehicle. The resulting methodology developed and the tools used will be presented and explained in a manner which gives context, perspective and hopefully a higher level of understanding of the detailed body of results included in the appended papers.

4.1 An Iterative Design Process

Figure 5 gives a graphical representation of the proposed design process to successfully achieve a design which is both multidisciplinary, and multifunctional. From first observation, it is obvious that the process is iterative in nature, and has several levels of feedback from the later stages of design to the earlier stages acting as the check points of the process. While chronologically speaking, this methodology in its current form appears first in paper III, from a retrospective viewpoint the author nevertheless feels it is appropriate, and prudent, to begin the discussion at this point.

The proposed design methodology consists of the following five steps:

1. Define the multifunctionality
2. Establish performance targets
3. Propose concept to fulfil requirements
4. Develop method to refine the concept
5. Evaluate the final results and assess the effectiveness of the method

While in itself, the above list of steps resembles many common engineering practices, two key aspects differ which are of significant importance. The first aspect is that of the focus on system functionality rather than component functionality. For a single engineer, or a small team, to be able to implement this design methodology, the exclusion of detailed component or sub-component studies is necessary.

The second key aspect lies in the final step of the method. Here, not only the final product of the design process is evaluated, but also the process, which led to it. A final result may fulfil the engineering specifications as defined, but fail to fulfil the functional requirements. This may be due to a poor understanding of the system functionality, or an unsuccessful attempt to define the proper methods of evaluation. In order to widen the basis for discussion, each step of the proposed multidisciplinary design methodology is detailed below:

Step 1: Define Multifunctionality
In the first step of the process, the engineer must examine the system as whole and assess which core functionalities are necessary. Effort should be made to avoid focusing on sub-problems or sub-systems and maintain a global perspective. This should include a detailed examination of the existing design specifications, the goal of which is to establish how the existing components contribute to the overall system. On what level does the specification enable a functionality to be achieved on the component or system level?

Step 2: Establish performance targets
As a novel concept shall be proposed, a literature survey of the limits of current technology to achieve the desired functionality should be performed. External sources of knowledge should be assessed in addition to any internally available material. New methods of achieving the desired targets may exist elsewhere which the designer is not aware of. Testing of the existing predecessor system should also be performed. Again, the focus should be on assessing the system behaviour rather than that of individual components.

Step 3: Propose concept to fulfill requirements
Any component suggested should be capable of fulfilling multiple system requirements rather than sub-problem requirements. All potentially positive aspects of the materials used should be taken advantage of and all negative aspects minimised as much as possible. No solution should be proposed which does not have the capacity to fulfil several system functionalities simultaneously or which would require significant problem solving afterwards to perform adequately.

Step 4: Develop method to refine the concept
The methodology described here is an iterative process, thus it lends itself well to the use of numerical tools rather than prototypes. The automotive industry is familiar with the time and cost benefits of switching to a numerical approach, however one additional positive aspect with respect to multifunctional and multidisciplinary design is in the ability to rapidly estimate the system response for a range of cases, in ways which may be considered
Figure 5: A graphical representation of the proposed design methodology
unconventional or perhaps impossible to achieve in a laboratory.

This step in the process emphasises the need for the engineer involved to understand the system as a whole. For the two primary areas of interest, structural and acoustic response, FE analysis is a viable approach which in combination with some sort of optimization scheme enables a multifunctional and multidisciplinary design approach. Implementing the correct sort of analysis to ensure that the system performance requirements are met lies, however, in the hands of the design engineer. This may necessitate some experimentation to establish the correct load cases and boundary conditions for the conceptual model before the optimization scheme is implemented. Once a satisfactory set of load cases has been established together with a relevant objective function for the optimization, the optimization can be run a sufficient number of iterations to achieve a stable result.

Step 5: Evaluate the final results and assess the effectiveness of the method

The results of the optimization should presumably fulfil all design constraints as decided in the beginning of the refinement step. Fulfilling numerical constraints within an optimization framework and fulfilling the overall system requirements are however two separate aspects. In the final step of the design process, not only should the final product of the design be evaluated, but also the method in which it was obtained. This means evaluating the functionality of the concept against the initial requirements as well as evaluating the method of calculating the functionality within the design loop. Questions that may be of interest are: were the load cases used adequate to completely describe the desired functionality or are more/different numerical models necessary? Was the method of modelling the desired functionality sufficient or should new tools be sought? Is it possible to use existing tools in a new way to evaluate the functionality differently or more effectively? In analysing the outcome of the process, the result may be considered sufficient, or it may require further iterations. In the case where further iterations are required, it is necessary to keep a focus on the overall system performance and not become blinded by the details of individual components.

4.2 The Case Study

The concept of a multifunctional body panel was loosely formulated prior to initiation of the work presented here. It was decided that the functions of structure and acoustic components within a passenger car would be considered, but was by no means clear on exactly which portion of the vehicle to address, or to which degree the functionalities should or could be accommodated. The initial step in the work then, was to establish an appropriate vehicle system to study which would be a candidate for the realisation of the idea of a multifunctional panel, i.e. a system which could be simplified via functional integration.

The roof section of a Saab 9-3 Sport Wagon was considered the most suitable choice due to its relative geometric simplicity, high number of constituent components, and strong influence on both structural and acoustic aspects of the final vehicle. Figure 6 shows a
schematic cross section of the roof system studied.

![Diagram of roof structure](image)

**Figure 6: Cross section of the roof structure chosen for study**

### Design Step 1

In accordance with the first step in the design process, a list of functionalities provided by the components in the roof system should be compiled. Using figure 6 as a starting point, this list of perceived functionality might appear as follows:

- **Outer sheet metal**: Protects passengers from the elements, adds torsional stiffness to the BIW, elements of styling also involved.
- **Anti-flutter**: Prevents the outer sheet metal from vibrating against transverse beams.
- **Panel damping treatment**: Reduces outer sheet metal vibrations.
- **Rear transverse beam**: Adds stiffness to BIW, predominately in side-impact scenarios, but also in torsion. Attachment point for gas springs on rear door.
- **Front transverse beam**: Adds further stiffness to the BIW in the same manner as the rear transverse beam.
- **Acoustic absorbent**: Controls airborne sound in the cabin and sound transmission through the roof.
- **Headliner**: Controls sound levels in the cabin, provides attachment for accessories, hides underlying structure from the occupants, elements of styling.

From the above list, several conclusions can be drawn. Functional redundancy is quite obviously present: anti-flutter and damping treatments are both necessary for the sole purpose of controlling vibrations of the outer sheet metal. Two beams are used to provide transverse stiffness in impact and torsional stiffness to the BIW. The headliner and the acoustic absorbent are both required for sound control. Styling aspects are handled strictly by the
headliner, which is necessary to hide the unpleasant appearance of the transverse beams, damping material, etc from the passengers.

This description is by no means all encompassing, however it serves as a good starting point. As the design philosophy is iterative in nature, any missing aspects can be added at a later stage when a better understanding of the problem has been achieved. Based on the component-wise breakdown of the structure, the following list of actual functional requirements might be compiled. This is the list which was used to begin the design process.

- Structural stiffness to the BIW both in torsion and side-impact like scenarios
- Acoustic absorption
- Vibrational damping
- Protection from the elements
- Styling, including places for lighting, etc

Design step 2

Having established the desired functionality, the next step is to evaluate the system and set performance targets for the new solution. To begin with, a thorough survey of both Saab’s own internal documentation, and the open literature was performed to establish the methods and tools considered state of the art at the time. The author has attempted to present this body of knowledge in a concise form in the foregoing chapters, in addition to a more extensive discussion presented in paper III. An initial campaign involving testing and simulation of this state-of-the-art vehicle system was also performed as a supplement to the information in the literature.

The testing and simulation work involved several steps. Initially, full-vehicle acoustic measurement of a production Saab 9-3 Sport Wagon was performed. Following this, the roof structure of the same production vehicle was removed and sent to KTH where laboratory measurement of acoustic and vibro-acoustic properties could be performed. Sound transmission loss (STL) testing was performed for both the full vehicle, and for the component according to international standards.

Comparison between the results of STL testing for full vehicle and component can be seen in figure 7. These two results showed very good agreement, and in addition to giving valuable information on the systems behavior, two additional conclusions of significance could be drawn which add to the body of scientific knowledge in the field, namely:

- With regards to sound transmission loss testing, for the roof panel at least, and most certainly for other body panels, in-situ testing yields sufficiently accurate results as to eliminate the need to disassemble a vehicle to measure individual components.
- In frequencies above approximately 1000 Hz, sound transmission through glass surfaces contributes significantly to the total sound transmission into the passenger compartment.
Vibro acoustic testing on the roof system was also done to measure the system’s frequency response to a structural excitation. An inertia shaker was attached to the drivers side A-pillar and excited using a white noise signal and a laser vibrometer was used to measure the vibrational velocity of the inner headliner and outer roof. Figure 8 shows the shaker setup and measurement points.

![Figure 8: Inertia shaker attachment and measurement grid points](image)

At this point, it was decided to further investigate the headliner’s mechanical properties to fully understand its contribution to the system. The headliner, shown in cross-section in figure 9, was composed of three structural components; two fibre reinforced plastic membranes, and a cellular foam core. Sections of the headliner were removed, the component layers separated and cut into suitably sized samples, and tested in a tensile testing machine to obtain the Young’s modulus for each material. The values of stiffness for the various
Table 2: Mechanical properties of headliner components

<table>
<thead>
<tr>
<th>Component</th>
<th>Testing</th>
<th>Literature</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outer Membrane</td>
<td>9100</td>
<td>4500-7500</td>
</tr>
<tr>
<td>Inner Membrane</td>
<td>4800</td>
<td>4500-7500</td>
</tr>
<tr>
<td>Foam Layer</td>
<td>8.80</td>
<td>3.45</td>
</tr>
</tbody>
</table>

layers were compared with values for similar materials within the literature and found to be reasonable, if somewhat above expectations. Table 2 shows the results of tensile testing compared to values within the literature.

![Figure 9: Closeup of headliner cross-section](image)

To further evaluate the acoustic performance of the headliner, static flow resistivity was also tested using samples from the central area of the headliner. The resulting values ranging between $6.12 \times 10^6$ and $6.24 \times 10^6$ (Pa*s/m²), implied that the headliner was, in principle, impervious to fluid flow.

While these tests gave a considerable understanding of the headliner in itself, a more detailed understanding of its role in the system was necessary. To achieve such an understanding, numerical simulations were performed. Two fundamentally different modelling approaches were used, one focused on realism, i.e. true geometry, and one focused on accuracy and computational effort, i.e. simplified geometry.

To perform such simulations however, it was necessary to include an accurate model of the headliner itself. In its basic construction, a headliner is already a multi-layer component, however as a goal of the work was to determine its performance on a more global structural-acoustic level, a simplified method of modelling the headliners properties was developed.
The approach chosen for the modelling was based on the concept of homogenised equivalent mechanical properties of the actual sandwich structure taking into account the thickness distribution and the mechanical properties in table 2. Due to its layered construction, the Young’s modulus in the sandwich varies through the thickness depending on the different material layers. Figure 10 shows the cross-section for an arbitrary sandwich with dissimilar faces like that of the headliner studied here.

![Figure 10: Arbitrary sandwich cross-section (redrawn from [17])](image-url)

Flexural rigidity (denoted as $D$), which describes a sandwich beams stiffness in bending, can for a unit width sandwich beam of arbitrary cross section (see figure 10) be expressed in the following manner [17]:

$$D = \int E z^2 dz = \frac{E_1 t_1^3}{12} + \frac{E_2 t_2^3}{12} + \frac{E_c t_c^3}{12} + E_1 t_1 (d - e)^2 + E_2 t_2 (e)^2 + E_c t_c \left( \frac{t_c + t_2}{2} - e \right)^2$$  \hspace{1cm} (2)

Where:

$$e = \frac{E_1 t_1 d}{E_1 t_1 + E_2 t_2}$$

$$d - e = \frac{E_2 t_2 d}{E_1 t_1 + E_2 t_2}$$

$$d = \frac{t_1}{2} + t_c + \frac{t_2}{2}$$
Lower case $t$ denotes a thickness, and $E$ is the Young’s modulus. Subscripts 1 and 2 denote the upper and lower face sheets. Subscript $c$ denotes the core material. In this case, the face materials are the two fibre reinforced membranes and the core the structural foam layer. Lower case $z$ denotes the vertical coordinate of the sandwich cross-section where $z = 0$ is the neutral axis of the sandwich structure. By inserting values of $t$ and $E$ into Equation (2), a value can be obtained for the flexural rigidity.

Thickness of the headliner was measured in a number of places, and in principle only the core material varied. Using a large number of measurement points and some estimations of the geometry, the thickness distribution shown in figure 11 was derived. This information, together with data from tensile testing was used to obtain the flexural rigidity of each section according to equation (2). Using a total nominal thickness for the entire headliner of 6.0 mm, an equivalent Young’s modulus for each of the sections was obtained for use in the FE model according to equation (3). The density of the model was based on the measured mass of the actual headliner.

\[
E_{equiv} = \frac{D}{\int z^2 dz} = \frac{D}{\frac{2 \times (3.0)^3}{3}} [MPa] \tag{3}
\]

These steps enabled the entire headliner to be modelled without detailed modelling of the individual layers.

For the simplified geometry modelling, a hierarchical finite element code developed in house was used (see [65, 66]), and sound transmission loss calculations for the roof structure with equivalent headliner properties of the thickest section of headliner were made through the frequency range of 100-500Hz. Results of the calculations and a comparison to the aforementioned testing can be seen in figure 12. The conclusions of this comparison is as follows:
Hierarchical FE modelling using the equivalent solid methodology is accurate in predicting sound transmission loss up to approximately 500 Hz. Above 500 Hz, the simplicity of the model prevents the accurate prediction of sound transmission loss; this is mainly related to the excitation method.

For the accurate geometry modelling, a coupled fluid-structure analysis using NXNastran was performed. A model of the headliner was created using the equivalent mechanical properties discussed above. Damping properties of the headliner were introduced in two ways based on estimated properties of the open porous layer; firstly using an equivalent surface impedance calculated based on theories for acoustical wave propagation within porous media [67], and secondly with advanced poroelastic-acoustic modelling tools coupled to Nastran [68]. A structural model of the component tested in the laboratory was created as was an acoustic cavity model accounting for the barrier created by the headliner. A cross-sectional view of the FE model can be seen in figure 13.

The model was excited in the same manner as in the laboratory test, i.e. a harmonic excitation was applied to the drivers side A-pillar, and vibration velocity levels of the headliner and outer roof were calculated. A basic parameter study was performed using the FE model with surface impedance. Effects of headliner stiffness, boundary conditions, level of damping, and even acoustic cavity properties on the vibration levels of both the headliner and the outer roof were evaluated. These results, together with the advanced poro-elastic...
NXNastran model, and measurements from vibroacoustic testing were compared and good agreement was obtained. Figure 14 shows one such example of comparison between measurements, the simplified, and the advanced FE model.

In comparing the FE results to testing results, the following conclusions were made:
The mechanical coupling between the roof structure and the headliner has the largest impact on the vibration level of the headliner among the parameters investigated.

- Stiffness properties of the headliner have relatively little effect on its vibrational behaviour
- The advanced poro-elastic model and the simplified model with surface impedance both provided accurate results

As the headliner in a vehicle interior is a large surface in direct contact with the air surrounding the vehicle occupants, its vibrational behaviour is significant to the level of sound within the cabin. Should such a design be used, from an acoustic standpoint, a better understanding of the nature of the attachment mechanism would aid in NVH development.

In addition to being quite accurate, in particular concerning the higher end of the frequency range studied, the modelling methodology developed was simplistic enough that implementation for other trim panels of interest should not be exceedingly difficult. While again, outside the main focus of work in this thesis, i.e. development of a multifunctional design methodology, these results contributed to extend the understanding of coupled structure-acoustic analysis of trimmed vehicles using finite elements in the low to mid-frequency range.

Having completed a vigorous campaign of testing and numerical modelling and evaluating the internal design requirements from Saab, and the literature, a list of performance metrics were decided upon. For reasons of confidentiality, and also because they do not add any significant information in the context of multifunctional design, the numerical values for the performance metrics are omitted from the list below.

1. The system shall provide sufficient localised static stiffness such that a given load applied over a given area does not exceed a given displacement.
2. The system shall provide sufficient global stiffness such that the first vibrational mode of the panel exceeds a given minimum frequency.
3. The system shall provide sufficient global stiffness that the global bending, torsion, or dynamic stiffness of the body in white will not be degraded.
4. The system shall provide equivalent resistance to buckling in the lateral loading as that of the existing system.
5. The system shall provide sufficient acoustic performance in terms of both structural damping and acoustic absorbance that the acoustic environment within the vehicle shall not be degraded and should preferably be improved.
6. The system shall support mounting of accessories such as cabling, lighting, etc.

In addition to evaluating the vehicle system to assess which functionality were necessary, this step of the process also involved assessing the maturity of tools which could be used to predict the concepts behavior in regards to the acoustic performance. At this stage, both the commercial tools used, and the more research based tools were both deemed reliable and accurate.
**Design step 3**

Having established a list of functionalities and performance metrics useful in their assessment, the next step in the design process is to propose a concept which fulfills the functional framework. In holding to the concept of multifunctionality, a highly tuneable and adaptable concept which can simultaneously address as many of the requirements as possible, was required. A sandwich construction with multiple layers, multiple materials, and thus highly adaptable functionality was proposed in dialogue with the engineers at Saab. Figure 15 gives a graphical representation of the concept.

![Figure 15: A schematic of traditional and sandwich panel design](image)

Two configurations of the concept were proposed, each consisting of four layers; external face sheets, a structural foam layer, a single layer of lightweight, open–celled viscoelastic foam, and an interior face sheet for structural and aesthetic functionality (see Figure 15). In one panel configuration, the interior face sheet was perforated to allow fluid interaction between the passenger cavity and the acoustic foam. This was done to evaluate the possibility of vibrational damping and sound absorption introduced by the interior layers of the panel.
Using the functional requirements and performance metrics derived in previous steps, an optimization framework was developed using FEA. Load cases and constraints were constructed which were deemed sufficient to simulate the required structural and acoustic functionality demanded in a new design. As this was the first attempt, relatively simple materials were chosen, isotropic outer face sheets, a standard grade of structural foam, and a typical visco-elastic foam used for acoustic treatments.

For the inner face sheet, a fixed hole pattern, as shown in figure 16 was assumed. A relationship to calculate equivalent solid material properties for the perforated was derived from the literature [69, 70, 71], and can be seen in equation (4). While such perforations would have a slight effect on the Poisson's ratio of the sheet, these effects were considered minimal and ignored. It should also be noted that in the present work, a square pattern was use, i.e. $S_x = S_y$ in figure16 as shown in equation (4). Table 3 shows the values of material data used in the optimization.

![Figure 16: Perforation Geometry of Inner Face Sheet](image)

$$E^* = E \left\{ 1 - \frac{\pi}{4} \left( 1 - \frac{S_x - d}{S_x} \right)^2 \right\}^{(2S_x-d)/0.8S_x}$$ (4)
Design step 5

In the final step of the design process, the results of the design should be evaluated as well as the process which was used to obtain them. Results of the mass optimization, including a comparison between the optimized panels’ masses and the original construction’s mass can be seen in table 4. In addition to the load cases within the optimization, an additional structural analysis was performed in the form of non-linear buckling analysis of the panel. This load case was excluded from the optimization due to excessive time for calculations and difficulties with the FEA software. However, both panel configurations performed acceptably according to this test criteria.

<table>
<thead>
<tr>
<th>Table 4: Optimization results for perforated and non-perforated panels</th>
</tr>
</thead>
<tbody>
<tr>
<td>Perforated Panel</td>
</tr>
<tr>
<td>Optimized Mass(% of Conventional)</td>
</tr>
<tr>
<td>$t$ Outer Sheet [mm]</td>
</tr>
<tr>
<td>$t$ Structural Foam [mm]</td>
</tr>
<tr>
<td>$t$ Inner Sheet [mm]</td>
</tr>
<tr>
<td>Active Constraint</td>
</tr>
</tbody>
</table>

After structural optimization, it was decided to evaluate the acoustic performance of the two sandwich configurations, and to compare it with the conventional solution. A direct comparison in this sense proved somewhat difficult using the model verified previously. At this stage, the conceptual model for panel was rather primitive and detailed aspects of design, such as how it should be attached to the existing structure, were unclear. In reality, implementation of such a panel concept would require a certain amount of re-design at the interface between the panel and the rest of the vehicle. Re-designing the rest of the vehicle structure and/or adapting the existing model to interface with the panel concept in a realistic and accurate way was deemed external to the central focus of the thesis. For this reason, it was impossible to structurally excite the sandwich panels in the same manner as was performed in the laboratory and an acoustic excitation was chosen instead. A coupled structure-fluid frequency response analysis was performed using a fluid cavity excited at a single node located at the drivers head and the average sound pressure within the cavity was calculated for the frequency region 100-500 Hz. This calculation was performed using both panel configurations as well as the conventional model. The results of these calculations can be seen in figure 17.

From a mass reduction perspective, the results in table 4 are excellent. From a robustness standpoint, the face sheets are without question far too thin. Herein lies an important point within the design method; one could simply draw the conclusion that the sheets are too thin and increase the thickness to achieve a satisfactory level of robustness and assume that the component was then over-dimensioned. This would be logical, but inherently wrong. The correct conclusion should be that as the method used has delivered an unsatisfactory solution, it is the method itself which is lacking and should be revisited. This
result demonstrates the inherent difficulty in translating between global requirements and component level requirements and emphasises the need for a thorough understanding in this area. There are no shortcuts to achieving a successful design.

From an NVH perspective, the results in Figure 17 showed that the general behaviour of all three configurations was quite similar. This result was significant; despite the fact that the mass of the system has been reduced by approximately 80%, no noticeable degradation of the acoustic absorption of the system can be seen. Figure 18 shows the same results as obtained in Figure 17, however levels have been averaged over 10 Hz intervals. Here, the differences between the conventional and sandwich configurations is much clearer and seemingly unaffected by the significant reduction of mass. It would also appear, that in the frequency range above 300Hz, the sandwich constructions in fact may perform better than the conventional design in terms of acoustic damping. While the absorption of the panels is only a single aspect of a complex NVH problem, it does bode well for the concept. Differentiating between the perforated and non-perforated configuration is somewhat more difficult. At some frequencies the perforated panel appears to perform better, and at others the non-perforated. This observation brings up an interesting point of discussion; while the non-perforated panel represents a minima in acoustic absorption, the perforated configuration, which was rather arbitrarily chosen, offers potential for further improvement.
in absorption characteristics.

Evaluation of the models and tools available for evaluating the performance of the system is a two part discussion. In terms of the structural model, the FE tools used proved adequate and reliable, and were an obvious choice for future study.

In terms of the acoustic problem to be evaluated, two different FE tools had been used so far. The advanced poro-elastic capability of using the NXNastran–CDH/EXEL combination offered the opportunity to use a realistic geometry in the analysis, however at the time, required substantially more resources in terms of computational power. The hierarchical code on the other hand, while precluding the use of curved surfaces, offered the potential to quicker calculations, in particular when considering more complex, layered acoustic treatments with various material parameters which might be of interest to study.

The difficulty in choosing a suitable approach to evaluate and include the acoustic behavior in the design problem further highlights to a degree the early stage of development of such tools. In this step of the design process the acoustic treatment was a passive component in the design. The goal of the research, however, was to actively include this aspect, and thus an appropriate method needed to be chosen. The conventional model could not eas-
ily be compared with the sandwich concepts more than already shown due the problem of achieving equivalent structural excitation. In addition, calculations with this software took considerably longer and were more complicated in setting up and executing. As mentioned, the hierarchical code offered considerably more flexibility in terms of studying the acoustic treatment. For these reasons, it was decided that future iterations of design process should use the in-house hierarchical code and a more relevant structural excitation. As the accuracy of the code had already been validated, the change in load case used was not considered a difficulty.

Having addressed the question of whether the tools and methods used were sufficient, which led to some obvious need to revisit the method of refinement, it was also interesting to investigate if the functional requirements have been achieved. In terms of the acoustic performance from the standpoint of acoustic absorption, good performance was obtained. For the case of the structural functionality the structural requirements established were fulfilled at a drastically reduced weight. As the final solution is however unsatisfactory, a return to the first step in the process, and a re-evaluation for the functional requirements of the system is necessary.

Iteration 2

As the step by step process has now been discussed in detail, and much of it remains static through the design process, the author feels it unnecessary repetition to review the subsequent iterations on a step by step basis. Instead, a more general review of the changes made will be given and the reader is encouraged to make the connections to the correct step in the design method on their own.

A second interaction initiated using the lessons learnt in the first iteration should have a higher chance of success. In working with such an iterative design approach, it should be remembered that failure to achieve a successful design immediately is not in fact a failure. Each unsuccessful attempt to achieve functionality and fulfil design constraints should be seen as a step towards a more complete understanding of the problem. Especially when dealing with a multidisciplinary design task, it is highly unlikely that all aspects of the problem or consequences of solutions proposed to address them, will or can be understood from the outset.

In the second iteration, the significance of the resulting face sheets from the first iteration were examined further. Clearly, these were too thin to be robust for daily use, but also, it could be interpreted that since so little material was necessary to achieve the structural requirements, a material of lower mechanical stiffness and lower density might be appropriate. The list of functional requirements was augmented as follows:

- The exterior surfaces shall have adequate robustness so as to be unaffected by everyday use
The system shall provide sufficient stiffness where required and avoid excessive stiffness where unnecessary.

The addition of the first of these two requirements, which may be considered obvious and trivial, is an excellent example of how essential characteristics can be overlooked because they are so fundamental as to be taken for granted. In this sense, an iterative approach is helpful.

Glass fibre composite was proposed for both the exterior and interior face sheets as it was both lower density and lower stiffness than aluminium, as used in the previous iteration. A directional laminate was proposed for the exterior face sheet, and a chopped strand mat (CSM) for the interior face sheet. A framework of structural foam was added around the edges of the panel, and a single narrow band transversely through the centre of the panel, to further increase global stiffness. As the previous solution with a perforated face sheet offered potential for improvement, such a solution was chosen for further study and the non-perforated concept was abandoned. Further focus was also placed on the acoustic treatment, and the single layer of visco-elastic foam was split into three separate layers. Figure 19 shows a cross-section of the proposed layup.

To explore the limits of mass reduction in the structure, the material properties of both the outer face sheet and the structural foam were introduced as design variables in the optimization scheme. Fiber volume fraction in each layer of the outer face sheet was used as a design variable to control the mechanical properties of the layer according to equations (5 - 9). This approach was chosen to allow the optimization algorithm to replace the higher density glass fibre with lower density matrix material where the structure would allow.

\[ E_{11} = (V_f)E_{f11} + (1.0 - V_f)E_m \]  

(5)
\begin{align*}
E_{12} = E_{13} &= \frac{V_f}{E_f} + \frac{1 - V_f}{E_m} \quad (6) \\
G_{12} = G_{13} &= \frac{G_m}{1 - \sqrt{V_f(1 - G_m / G_{f12})}} \quad (7) \\
G_{23} &= \frac{G_m}{1 - V_f(1 - G_m / G_{f12})} \quad (8) \\
\nu_{\text{lamina}} &= V_f \cdot \nu_f + (1 - V_f) \cdot \nu_m \quad (9)
\end{align*}

Structural foam density was used to control its mechanical properties according to relations established in the literature [72]. A closed cell foam was assumed using the bulk properties of PET. The relationship between the structural foam’s density and its mechanical properties can be seen in equations (10 - 12). Material properties of the inner face sheet were controlled with the degree of perforation, as shown previously in figure 16 and equation (4).

\begin{align*}
\frac{E_{\text{foam}}}{E_{\text{solid}}} &\approx \phi^2 \left( \frac{\rho_{\text{foam}}}{\rho_{\text{solid}}} \right)^2 + (1 - \phi) \frac{\rho_{\text{foam}}}{\rho_{\text{solid}}} \\
&\quad + \frac{P_0(1 - 2\nu_{\text{foam}})}{E_{\text{solid}} - \rho_{\text{foam}} / \rho_{\text{solid}}} \quad (10) \\
\frac{G_{\text{foam}}}{E_{\text{solid}}} &\approx \frac{3}{8} \left( \phi^2 \left( \frac{\rho_{\text{foam}}}{\rho_{\text{solid}}} \right)^2 + (1 - \phi) \frac{\rho_{\text{foam}}}{\rho_{\text{solid}}} \right) \quad (11) \\
\nu_{\text{foam}} &\approx \frac{1}{3} \quad (12)
\end{align*}

Slight changes were also made to the structural analysis scheme. The method of loading for the localised load was slightly adjusted, and an additional static load case in the form of distributed pressure over the entire panel was also added to increase the robustness of the solution.

To explore the potential of the acoustic treatment, acoustic calculations were added to the optimization scheme to introduce an active acoustic design aspect rather than a reactive evaluation tool. This was done using the hierarchical FE code with a simplified geometry model of the panel. By introducing the acoustic model with a separate finite element solver, the optimization scheme effectively became a two step process. Structural calculations were best addressed using the conventional FE solver, and acoustic characteristics could only be addressed using the hierarchical FE code.
It was decided that the objective for both structural and acoustic optimization should be to minimise the mass of the panel. Constraints for the structural case were the same as previously, with the addition of a displacement constraint on the new load case, and in the acoustic analysis case, a maximum possible sound pressure level (SPL) was permitted within the cavity model for a certain type of dynamic excitation.

The new optimization process required a stepwise iterative approach to achieve convergence of both the structural solution and the acoustic solution to the same design. During structural optimization, acoustic design variables were held constant, and vice-versa.

Results for the design variables and the normalised mass for the second iteration can be seen in Table 5. Without placing too much focus on the exact numerical values, some important conclusions can be drawn. Regarding the fibre reinforcement, in none of the layers did demands for mechanical properties allow for a lower fraction of fibre. Certain layers were however deemed more important which can be seen by the thickness of each layer. Total thickness of the laminate, in this case a quadraxial symmetric layup, is approximately 3mm in thickness. While clearly robust, this could indicate that glass fibre was not the optimal choice and fibres offering higher mechanical properties may be advantageous. For this configuration, structural foam density achieved the minimum allowable value, indicating that the "standard" foam chosen in the previous iteration was substantially over-dimensioned according to the available performance metrics. Mass of the panel has increased somewhat from the previous solution, which should be expected in that further robustness criteria have been added. The mass is however still significantly below that of the conventional solution.

The acoustic optimization maintained a constant maximum level of sound pressure within the acoustic cavity, however it was decided to evaluate the robustness of this solution. The question posed was how sensitive is the acoustic solution to changes in thickness of the layers for this configuration? Acoustic foam layers were allowed to vary and the resulting sound pressure in the cavity was calculated. Results of this sensitivity study can be seen in
What can be seen in these two figures is the obvious fact that for a maximum and minimum thickness of acoustic foam, the maximum and minimum weight penalty and minimum and maximum SPL in the cavity are achieved. What is more intriguing is the results of figure 21, which show that for a configuration with a relatively thick layer 1 and relatively thin layer 3, the SPL is nearly as good as that of the maximum thickness acoustic treatment.

Having completed the final step in the design phase once again, it was necessary to re-evaluate the results, and the methods used. This time, the output of the design process fulfilled all the necessary functional requirements and had a lower mass than the benchmark system. Adding the active acoustic design portions to the problem increased the complexity of the analysis, but added significant worth to the results. In examining the results of the acoustic sensitivity study, it was decided that further examination was necessary. If variations of thickness of two foams chosen based only on a best estimate of applicability, what could a more systematic approach to choosing an acoustic treatment yield in terms of performance, and what effect might it have on the structure?

**Iteration 3**

In the third iteration, it was decided to further investigate the potential for weight savings in the structure by using a higher performance and lower mass composite system, i.e. carbon fibres instead of glass. In terms of the acoustic system, it was decided to investigate the effects of stacking sequence and material choice on the panels performance. Variables which describe the elastic, acoustic, and dissipative properties of the acoustic foam layers were introduced into the optimisation framework. For a detailed description of these variables...
and the exact method used, the reader is referred to the literature [73]. To enable this process, it was necessary to fix the degree of perforation in the inner face sheet, however this was not considered a significant change to either the structural or acoustic problem, sufficient material remained for structural integrity and significant perforation was introduced to achieve acoustic transparency. In addition, it was decided to examine the potential of introducing an air gap between the porous foam and the inner face sheet. This is a method often employed in other forms of acoustic solutions, and has been shown in the literature to increase the level of damping provided by the foam [74].

As the acoustic phenomena should be studied in more detail, it was decided that a change in the structural model should be made to achieve better concurrence between the models. The true geometry structural model used until this point was replaced with a simple, 1/4 flat panel model with symmetry constraints. Load cases were retained constant, and boundary conditions adjusted for best correspondence between the structural and acoustic models. Acoustic materials in the structural model were modelled as a single isotropic layer. In the acoustic model, the outer face sheet was modelled as a single anistropic layer with
equivalent properties for the laminate. A cross-section of the configurations studied is shown in figure 22.

![Figure 22: Cross-section of configurations studied](image)

The iterative approach of first structural and then acoustic optimization was repeated with one significant change. Rather than minimizing mass for a maximum SPL as was done in the previous iteration in the acoustic optimization step, the SPL in the cavity was made the objective function. Limits on the design variables used were the only constraints used in the acoustic optimization.

The results of the optimization proved quite interesting. The first and most significant result of the new modelling scheme was seen immediately; that the double curvature of the previous structural model played a significant part in its mechanical robustness. This is perhaps another key example of why an iterative procedure should be used to avoid missing what might be considered obvious. While it was understood that this would be a factor, it was not understood initially the degree to which it would affect the structural performance. In fact, using the previous core configuration with a single transverse beam of foam proved inadequate for reasonable values of face sheet thickness. An additional lengthwise foam beam was added, effectively enclosing the acoustic foam treatment in a "window frame" structure of structural foam. This configuration, though chosen arbitrarily, proved more reasonable in achieving acceptable values of face sheet thickness.

The results in table 6 gives an overview of some of the resulting design variables. While the exact numbers shown for the structural variables are uninteresting, it should be noted that in certain layers of the laminate, carbon fibres have been replaced by matrix material and that the thickness of all layers was minimised. The appropriate density of structural foam also varied between the configurations. Thickness of the inner face sheet also varied due to the effect of the coupling to the highly compliant acoustic foam layers. The effect of this coupling proved in fact to be a central element in the panel’s behavior. As seen in table 7, the first eigen mode of the panel varies significantly depending upon the presence or absence of the air gap. Indeed, the first mode of vibration was also distinctly different.
Table 6: Final values for design variables in the 3rd iteration

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Variable</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$V_{f0,90}$</td>
<td>0.60</td>
<td>0.60</td>
<td>0.60</td>
<td>0.60</td>
</tr>
<tr>
<td></td>
<td>$V_{f45}$</td>
<td>0.27</td>
<td>0.26</td>
<td>0.29</td>
<td>0.29</td>
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<tr>
<td></td>
<td>$t_{0,45,90}$</td>
<td>0.15</td>
<td>0.15</td>
<td>0.15</td>
<td>0.15</td>
</tr>
<tr>
<td></td>
<td>$t_{PETfoam}$</td>
<td>25.0</td>
<td>25.0</td>
<td>25.0</td>
<td>25.0</td>
</tr>
<tr>
<td></td>
<td>$\rho_{PETfoam}$</td>
<td>134.2</td>
<td>128.2</td>
<td>142.5</td>
<td>140.5</td>
</tr>
<tr>
<td></td>
<td>$t_{CSM}$</td>
<td>2.91</td>
<td>3.50</td>
<td>2.86</td>
<td>3.50</td>
</tr>
<tr>
<td></td>
<td>$\rho_{PU}$</td>
<td>38.62</td>
<td>137.97</td>
<td>137.74</td>
<td>137.97</td>
</tr>
<tr>
<td></td>
<td>$\sigma_{PU}$</td>
<td>5.45e3</td>
<td>1.11e5</td>
<td>1.10e5</td>
<td>1.11e5</td>
</tr>
<tr>
<td></td>
<td>$t_{PU}$</td>
<td>23.04</td>
<td>48.00</td>
<td>47.19</td>
<td>41.49</td>
</tr>
<tr>
<td></td>
<td>$\rho_{pi}$</td>
<td>9.31</td>
<td>1.48</td>
<td>2.46</td>
<td>3.86</td>
</tr>
<tr>
<td></td>
<td>$\sigma_{pi}$</td>
<td>1.39e6</td>
<td>2.94e4</td>
<td>8.29e4</td>
<td>2.12e5</td>
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<tr>
<td></td>
<td>$t_{pi}$</td>
<td>26.96</td>
<td>1.000</td>
<td>2.4617</td>
<td>4.585</td>
</tr>
</tbody>
</table>

Table 7: Results of optimization

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Value</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Total Thickness [mm]</td>
<td>79.1</td>
<td>78.7</td>
<td>78.7</td>
<td>75.8</td>
</tr>
<tr>
<td></td>
<td>Total Mass [kg]</td>
<td>18.7</td>
<td>27.3</td>
<td>27.8</td>
<td>26.7</td>
</tr>
<tr>
<td></td>
<td>SPL [dB]</td>
<td>60.1</td>
<td>59.3</td>
<td>57.9</td>
<td>58.5</td>
</tr>
<tr>
<td></td>
<td>First Eigen Mode [Hz]</td>
<td>71.8</td>
<td>46.9</td>
<td>64.7</td>
<td>47.0</td>
</tr>
</tbody>
</table>

For configurations with an air gap, the inner face sheet began to vibrate and its thickness
was dictated by this fundamental vibration mode. For the bonded configuration, the panel
behaved more as a unified structure and the mode of vibration was a global mode for the
entire panel.

The variation in acoustic variables, and the resulting mass and SPL are also rather striking.
By simply changing the stacking sequence of the foams, large differences in mass, and
significant differences in sound pressure were obtained. While the highest and lowest SPL
was obtained for the lightest and heaviest panels respectively, the difference in SPL cannot
be accounted for in terms of mass alone. In this sense, while differences in geometry
prevent a direct comparison to the original solution, the lightweight panel would be most
attractive despite its noticeably poorer acoustic performance. Properties of the foams prove
here to be a crucial attribute in the panels acoustic functionality. This again highlights the
dangers of using an arbitrarily chosen acoustic treatment and the need for a more systematic
approach.
In assessing the overall results from this iteration, it can be said that the general methods of using material properties and thicknesses for both the acoustic and structural optimization appear rather promising. The fundamental problem with the solution however is the arbitrarily chosen "window frame" configuration of the structural foam core. The chances of such a design being the optimal lightweight solution are very low. In fact, it is most likely a contributing factor to the large difference in mass, and the significantly higher structural foam density in comparison to the previous iteration. One might experiment with such a configuration, varying the width, number, and placement of such beams, however this would be a very unstructured approach and require a certain degree of luck. Such an ad-hoc process might be considered the anti-thesis of the present paradigm of multifunctional design. A more structured, logical approach for developing the structural framework should thus be implemented.

**Iteration 4**

In the fourth and final iteration of the design loop discussed within this thesis, the topology of the structural foam core was examined in detail. In previous iterations, an rather arbitrarily chosen "window frame" design was used. The limitations of such a design became expressly apparent, firstly when switching to a flat panel model from a double curved surface, and secondly when the air gap was added to the acoustic treatment. While the window frame concept was functional, it was most certainly not optimized regarding any of the relevant performance metrics discussed.

In addition, results from the previous iteration indicated that the air gap configuration examined added negative attributes to the structural performance, but did appear to offer a slight improvement in terms of acoustic functionality. One difficulty in assessing the air gaps affect was its strong influence on the structures overall behavior. The author would assert in fact, that the introduction of such an air gap fundamentally changes the structures behavior, to a lesser degree in static load cases, but to a largely significant degree in its dynamic response. This makes the comparison between an air gap and non air gap configuration of the window frame design difficult if not impossible from a combined structural and acoustic functionality standpoint.

Topology optimization was the approach chosen to deal with these problems. As mentioned in the previous chapters, the BESO method was chosen as a starting point for this task. The BESO method is well documented [59, 75, 76, 60, 77, 78, 79] and as a detailed discussion of its principles would not add worth to the present discussion, the interested reader is directed to the literature. For the sake of understanding however, a few simple principles will be reviewed.

In short, the BESO method strives to create an effectively stressed structure by adding or subtracting element within a finite element model based on their so called sensitivity number. Sensitivity numbers differ according to the type of load case assessed, and for the case of multiple load cases, a method of normalising sensitivity numbers between load
cases is used.

For static cases, the sensitivity number, $\alpha$ for each element is calculated according to [76]:

$$\alpha_i = \frac{1}{2} \{u^i\}^T [K^i] \{u^i\}$$  \hspace{1cm} (13)

Where $u^i$ is the displacement and $K^i$ the stiffness matrix of the $i$th element. This sensitivity number is equivalent to the element strain energy for a given element, and is readily available within most commercial FEA software.

For normal modes analysis, the sensitivity number is calculated in a slightly different manner [75]:

$$\alpha_i = \frac{1}{m_i} \{u^i\}^T [\omega_i^2 [M^i] - K^i] \{u^i\}$$  \hspace{1cm} (14)

Where $u^i$ is the normalised displacement, $[M^i]$ the element mass matrix, $[K^i]$ the element stiffness matrix, and $m_i$ the actual mass of the $i$th element. $\omega_i$ is of course the eigen frequency.

Until this point in the design process, a load case for transverse loading of the panel within the optimization loop was not included. The panels functionality in this regard had in fact been assessed, and included to some degree in the arbitrary window-frame design, however this functionality was passively assessed rather than actively designed, much like the acoustic treatment in the first iteration. In adopting a topology optimization framework, the omission of such functionality was readily apparent.

According to the principles of topology optimization, material will only be placed where stresses in the structure dictate it is necessary. If no load case is present which stresses the structure in the desired manner, one cannot expect the algorithm to interpret the designers intentions and allot material for such a load case. For the present study, in-plane loading could have been applied using a static load case and existing BESO algorithms. Static loading was deemed inappropriate however, as in the case of side impact, the load is highly non-static and the structure is likely to behave in an unstable manner and most certainly deform out of the plane of loading. This sort of behavior cannot be predicted using static load cases in a linear elastic model. This implied the need to study the buckling problem which was done using linear buckling analysis in Abaqus CAE 6.9.

Within the previously mentioned literature on BESO, a method for calculating the element sensitivity number for the case of unstable buckling of plates and truss-like structures has also been proposed. This method, however, requires access to the geometric stiffness matrix of the model. As no method of obtaining such information from the commercial FEA tools used could be found, an alternative way of obtaining an element sensitivity number was necessary. This was the basis for the first adaptation of the BESO method to the present application.
Simply stated, the proposed adaptation used the buckling deformation predicted by the FEA solver, i.e. the eigen mode, and its corresponding element strain energy to calculate the sensitivity number according to equation (13). The justification for this was that the out of plane deformations predicted would provide relevant strain energy so as to inhibit such unstable behavior. To include the sensitivity number in the global solution scheme however, a method of scaling it to match the other cases studied was necessary. This was done using the critical buckling load, i.e. eigen value, in the exact same manner as the eigen value in the normal modes analysis case was used. Scaling sensitivity numbers case by case according to the degree of fulfilment of constraint allows multiple cases to be included in the same analysis. With the exception of the buckling sensitivity proposed herein, these methods are well developed within the previously mentioned literature.

The second adaptation to the BESO method was to accommodate for the structural contribution of acoustic foam in the core of the panel. In the standard BESO method, elements are simply included in or excluded from the mesh that is analysed depending on whether they are deemed necessary based on the element sensitivity criteria. This standard method was appropriate for the air gap configuration, as there is no coupling between the acoustic foam and the inner face sheet, thus no shear loads can be transferred and the contribution of the acoustic foam is negligible. For the non-gap configuration, rather than remove the elements deemed unnecessary for structural purposes from the mesh, their material properties were swapped to those of a homogenised acoustic foam of significantly lower stiffness than the structural foam. A mathematical justification of this swap functionality was not done, rather the results of implementing such a change were studied using smaller models, and deemed reasonable. As the use of any optimization scheme, topology optimization especially, cannot guarantee that the absolute optimal solution has been reached, this adaptation was considered justified in the context of the work performed.

Using the same load cases as in the previous iterations, and adding a load case for in-plane loading of the panel, a topology optimization of the panel was performed using the proposed adaptation of the BESO method and a 1/4 model with symmetry constraints. This procedure resulted in two different topological configurations which can be seen in figure 23.

The results of the topology optimization provided a new starting point to begin the same two step structural and acoustic optimization loop as performed in the previous iteration. In this sense, the use of the BESO method offered the practical benefit of providing a complete mesh for the structural optimization step. For the acoustic optimization step however, a new mesh needed to be constructed using larger elements due to the finite element basis used for the acoustic calculations. A comparison of the structural and acoustic meshing of the structural foam core can be seen in figure 24. For this iteration of acoustic optimization, a slightly different choice of acoustic foam configuration was studied based on the results of the previous iteration. The same two materials were used however a stacking sequence with two layers of a single material was tried in both the gap and non-gap configuration. A schematic diagram of the stacking sequences studied can be seen in figure 25.

Tables 8, 9 and 10 show the results of the design variables and global properties of the four
panel configurations studied. For the structural design step, the general implications of the method of achieving the results has to a large degree already been discussed in the previous iteration. To a certain degree, this is also true of the effect of the air gap on the structural design variables. The top sheet laminates obtained were of nearly identical thickness, but of significantly differing mechanical properties. While not immediately obvious from inspection of the design variables, the non-gap configuration proved to have much lower stiffness than the gap configuration. Density of the structural foam also varied somewhat, being slightly higher and thus slightly stiffer in the air gap configuration. These were the two primary differences between configurations and would tend to further reinforce the notion that the acoustic foam treatment does indeed contribute to the structural efficiency of the panel.

Differences in the acoustic foam design variables were much more striking. As seen in table 9, thickness, density, and flow resistivity varied considerably between configurations. In addition, it was found that by using one acoustic material in two layers, improved results were obtained by varying the properties of the foam. Presumably, this is to achieve changes in relative acoustic impedance which agrees well with current theories within the field of porous materials in acoustic applications.

Returning to the overall global objective of the methodology, the results in table 10, are perhaps much more interesting. From a structural perspective, all four configurations have
Figure 24: A comparison of structural (above) and acoustic (below) FE meshes of the structural foam core material.

Figure 25: Conceptual visualisation of the four different configurations in the acoustic optimization. Note that structural foam topology (dark grey) differs between the air gap and non air gap configurations.

fulfilled the functional requirements equally well. From the acoustic perspective, there exists a quite significant difference in performance, approximately 5.6 dB, between the best and worst performing solutions. Also striking is the significant difference in mass, nearly 18 kg, between the heaviest and lightest panel. Special attention should be paid to the fact that this figure is for the entire panel and not only the 1/4 model. That there
Table 8: Final values of structural design variables

<table>
<thead>
<tr>
<th>Variable</th>
<th>Configuration</th>
<th>Air Gap</th>
<th>No Air Gap</th>
</tr>
</thead>
<tbody>
<tr>
<td>$V_{f0,45,90}$</td>
<td>0.000-0.600</td>
<td>0.598,0.361,0.388</td>
<td>0.596,0.597,0.071</td>
</tr>
<tr>
<td>$t_{0,45,90}[\text{mm}]$</td>
<td>0.010-2.875</td>
<td>0.320,0.016,0.563</td>
<td>0.129,0.222,0.350</td>
</tr>
<tr>
<td>$t_{PET\text{foam}}[\text{mm}]$</td>
<td>5.000-75.000</td>
<td>74.926</td>
<td>74.761</td>
</tr>
<tr>
<td>$\rho_{PET\text{foam}}[\text{kg/m}^3]$</td>
<td>50.000-300.000</td>
<td>120.359</td>
<td>105.324</td>
</tr>
<tr>
<td>$t_{CSM}[\text{mm}]$</td>
<td>0.500-5.000</td>
<td>0.654</td>
<td>0.675</td>
</tr>
</tbody>
</table>

Table 9: Final values of acoustic design variables

<table>
<thead>
<tr>
<th>Acoustic Layer 1 - PU foam</th>
<th>Configuration</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\rho_1^*$ [kg/m$^3$]</td>
<td>36.34</td>
<td>13.51</td>
<td>6.801</td>
<td>5.009</td>
<td></td>
</tr>
<tr>
<td>$\sigma_1^{\text{static}}$ [kg/m$^3$/s]</td>
<td>4.766e3</td>
<td>582.4</td>
<td>142.0</td>
<td>76.26</td>
<td></td>
</tr>
<tr>
<td>$t_1$ [mm]</td>
<td>72.9</td>
<td>1.00</td>
<td>1.00</td>
<td>4.08</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Acoustic Layer 2 - PU or pi foam</th>
<th>Configuration</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\rho_2^*$ [kg/m$^3$]</td>
<td>5.286</td>
<td>138.0</td>
<td>1.964</td>
<td>27.90</td>
<td></td>
</tr>
<tr>
<td>$\sigma_2^{\text{static}}$ [kg/m$^3$/s]</td>
<td>4.103e5</td>
<td>1.109e5</td>
<td>5.228e4</td>
<td>2.685e3</td>
<td></td>
</tr>
<tr>
<td>$t_2$</td>
<td>1.00</td>
<td>72.9</td>
<td>73.8</td>
<td>70.7</td>
<td></td>
</tr>
</tbody>
</table>

should be a tradeoff between mass, and acoustic performance is to be expected, however the magnitude of mass difference in this case is extreme. While no direct comparisons can be made with these results and the original vehicle studied, it is apparent that from a mass savings perspective the layup of the acoustic treatment is critical.

Table 10: Results of optimization

<table>
<thead>
<tr>
<th>Configuration</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Value</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total Thickness [mm]</td>
<td>77.4</td>
<td>77.4</td>
<td>77.3</td>
<td>77.3</td>
</tr>
<tr>
<td>Total Mass (entire panel) [kg]</td>
<td>18.2</td>
<td>31.6</td>
<td>14.0</td>
<td>17.1</td>
</tr>
<tr>
<td>SPL [dB]</td>
<td>70.5</td>
<td>68.7</td>
<td>74.3</td>
<td>71.6</td>
</tr>
</tbody>
</table>

Also of interest is the frequency response of the different configurations across the entire frequency range, as shown in figure 26. Here, the differences between the configurations and stacking sequences becomes even more apparent. Despite being optimized to fulfil the same constraints resulting in principally equivalent structures, the dynamic response of
the systems in the acoustic sense varies considerably. As can be seen, the frequencies of
peak response vary somewhat between all configurations. Observing this phenomena, the
question of which configuration is actually superior also becomes more difficult to answer.
While it can be determined which configuration provides the lowest SPL for the given ex-
citation, it is not as clear which configuration would be "best" from the driver or passengers
standpoint. This varying form of response also emphasising the need to use a broad range
of frequencies in sufficient resolution to obtain a good result from optimization. Studying
a single frequency or a small band of frequencies, or using too low frequency resolution
could result in the optimization missing one of the peaks, which could contribute signifi-
cantly to the overall acoustic performance or occur at a particularly troubling frequency.

![Figure 26: Frequency response function of acoustically optimized configurations.](image)

In assessing the overall results of the fourth iteration, a few general conclusions can be
made. Firstly, the topology optimization scheme used, while not a guarantee to achieve the
ultimate optimal solution, offers a significant step towards improving the solution. Further
evidence for the idea that the low stiffness acoustic foam contributes to the structural per-
formance was also found in terms of the stiffness of the outer face sheet, and the structural
foam. In the acoustic optimization, the air gap does seem to provide additional acoustic
performance but at the cost of extra mass compared to the non air gap configuration. In addition, one rather obvious fact was that the stacking sequence of acoustic foams was critical to both the acoustic performance, and even more so to the final mass of the panel.

In this iteration, the mass of the panel became a much more sensitive quantity. Depending on which configuration in studied, the mass may have increased or decreased significantly compared to the previous iteration. It can be stated that the structural mass of all four configurations is significantly improved compared to the previous iteration, and nearly as low as in the very first iteration. This trimming of structural mass has occurred despite the removal of geometric stiffening effects such as double curved surfaces, and significantly more stringent structural load cases. The crux of the problem however lies in mass of the acoustic treatment necessary to maximise such a structures acoustic performance. This result highlights both the promise of using the two-stage iterative optimization scheme, as well as the difficulty in achieving a balance between the structural and acoustic requirements.

In evaluating the method in itself as used in the final iteration, it can be said that good results were obtained which for the most part, fulfil the functional requirements which have been developed throughout the process. In the context of the current work, the author would assert that the method has matured to the degree that further iterations are not necessary to fulfil the objectives set out from the beginning, to create a design tool for simultaneously addressing the conflicting structural and acoustic requirements present in modern vehicle body design.
5 Conclusions and Summary

In the development of the design process explained herein, a number of distinct contributions to the areas of engineering science associated with modern vehicle design have emerged.

The single largest contribution of the present work lies in the iterative design method which has been proposed, and explained using a vehicle roof as a case study. Here, the paradigm of multifunctionality has been successfully implemented, and a process has been developed so that a single engineer can effectively and efficiently produce a structural vehicle body panel which effectively balances both structural requirements and acoustic performance. One of the central concepts highlighted in the method is the focus placed on system functionality rather than component performance. In the current context, a successful design can only be obtained by simultaneously addressing all the functional requirements of the system. Focusing on a single functional requirements independently can only lead to a final result full of compromise and optimal for none of its tasks. In the present context, from the perspective of the structural and acoustic requirements of a modern vehicle BIW, a great majority of these functional requirements have been successfully addressed. The iterative nature of the process is fundamental in helping the engineer using it to understand the design problem at hand, which in the case of a multidisciplinary problem, may be too difficult.

Furthermore, the concept of a multifunctional body panel has been defined and a considerable body of knowledge and understanding has been created and presented within the present work. In the research performed here, the concept shows substantial promise in helping to reduce vehicle weight while maintaining the structural and acoustic performance. This weight reduction will prove central to the vehicle industry meeting the fuel consumption and emission requirements in the future, both in the short term and long term.

In addition, in this thesis it has been shown that the realisation of such a multifunctional body panel requires a level of concurrency which up to now has not been state-of-the-art, neither in academic research nor in industrial practice. Its realisation in the frame of the present work has involved the establishment of a multidisciplinary methodology based on lightweight materials, sandwich construction, elasto-acoustic models of the foam materials studied and numerical analysis and optimization tools. In addition, modelling methodology
has been introduced enabling the design of tailored elastic and acoustic properties, for the lightweight materials of interest in the current context.

As the research work discussed has progressed and deepened, it became clear that in order to achieve multifunctionality in the current sense, a three dimensional perspective of the topological layout of the body panel investigated was necessary. While initial concepts, inspired by current vehicle designs, explored the sizing of multiple layers through the thickness of the panel, the complexity of the topological layout resulting in the final concept studied holds a substantial promise for future design studies in this area of application.

**Further Implications**

While not the main objective, a number of contributions to the area vehicle NVH were made. The first of which was regarding the acoustic testing of the roof, i.e. an acoustically trimmed panel. Here, results of in-situ testing showed excellent agreement with that of component testing. This implies that the need for removing panels, or disassembling a vehicle for such testing is unnecessary, which could save both time and money in development. It was also found that above the frequency of approximately 1000Hz, the windows become a significant contributor to the overall sound transmission loss between the passenger compartment and the environment. Regarding the use of FE in predicting the behavior of trimmed body panels in the existing vehicle, several conclusions can be made. Firstly, using the hierarchical FE solution to calculate sound transmission loss was an efficient method up to the frequency of approximately 500 Hz, after which the simplicity of the model and the method of excitation was no longer accurate enough. Regarding the method of modelling employed using NXNastran, i.e. using equivalent properties, while simple, proved to be accurate from a vibro-acoustic standpoint. The two methods used to introduce damping into the trim panel, using a simple surface impedance or advanced poro-elastic material tools, provided good results. The impedance method, while simple, proved accurate and relatively simple to implement. The poro-elastic method, while slightly more complicated, was also accurate, and offers significantly more room for improvement. The first method may offer a starting point, or even a compliment to the more advanced form. Generally speaking, modelling of interior trim components using three dimensional finite element models is still in its infancy for both academic and industrial applications. This observation is based on a thorough literature survey and discussions with experienced individuals in industry. Accounting for the acoustic and vibrational aspects of interior trim by modelling the actual components is the next frontier within vehicle NVH and offers a powerful tool for prediction during design. In particular, the ability to do this quite accurately without any additional modelling tools may have significant implications.

Using some creative thinking, it is quite possible to imagine further potential areas of improvement a modular sandwich based construction could achieve for the case studied in this thesis. Among other things, it is reasonable to presume that improvements within the following areas should be attainable:
• Reduced assembly line time
• Improved assembly ergonomics
• Improved aerodynamic drag via reduced frontal profile OR
• Improved interior headroom for the same frontal profile
• Increased styling flexibility (assuming use of composites materials)

While these areas of potential improvement are perhaps not fundamental to vehicle construction, they do add further worth to the concept to encourage its further development.
6 Future Work

The work within this thesis has focused on the development of design methods and tools to create multifunctional vehicle structures. The "blue sky" goal of the work has been to find ways to reduce the economic and ecological impact of vehicles within the foreseeable future. To a certain extent, this has been achieved, however due to the level of complexity of the problem, this has been mainly achieved through mass reduction. Not all materials are created equal in terms of density and mechanical properties, nor in terms of their effect on the environment. This aspect is often addressed using tools as life cycle analysis (LCA), often employed by environmental scientists, mostly as a means of planning large scale projects or assessing existing products. Correctly applying such tools requires significant knowledge and experience. Including a complete LCA as an active component within the design process of a vehicle is not currently possible, or in some senses even meaningful. There exists however, a possibility that some of these tools and methods could be helpful and applicable to engineering problems. Perhaps the next frontier in ecological and economical design could involve importing certain elements of such tools into the optimization design process presented here. It is my industrious objective to explore the possibilities in doing so in the near future.

It could also be stated that the design methodology proposed is by no means an exhaustive and all inclusive method. Room for improvement exists, and additional functionality, such as manufacture-ability, thermal efficiency, or recycle-ability, could, and should be included in future incarnations. The only primary rule in implementing further analysis and areas of study to the method is that the overall behavior of the system must remain the focus. If the weighting of an individual functionality or area of performance becomes too large when comparing the performance metrics in the design process, then the process is no longer multi-functional. This would mean a return to the state of things as they have been, which is not the path to follow should progress be made.
Bibliography


Division of work between authors

Paper I

Measurements were performed by Cameron and with assistance from Saab and MWL Personnel. Post Processing of results was performed by Cameron. Nastran modelling was performed by Cameron and Hierarchical modelling by Göransson. Analysis and interpretation of results was jointly performed by Cameron and Göransson. Cameron wrote the paper with support from both Göransson and Wennhage.

Paper II

Modelling and analysis in Nastran was performed by Cameron. Cameron and Wennhage developed the optimization scheme together and Cameron executed it. Göransson provided the basis for the poro-elastic modelling. Cameron and Göransson implemented the acoustic analysis together. Cameron wrote the paper with support from Göransson, Wennhage and Rahmqvist.

Paper III

Cameron developed the design methodology. Cameron and Lind set up the iterative optimization scheme together with input from Wennhage and Göransson. Cameron performed structural analysis and Lind performed acoustic analysis. Cameron wrote the paper with support from Wennhage and Göransson.

Paper IV

Cameron and Lind-Nordgren set up the optimization scheme with input from Wennhage and Göransson. Cameron performed structural analysis and Lind-Nordgren performed
acoustic analysis. Cameron and Lind-Nordgren compiled the results and wrote the paper with input from Wennhage and Göransson.

**Paper V**

Cameron implemented the topology optimization scheme and defined the structural problem. For the iterative structural acoustic optimization, Cameron was responsible for the structural optimization and Lind-Nordgren for the acoustic optimization. Cameron and Lind-Nordgren wrote the paper with input from Wennhage and Göransson.