

A Design Methodology for the Utilisation of Structural Orthotropic Composite Materials for Low Volume Production

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Any inadequacies that remain in this work are my own – sorry!

Abstract

Numerical techniques have transformed the design process. It is now possible to carry out analysis that, until relatively recently, was thought to be impossible. This work demonstrates how these techniques can be used by Small and Medium Enterprises (SMEs) to deliver better products. This is demonstrated by taking lessons in Finite Element Analysis (FEA), that have been learnt from the Automotive and the Aerospace industries and applying them to the case study of developing a Kitesurf board. The Kitesurf industry is an example of a business that traditionally uses an iterative approach to manufacture physical prototypes to develop a product. With recent reductions in hardware and software, such small industries, could and should, be using the numerical techniques that have been developed elsewhere, for their products. This work shows that this is possible, the benefits of doing this, and how to develop products in such a way.

At the same time that numerical techniques have been developing, there have been significant increases in the use of composite materials. These orthotropic materials present great challenges when developing products that historically have been manufactured using isotropic materials. However, these also provide great opportunity to create more efficient structures if numerical techniques are used to optimise these materials correctly. To demonstrate this, this work not only designs a product that would be developed by SMEs, in an efficient manner, it also does this with a product that is manufactured from orthotropic materials to take advantage of the optimisation techniques that are available.

Material properties for the composite used in this study are established through correlation to vibration tests and then verified through ultrasonic testing techniques and these are then fed into the numerical techniques used subsequently in the case study presented. These numerical techniques show how a product should be manufactured so that it has the desirable mechanical responses that are initially established in this work. It develops an experiment for this industry, in a controlled laboratory environment, to mimic what the product would be exposed to during use. It then goes on to show how typical boards would behave in this environment and also shows that the proposed product is fit for purpose.

To verify the proposed methodology, scale representations of the physical products discussed in this work are manufactured and tested. An arbitrarily laid-up representation of a kitesurf board is manufactured and tested. An alternative board that has the same structural characteristics is developed and manufactured using the proposed design methodology. The board that is developed using the proposed design methodology in this work, is shown to be more efficient whilst maintaining the desirable structural characteristics in both the FE models and the physical parts.

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Glossary

ABS	Acrylonitrile Butadiene Styrene
CAD	Computer Aided Design
CAE	Computer Aided Engineering
CFRP	Carbon Fibre Reinforced Plastic
CMM	Co-ordinate Measuring Machine
CNC	Computer Numerically Controlled
DoE	Design of Experiments
DOF	Degrees Of Freedom
DTI	Dial Test Indicator
E1	Young's Modulus (stiffness) in the fibre direction
E2	Young's Modulus (stiffness) perpendicular to the fibre direction
E3	Young's Modulus (stiffness) perpendicular to the plane of the fibres
FE	Finite Element
FEA	Finite Element Analysis
FEM	Finite Element Model
FFT	Fast Fourier Transform
G	Shear Modulus (stiffness)
I	Second Moment of Area
J	Polar Second Moment of Area
LDV	Laser Doppler Vibrometer
NDT	Non-Destructive Testing
NVH	Noise Vibration and Harshness
OEM	Original Equipment Manufacturer
PU	Polyurethane
SME	Small or Medium Enterprise
UD	Uni-Directional
UT	Ultra-sonic Testing
UWTSD	University of Wales Trinity Saint David

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Resources Used

This work was carried out at The University of Wales Trinity Saint David (UWTSD).

The following major resources were used. Note that small items such as scales etc. are not listed;

Software

- FEA: Vrand Design Studio and Genesis.
- CAD: Dassault Solidworks.
- Vibration Analysis: Polytec PSV.

Hardware

- Laser Doppler Vibrometer: Polytec PSV-400.
- Composite Manufacturing: Autoclave Serial No. AC062.
- 5 Axis CNC Machine: CMS Serial No. 5221.
- Phased Array Ultrasound: Omniscan MX2 & 1MHz Transducers.
- CMM: KEMCO 400E with QCT software.

Publications as a Result of the Research

It should be noted that some sections of this work have been presented and discussed previously by the author as follows;

Section 3.3: This contains material presented by the author at the 8th World congress on structural and Multidisciplinary Optimisation in 2009 (1), the International Journal of Engineering Simulation in 2010 (2) and the 13th International Conference on Modern Information Technology in the Innovation Processes of the Industrial Enterprises in 2011 (3).

Section 3.5.1: This was presented at the “Engineering of Sport (ISEA10)” Conference in Sheffield, U.K. in 2014 (4).

A summary of the progress of the work was presented at the 2nd composites in sports conference in Loughborough in September 2018 (5).

The above published journal articles and conference proceedings written by the author are included in the appendices of this work.

1 Introduction

Optimisation is now used routinely in the Prescriptive design methodologies used by the Automotive and the Aerospace industries. Traditionally this has been used in large multinational organisations (as opposed to SMEs), partly due to the high costs of the hardware and the software needed in order to carry out this analysis. With recent advances in hardware and software, this should now be used in the development of products that are produced by SMEs that traditionally have not been able to afford this. The work presented demonstrates this through the use of these techniques being applied to the Structural Orthotropic Composite Materials in Kitesurf boards.

1.1 Background to Kitesurfing

Kitesurfing is a relatively new sport that has been made possible by the advances in the equipment used in this sport. A kitesurfer can be seen in Figure 1-1.



Figure 1-1: Kitesurfing (6) (7).

The participant uses the force generated by a kite to pull them across the water through lines that are attached to the kite and a waist harness. The participant rides a kiteboard such as the one shown.

In recent years, board sports such as snowboarding, wakeboarding and kitesurfing have seen significant levels of participation. 115 million people participated in winter sports worldwide in 2013 according to an online magazine that focusses on the board sport industry (8). In the US, the number of people participating in recreational surfing grew by 49.4% between 1987 and 2001 and skateboarding was identified as the fastest growing sport with more than 10.1 million participants (9). At the same time, there have been significant increases in the use of advanced composite materials such as carbon fibre reinforced plastics in the manufacture of boards. These boards range from those aimed at the novice rider, to those aimed at an experienced rider. What this relates to in terms of the structural performance of the board, for example the torsional stiffness, bending stiffness and natural frequency characteristics etc. remains largely unpublished. This area of research has a limited amount of published data on some more established sports such as snowboarding and ski-ing, however in contemporary sports such as kitesurfing, this does not appear to have been the subject of major research previously. The construction of these types of boards can be seen in Figure 1-2.

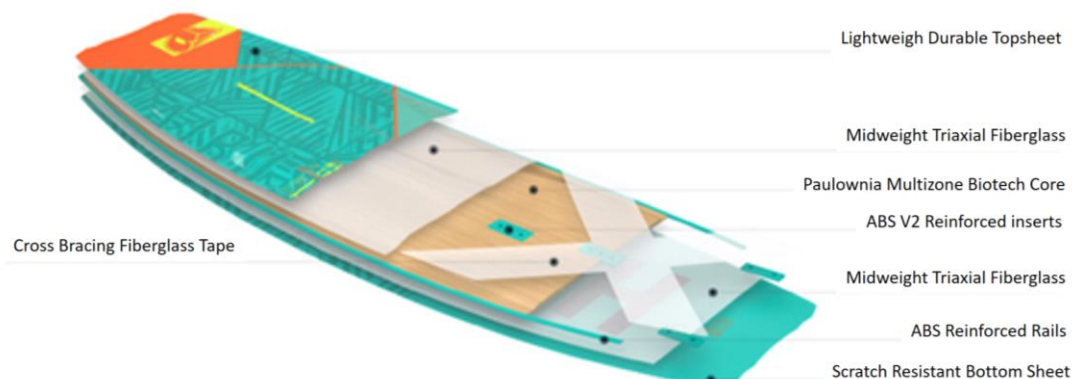


Figure 1-2: Construction of a Kitesurf Board (10).

These boards are a complex assembly of the following layers;

1. A non structural graphics layer.
2. Structural Layers of orthotropic composite materials – typically glass fibre reinforced plastic and/or carbon fibre reinforced plastic.
3. A lightweight wooden core to provide damping and to increase the second moment of area of the section.
4. Further structural Layers of composite materials – typically glass fibre reinforced plastic and/or carbon fibre reinforced plastic.
5. A final non structural graphics layer.

These boards are also fitted with Acrylonitrile Butadiene Styrene (ABS) or Polyurethane (PU) 'rails' to seal the above layers and provide protection to the board. These rails are also extended inwards to provide mounting points for the fins which are used on these boards, finally steel inserts are embedded in the structure to provide mounting points for the bindings used on these boards.

Despite the complex materials and the many layers that are used in the construction of these boards, according to Poodts et al, they appear to be manufactured using trial and error in order to get the properties that the manufacturer requires (11). This type of approach involves many iterations of manufacturing prototype (or 'approval') boards as highlighted by Reinforced Plastics magazine (12), getting experienced riders to test them and then giving subjective feedback to the designers for the next iteration, until a final board is obtained that is suitable for the target market. Not only is this process inefficient in terms of development times, the finished product is almost certainly heavier than it needs to be and uses too much material, as optimisation of the layup of the composite is far from a trivial task. It is hypothesised that with recent advances in

hardware and software that this should be achievable via a prescriptive design methodology. This work thus applies experimental testing techniques to establish the mechanical properties of materials used in this product and the structural characteristics of two existing boards in order to understand the desired responses. A design methodology is then proposed using finite element analysis and optimisation techniques to replicate the properties of the benchmark board, before manufacturing a representative sample to validate the proposed methodology.

1.2 Aim and Objectives

1.2.1 Aim

The aim of this work is to demonstrate how, by using modern numerical techniques and in particular, FEA and optimisation techniques, it is possible to produce better products whilst at the same time reducing the development time. This is achieved through the case study of designing a Kitesurf board. In order to apply numerical techniques to this case study, the required structural characteristics of kiteboards are researched so that these can be replicated.

1.2.2 Objectives

The research in this work has the following objectives;

1. Demonstrate how bringing Computer Aided Engineering (CAE) forward during design process makes this more efficient and effective.
2. Show how and why the methodology used by multinational companies for the development of their products should now be used by SMEs.
3. Carryout a case study to design a kiteboard using the proposed methodology in point 2. This case study involves the following.

- a. Establishing the mechanical properties of the composite material that is utilised to replicate the board used throughout this work. Including;
 - i. Establishing the elastic properties of carbon fibre samples using ultrasonic testing techniques.
 - ii. Testing the natural frequencies of carbon fibre samples to investigate how this is influenced by the lay-up.
 - iii. Creating and correlating FE models of the tests above.
- b. Component level testing of the board is replicated in FE models during optimisation. In order to do this, two boards are tested that are aimed at different styles of riders so that the significant differences in their mechanical behaviour can be understood so that these can be replicated in the board that is designed. As such, the following is determined;
 - i. Bending stiffness of boards aimed at different riders.
 - ii. Torsional stiffness of boards aimed at different riders.
 - iii. Natural frequency of boards aimed at different riders.
- c. Mock field testing to establish the behaviour of the structures during use.
- d. Propose and use a design methodology to recreate the characteristics of a board aimed at a particular rider as established in b.
- e. Propose a design for a board developed using the proposed methodology in d. FE models of this board are tested to show that this responds in the same way as the benchmark board.
- f. Manufacture and test a sample representative of the board that has been developed in point e to prove the methods used.

Although this work investigates the structural performance of advanced composite materials in a particular high-performance application, fluid flow over this component is not investigated (through the use of channels etc.). The stiffness and vibrational characteristics are achieved through the optimisation of the materials used in the construction of the boards.

1.3 Thesis Organisation

This thesis is structured in the following way;

Chapter 2 contains research into the component in the case study and the numerical methods used in this work. In order to demonstrate application of the design process to the component used and to show how numerical techniques could be applied, it must first be understood what the desired characteristics of the component are. Once this has been established, research has been done on the numerical techniques used in the proposed methodology.

Chapter 3 looks at design methodologies used in product development. This explains the methodologies used by SMEs as well as multinational companies. Typically SMEs use qualitative (descriptive) methods, whereas multinationals use quantitative (prescriptive) models. This is so that the existing methodology can be understood when producing components made from Structural Orthotropic Composite Materials in High-Performance Systems, such as the one considered in this work and how these can be improved by SMEs through transitioning from descriptive to prescriptive models.

Chapter 4 establishes the mechanical properties of the subsystems i.e. the material properties used by the proposed component, so that these can be fed into the FE models used subsequently. This enables the prescriptive tools in the proposed methodology to be used.

Chapter 5 establishes the desired characteristics of the component being developed so that the design can be optimised accordingly. In order to optimise anything, variables, objectives and constraints are required. The variables are the position, orientation and number of plies used in the composite materials as well as the core dimensions, the objective is to minimise mass, however the constraints were not known. As such, chapter 5 establishes the constraints that were used. These are the mechanical responses that are required from the components being produced, namely the bending and torsional stiffness' as well as the vibrational response.

Chapter 6 uses the material properties established in chapter 4 in FE models in the methodology proposed in chapter 3 to design a component having the desired characteristics found in chapter 5 to meet the needs of the customer reviewed in chapter 2.

Chapter 7 verifies the design methodology by testing the desired characteristics of a known arbitrarily laid-up component that has similar construction to the boards considered in this work. A component that has the same characteristics is then designed, manufactured and tested using the proposed methodology. This demonstrates that the design methodology can be successfully applied by SMEs to components such as the one in this work.

Chapter 8 contains the conclusions and the future work suggested as a result of this research.

1.4 Research Methodology

Details of the initial research undertaken is provided in section 1.1 (Background to Kitesurfing) and section 2 (Previous Research and Contextualisation). However, additional previous literature is referenced throughout the work to arrive at conclusions that help guide it.

1.5 Contributions to Knowledge

Through discussing the methodology proposed and then applying this to the case study, this work contributes to knowledge in many ways. It is acknowledged that the methodology that is proposed is currently being used by multinational companies, however this work demonstrates how this can be used by SMEs when using Structural Orthotropic Composite Materials in High-Performance Systems. It is only relatively recently that reductions in software and hardware have meant that these methods are viable for use by SMEs.

The benefits of integrating optimisation into the design process is demonstrated and quantified. How much more efficient optimisation techniques make the design process is shown. Real world examples are taken from a first tier Automotive manufacturer and these are compared with the time and cost that it is estimated that these examples would have taken if a more 'traditional' development process was used. In addition, how effective methodologies are that use optimisation techniques is shown through a design example carried out by practicing and future engineers studying at University of

Wales Trinity Saint David. This study compares their designs with the optimised design that was suggested by optimisation software to see how effective they are.

Material properties were obtained from material suppliers, however, through correlating the vibrational behaviour of FE models that use these material properties, it is demonstrated how these results can be inaccurate. These are then scaled in vibrational models, to obtain correlation to test, thus negating the need for lengthy and costly material testing. Material properties for the composite material used in this study are then verified through ultrasonic testing techniques.

The bending stiffness and torsional stiffness distributions, as well as the NVH characteristics of the kiteboards being used in the case study at a component level are established and methodologies for replicating these are discussed. Whilst these characteristics have been tested on similar products, in terms of kitesurfing, these characteristics are often qualitatively described, however this does not appear to have been quantified previously and no publications could be found.

Clear differences in the mechanical properties are seen between the kiteboards being tested in the case study. It is these differences in mechanical properties that tailor these boards towards riders of differing abilities. Through the testing of these boards these mechanical properties can be quantified and then related to the riders that these boards are aimed at.

The vibrational response of the kiteboards being used in 'the field' are mimicked through 'mock field' testing and a methodology for doing this is established. This

'mock' testing takes the loads and constraints that have been recorded in 'the field' in other published research and uses these to load the boards in a similar manor in a controlled, repeatable fashion. Differing boards can be compared to see their responses and boards from the same manufacturing batch are able to be compared to see how repeatable the manufacturing process of these board is.

Numerical techniques are used to replicate the mechanical properties of the boards that have been tested. A methodology is proposed that establishes the lay-up needed when producing products in the case-study, depending on whom the product is aimed at. It does not appear that the lay-ups needed for boards aimed at different riders has been stated before due to the fact that this is seen as commercially sensitive by the OEMs. In addition, the required lay-ups appear to have been achieved by iterative techniques through the use of testing, rather than the numerical techniques that are proposed in this work.

By building an FE model of the mock field test that has been proposed, it is shown how the failure index can be estimated in these products during use. The durability of these products can also be assessed and improved.

A sample that is representative of the product that has been designed in the case study is manufactured, tested and correlated to FE models. This establishes how effective the techniques are that are used in this work. As such conclusions can be drawn and recommendations for future work can be stated.

2 Previous Research and Contextualisation

2.1 Introduction

There is very little published data on the design and development of composite structures in high-performance applications. Structures such as kitesurfing boards and snowboards are manufactured with a target market in mind – a board will either be a ‘high-performance’/freestyle board aimed at an experienced rider, or an ‘all-purpose’/freeride board aimed at a novice rider, or somewhere in between. The distinction between sector terminology which is qualitative and often referred to as ‘feel’ and a quantified scientific rigorous link has yet to be made. In fact, flex is described by North (a major manufacturer of kitesurf equipment, now called Duotone), as ‘just as important for the performance of a kiteboard as its rocker or outline’ (13). What this research aims to address is what structural response is required from a board and how to design these attributes into the board using numerical techniques.

2.2 Previous Research in Relevant Applications

In recent years, Kitesurfing has seen a significant increase in global participation. It is reported in the PhD by Valsecchi, to be one of the fastest growing sports, partly due to the fact that it is relatively easy to learn and the equipment is portable (14). To date there has been limited research into the sport and what does appear in literature tends to look at injuries related to the sport, or the kite design. Having said that, Valsecchi (14) has a good summary of the historical development of the sport and he looked into whole body vibration in kitesurfing. A single axis accelerometer was placed on the kitesurf board such that vibrations during use in the field could be studied and related to the magnitude of the waves, the wind strength and the kiter position and velocity.

Vibrations were shown to be dependent on the riding speed, whilst the other factors were shown to have little influence. This drove the decision to try to remove this variable from the field testing by introducing 'mock field' testing using a vibration rig as discussed in section 5.4. The stiffness of the board however was not considered, despite the fact that in a subjective survey carried out (although only with 21 respondents), approximately $\frac{1}{4}$ of people stated that the board characteristics were an important influencing factor on the vibrations felt. Interestingly, this work also looked at the influence of the footpads on the vibrations felt by the rider. Valsecchi studied the influence of the data acquisition enclosure that was rigidly bolted onto the board (see Figure 2-1) during testing between the feet of the rider and concluded that neither of these factors influenced the vibrations being measured. This seems to contradict the findings by Wolfspurger and Rhyner (15) who found that binding plates on snowboards that join the bindings on snowboards increased both bending stiffness and damping of the board.

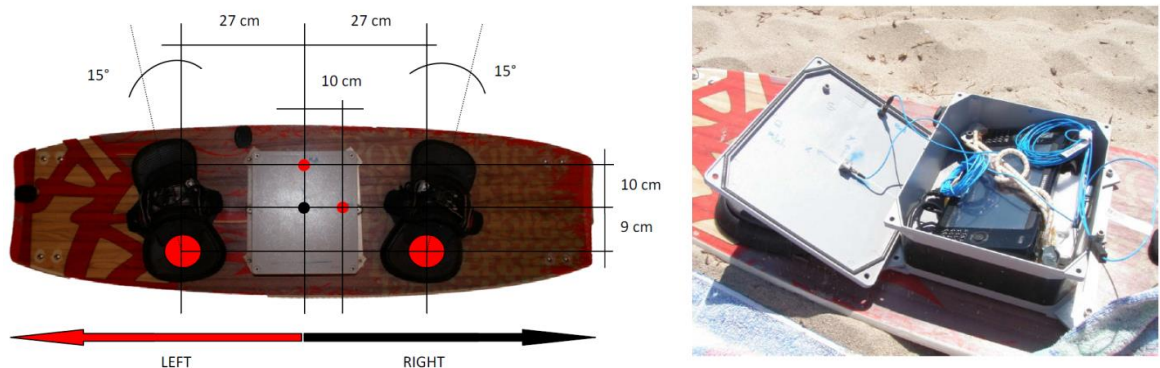


Figure 2-1: Data Acquisition Box Fitted by Valsecchi (14).

The study by Valsecchi concluded that kitesurfing could lead to degenerative back conditions due to the vibrations the rider is exposed to. Further studies by Lundgren et al (16) reported into injuries and stresses seen on the body during participation. This

study compares the injuries seen to those in wakeboarding due to the Anterior Cruciate Ligament and shoulder injuries sustained as a result of unsuccessful tricks and jumps.

There have been studies into the testing procedure when researching the mechanical properties of structures similar to these. Rainer et al stated that this is particularly important for the ski-ing industry so that the continuity of manufacturing can be compared to ensure that a pair of skis have identical stiffness' (17).

The sport of kitesurfing seems to have more published data on the design of the Kite. Data acquisition has been used on kites before by studies by Lauren et al (18), possibly due to their potential use as a renewable powersource in applications such as those used by skysails as seen in Figure 2-2 or in energy generation.



Figure 2-2: Using a Kite as a Renewable Energy Source (19).

Stresses seen in kites during use have been investigated by Verheul et al (20) and both the rip-stop polyester used in the canopies due to its better stress/strain and moisture absorption properties over Nylon and the heavy polyester cloth (tradename Dacron) used in the reinforcements and the inflatable struts have been evaluated. There have

also been studies by people such as Sala looking into the fluid flow around kites (21). Sala was then able to give direction in the design and manufacture of these kites.

In terms of the board design, there appears to be very limited published data. The study by Valsecchi (14) referred to subjective design guidelines in terms of the geometric shape of the board such as;

“The sharper the rails the more aggressively the board will bite into the water.”
“Square tails aid jumps as the board have more pop.”
“Rounded tails has a surfboard feel.”
“Grooved tails grip better.”
“A high rocker feels loose.”
“A low rocker is more stable.”

However, the structural performance of the board has not been investigated. North are reported to achieve the ideal balance between stiffness and flexibility by constructing their boards using carbon fibre reinforcement, as reported in Reinforced Plastics Magazine, however details of this and what is desirable remain unpublished (12).

At this stage, it makes sense to look at published work on similar structures used in other sports – for example Ski-ing, Snowboarding and Wakeboarding.

Whilst looking at ski-ing Glenne et al and Foss et al document the connection between ski properties and ski performance (22) (23). Foss et al reported that torsion characteristics play a predominate role in rider control in terms of how the rider deals with vibrations (23) – despite this, the manufacturing of skis and snowboards is still reported to involve considerable trial and error. Having said that, vibrations were shown to be a much lower hazard on soft snow, which provides smaller excitations and large damping – perhaps similar to the vibrations sustained on a kitesurfboard in

water? This study also concluded that laboratory testing of mechanical behaviour of skis was different to that of a ski 'in the field', hence the need to compare this kind of structure both in a laboratory and by emulating it, in its working environment, through 'mock' field testing. In addition, when looking at damping, this study concluded that damping saw a significant increase once bindings were introduced to the ski – which again seems to contradict the conclusions drawn by Valsecchi (14). In terms of damping, testing showed that fibreglass skis typically had higher material damping over metal/fibreglass skis, yet the study then goes on to contradict this by saying that metal/fibreglass skis were reported to be quieter and more stable at high speed in the field. Finally, this study concluded that the dynamic property most responsible for adverse ski behaviour at high speeds was the torsion mode – highlighting how important this property is in this kind of structure. Bending modes were also shown to affect performance – but to a lesser degree.

In the world of snowboarding, bending, torsion and natural frequencies of a snowboard have been discussed by Biancolini et al (24) along with their influence on stability. This study uses the international standard ISO5902 which is aimed at testing skis, but is applied to snowboards as no similar standard existed for snowboards. This standard, measures the modal response of a ski by clamping it in its central section, as shown in Figure 2-3.

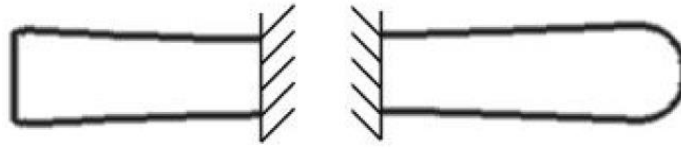


Figure 2-3: Clamping of a Ski to Measure its Natural Frequency.

This type of approach is supported by Valsecchi (14) who has shown that the central section has limited influence on the vibrational characteristic of a kitesurf board, however Biancolini et al (24), state that the central stiffness influences the stability and the manoeuvrability of the board so by using this approach this characteristic of the board would not be recorded, suggesting that alternative techniques should be used. The results of this study indicate that the first mode in the forebody (shown on the right of the clamping structure in Figure 2-3) was typically between 8 and 10 Hz, whilst the first mode in the afterbody (shown on the left of the clamping structure in Figure 2-3) was between 13 and 14 Hz.

Models have been developed to calculate the mechanical characteristics such as bending and torsion of snowboards by Clifton et al and Brennan (25) (26). Clifton et al (25) use these models to predict the characteristics of a snowboard based on the construction using composite materials, whilst Brennan (26) uses these models to try to predict how the structural performance of the board will influence the performance of a rider in terms of completing a set course in order to try to get away from the reliance on qualitative reviews from riders. Despite this, subjective statements such as “A soft flex is good for beginners” are still widely used and published in articles such as that of evo.com (27). Investigations into these subjective performance parameters of snowboards have been carried out by Clifton (28), which showed that together with bending stiffness, torsional stiffness and camber have been shown to strongly affect

the feel and performance between major riding styles of snowboarders. This study uses a questionnaire of riders of different styles to subjectively rate Stability, Manoeuvrability, Accuracy, Edge Grip, Speed, Feedback, Forgiveness, Board liveliness, Transition Smoothness, Stiffness and Durability and came to the following conclusions based on the results of this survey;

- Freeride snowboards should have a balanced blend of all the performance attributes listed.
- Manoeuvrability, forgiveness and correct stiffness levels are the factors most frequently identified as missing amongst the freeride snowboards on the market.
- Freestyle snowboards should be lively, with respondents mentioning manoeuvrability, forgiveness and stiffness whilst the other subjective parameters are almost non-existent in the responses.

As a result of this survey, various boards are tested and it is shown that;

- The bending stiffness distribution and mass of the main body are crucial in any optimal feel design.
- Torsional stiffness is important for both freeride and freestyle snowboards.
- Freeride boards possess the greatest levels of bending stiffness and torsional stiffness.
- The freestyle bending characteristic has considerably less stiffness fluctuation.

The conclusion of this work is that varying the bending stiffness, torsional stiffness and the camber appears to be the key approach to altering the feel and performance of a

snowboard across the major riding styles. In terms of stability and manoeuvrability, it was expected that low stiffness in the centre of the board would allow independent foot movement and hence improve rider control, yet it was suggested that as stiffness increased in the centre of the board, the response to the rider input is quicker, increasing the overall manoeuvrability, contradicting the statement made by Biancolini (24) that a low central stiffness was required for manoeuvrability. Meanwhile, increased heel and shovel stiffness of the board were shown to increase feedback. This paper also reviews previous work carried out by Buckingham (29) which showed positive correlation between on-slope subjective assessments and static testing results which contradicts the findings of similar studies done by Foss et al on skis (23). Biancolini (24) showed that there has been a trend of increasing stiffness towards the centre of the snowboard, whilst Borsellino (30) showed that the core plays a large role in the properties of the final composite and Grewal and Rossetter (31) went further to conclude that the core geometry will influence second and polar moment of area of the section and thus bending and torsion, thus the core geometry is of more influence than the material that it is made from. This drove the decision that core shape should be considered, by altering the thickness of this core, however, core material was limited to Paulownia wood only.

In terms of wakeboarding, there appears to be very limited published work, however there have been publications by Panciroli et al and Poodts (11) (32) which investigate failure mechanisms and the hydroelastic loads during harsh landings in wakeboarding and these then go on to provide practical guidelines which can be used during design.

In terms of testing the vibrational performance of snowboards, various work has been done such as that of Fuss et al, Sakata et al and Buffington et al (33) (34) (35). Fuss et al (33) showed that vibrations of snowboard are closely related to their performance and observed that the mass of accelerometers and their cables during modal analysis influence the vibrational behaviour of a snowboard, suggesting that non-contact testing methods such as Laser Doppler Vibrometry should be used (as has been done throughout this work). Meanwhile Sakata et al (34) concluded that the natural frequency of the snowboard is one of the most important properties, whilst Buffington et al (35) showed that 'Soft' snowboards have a lower natural frequency and larger damping ratios than similar boards designed for advanced riders which are characterized as 'stiff'.

Meanwhile in skis, an active approach to damping has been taken. Historically this is achieved through mechanical devices attached to the top of skis to dampen vibrations and more recently this has been through the use of piezoelements integrated into the structure of the board as discussed by Rothemann et al (36). Vibrations in skis (both amplitude and frequency) have been shown to influence ski gliding properties by Koptug et al (37) (38).

So, the performance of structural components such as skis and snowboards have been shown to be influenced by characteristics such as torsion, bending, damping and natural frequency.

The makeup of these components not only consists of a complex arrangement of many components, but the materials that make up these components are increasingly made

of advanced orthotropic materials such as carbon fibre reinforced plastics and glass fibre reinforced plastics, hence traditionally these components have been developed subjectively using an iterative approach to obtain the desired characteristics due to their complexity.

Skis, snowboards, fishing rods, golf clubs and racing bikes have all been reported to be improved through the use of composites by reinforced plastics magazine (39) and carbon fibre composites have been shown by Kutz to be the material of choice if cost is not of concern (40), in terms of stiffness to weight ratios. Additionally, carbon fibre has been shown to dampen vibrations in tennis rackets for example, which can lead to “tennis elbow”. According to Schroeder, Carbon fibre bikes are lighter, stiffer, and more shock absorbent (41) and design guidelines for mountain-bikes have been published by Case (42). Other sporting applications where advanced composite materials are used, such as in golf clubs, have been investigated by the PhD by Slater (43), which looks at the material in terms of its behaviour in high strain rate applications.

In terms of board sports, carbon fibre has been used in skateboards and is said to produce more ‘pop’ than an equivalent wooden board, whilst being 20% lighter as reported by Reinforced Plastics Magazine (44). In snowboarding, benefits of anisotropic materials have been investigated in snowboard edges by Kaufmann et al (45) and they are also reported to show improvements in strength and control in Reinforced Plastics magazine (46). The same publication reported that in surfing, carbon fibre boards have been produced by companies such as Aviso in order to reduce the weight of surfboards and Nano enhanced resins have been used to improve durability (47). Meanwhile, in wakeboarding, they report that boards are made with

composite materials through wet lay-up, filament winding or resin transfer moulding (48). However, the influence of the use of these materials on the structural characteristics and hence the performance of these boards is either unknown by those producing the boards or a closely guarded secret in order to obtain a competitive advantage.

2.3 Previous Research Using Optimisation

In multinationals, the study of composites is fairly well established and published, where lessons learnt should be reinterpreted in the context of the applications discussed and used by SMEs, rather than relying on iterative development cycles, based on subjective feedback in order to manufacture components such as these. For example, the influence of fibre orientations on the flexural natural frequencies has been investigated by Dutti and Shivanand (49), studies into the magnitude of modes and modal separation through optimisation of fibre orientation has been published by Kayikci and Sonmez (50) and free-free vibrational behaviour of composite plates has been researched by both Mishra and Sahu and Nayak (51) (52).

In terms of applying analytical techniques to the design of composite components, approaches to stacking sequence optimisation of composite laminates has been investigated by Levia et al (53) and optimisation techniques have been developed for frequency responses by Apalak et al (54). Specific optimisation software such as Altair Optistruct and Vrand Genesis are now available and are regularly used in the Automotive and Aerospace industries. Studies have shown that significant weight savings can be made to Automotive body structures by Cramer and Taggart (55),

(although the same study summarised that the use of materials has been limited by lack of experience and knowledge in how to design with advanced composites). Genesis and Optistruct have been shown to provide significant savings in mass and improvements in structural performance of structures in Formula 1 cars and Automotive components by Gambling and Gardner, as well as, Eleftheriadis and Altair (56) (57) (58) produced from composite materials. In addition, the aircraft industry now uses other techniques such as Topology optimisation as an integral part of the design process as discussed in the work by Saleem et al (59).

Previous studies by the author at the World congress on Structural and Multidisciplinary Optimisation, the conference on Modern Information Technology in the Innovation Process and the International Journal of Engineering Simulation (1) (2) (3) have shown how using numerical techniques can improve designs. It has been shown that in industries where the use of Computer Aided Engineering (CAE) is common place, bringing the use of this forward in the design process (in front of Computer Aided Design, CAD) as shown in Figure 2-4, improves productivity. Savings of about 75%, both in terms of costs and times have been demonstrated.

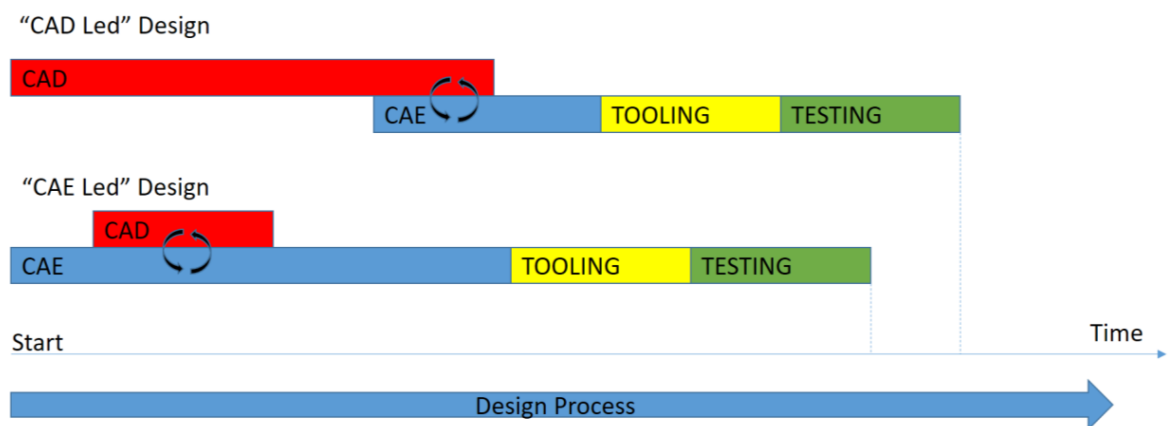


Figure 2-4: CAE v CAD Led Design (1).

Historically, the above tools have been relatively expensive to utilise due to high hardware and software costs, as well as the expertise needed, and as such, have been used only by industries that can absorb these costs. In industries such as the ones discussed in this work, where the use of CAE is apparently not used, it is expected that significant cost and time savings will be seen if this were to be incorporated into the design process. By introducing CAE into the design process both the proto-type manufacture stages and the proto-type testing phases can be reduced. In addition to this, not only will the designs get to market quicker and cheaper, it is expected that significant performance benefits will also result.

2.4 Conclusions taken from Previous Research

A summary of the findings related to the structural characteristics of relevant applications researched and discussed can be seen in Table 2-1;

Observation	Characteristic	Field	Ref.
The influence of bindings has little impact on the frequency response of the board.	Frequency	Kitesurfing	(14)
Vibrations of a board are closely related to their performance	Frequency	Snowboarding	(33) (34)
'Soft' boards have a lower natural frequencies and larger damping	Frequency	Snowboarding	(35)
Damping of the ski is significantly influenced by bindings	Damping	Ski-ing	(23)
Carbon fibre dampens vibrations	Damping	Tennis	(40)
Torsional characteristics play a predominate role in dealing with vibrations, and to a lesser extent, Bending modes	Torsional stiffness	Ski-ing	(23)
Torsional stiffness is important for freestyle and freeride boards	Torsional stiffness	Snowboarding	(28)
Freeride boards possess the greatest levels of bending stiffness and torsional stiffness	Torsional stiffness	Snowboarding	(28)
The stiffness of the centre section of the board has little impact on the frequency response of the board.	Centre stiffness	Kitesurfing	(14)
Manoeuvrability increases with centre stiffness	Centre stiffness	Snowboarding	(28)
The central stiffness of the board influences manoeuvrability and stability	Centre stiffness	Snowboarding	(24)
There has been a trend in increasing the centre stiffness	Centre stiffness	Snowboarding	(24)
'Give' is necessary for a novice	Global stiffness	Snowboarding	(27)
Bending stiffness distribution is crucial for feel	Global stiffness	Snowboarding	(28)
Freestyle bending stiffness characteristics have less fluctuation	Global stiffness	Snowboarding	(28)
Increased stiffness at the heel and shovel increase feedback	Global stiffness	Snowboarding	(28)
Freeride boards should have a balanced blend of performance attributes	Global behaviour	Snowboarding	(28)
Freestyle boards should be 'lively', manoeuvrable, forgiving and stiff	Global behaviour	Snowboarding	(28)
The core material influences the final properties of the structure	Global behaviour	Snowboarding	(30)
The core geometry influences the final properties of the structure	Global behaviour	Snowboarding	(31)
Carbon fibre is stiffer and more shock absorbent	Global behaviour	Cycling	(41)
Carbon fibre boards produce more 'pop'	Global behaviour	Skateboarding	(44)
Carbon fibre boards have more strength and control	Global behaviour	Snowboarding	(46)

Table 2-1: Summary of Previous Research on Board Mechanical Properties.

As can be seen in Table 2-1, much of the published information is descriptive, subjective and contradictory.

This work proposes to take the design methodologies used the Automotive and the Aerospace industries, combine these with lessons learnt in terms composites research and commercially available software and apply these to the novel application, of a kitesurf board. Once the desired characteristics of such boards are scientifically quantified, these can then be designed analytically in order to create the characteristics that are appropriate for the rider that the board is aimed at, without the need to resort to many iterations of development.

3 Design Methodology

The design process can basically be described by either Descriptive Models or Prescriptive Models. As shown by Cross (60), Descriptive Models, as the name suggests, simply describe the sequences of activities that typically occur in designing, whilst Prescriptive Models attempt to prescribe a better or more appropriate pattern of activities.

3.1 Descriptive Models

Descriptive models are solution focused. That is to say that a solution is proposed early in the design process based on 'engineering experience' and this proposed solution is then tested and iterated until a final solution is reached. This process is described in French's model of the design process as shown in Figure 3-1. Previous work done by the author has referred to this as CAD led designs.

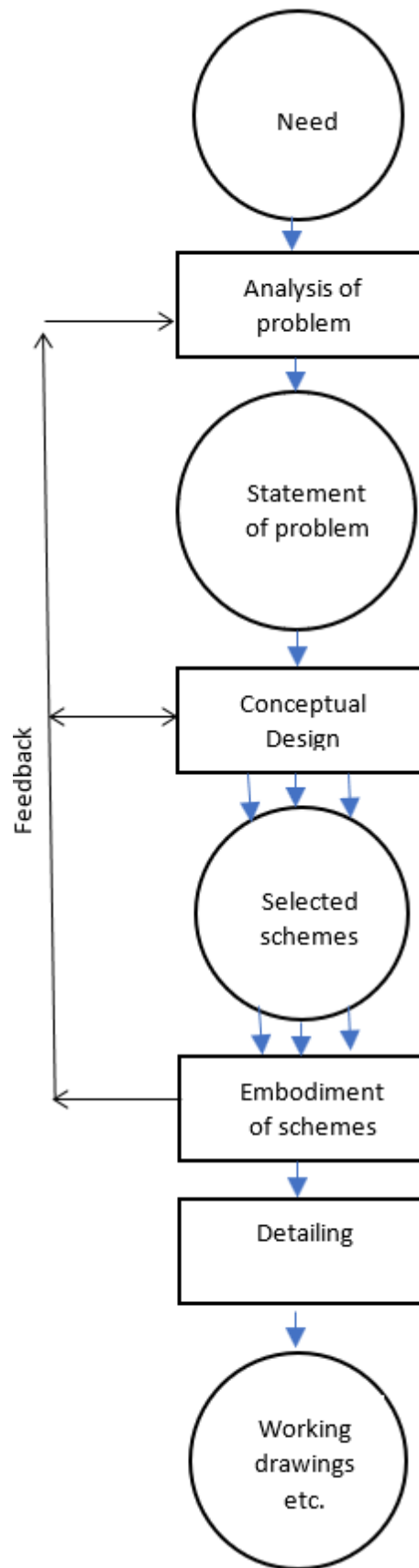


Figure 3-1: French's Model of the Design Process (60).

This model splits activities down into the analysis of the problem, conceptual design, embodiment of schemes and detailing. Within the diagram, circles represent outputs and rectangles represent activities. This design process is also referred to as the Iterative Design Procedure where an initial design is proposed and then refined iteratively until a solution is found. This method assumes that the design becomes 'better' with each iteration until an optimum is found. An example of the iterative design procedure can be seen in Figure 3-2.

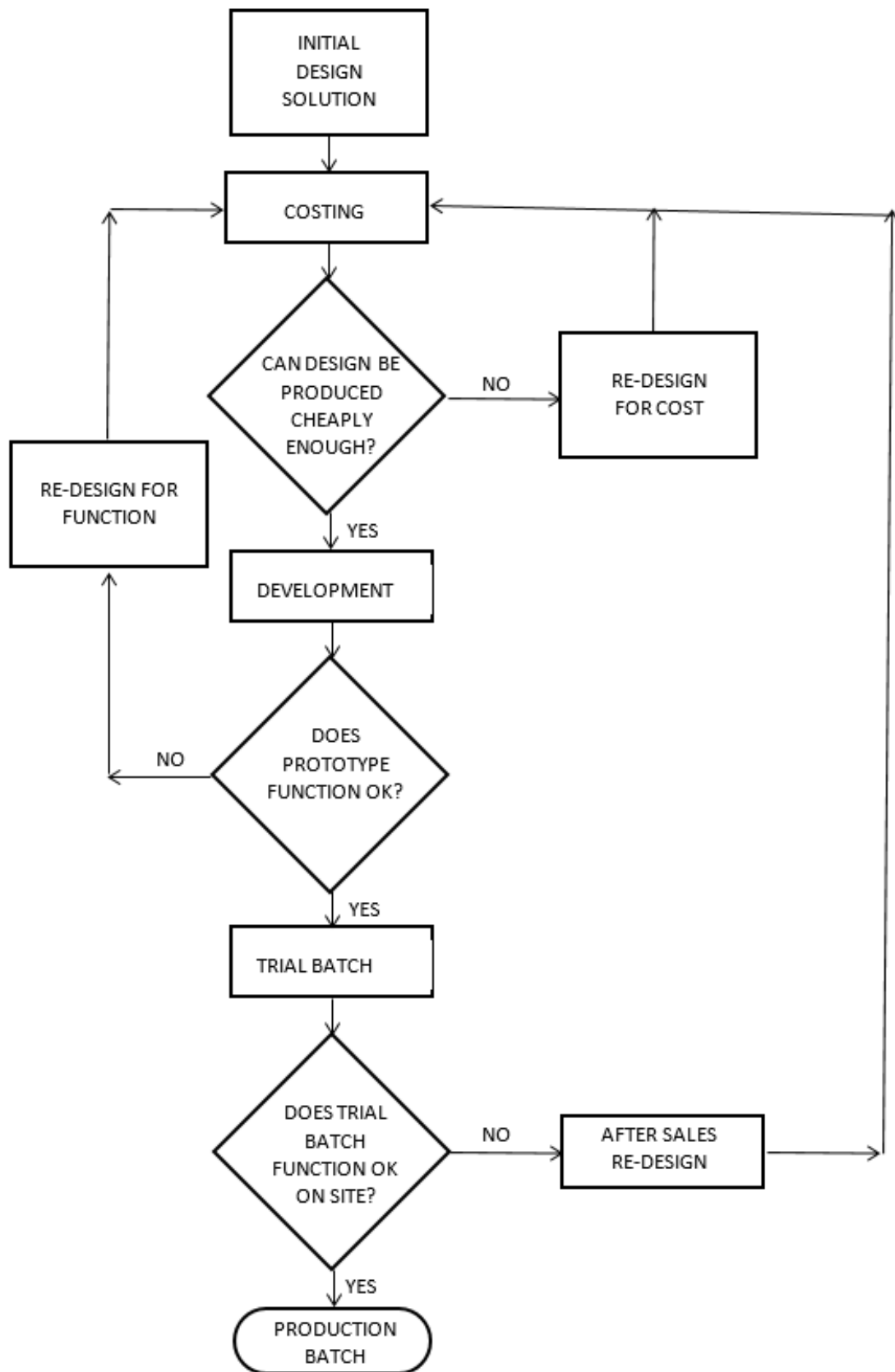


Figure 3-2: Iterative Design Procedure (61).

Many engineering problems, including the response of Structural carbon fibre components, particularly those discussed in this work, still appear to be using Descriptive or Iterative design processes. The fundamental flaw with such a design, is not only in the time taken to do iterative design, but that a 'local minimum' may be found as shown in Figure 3-3.

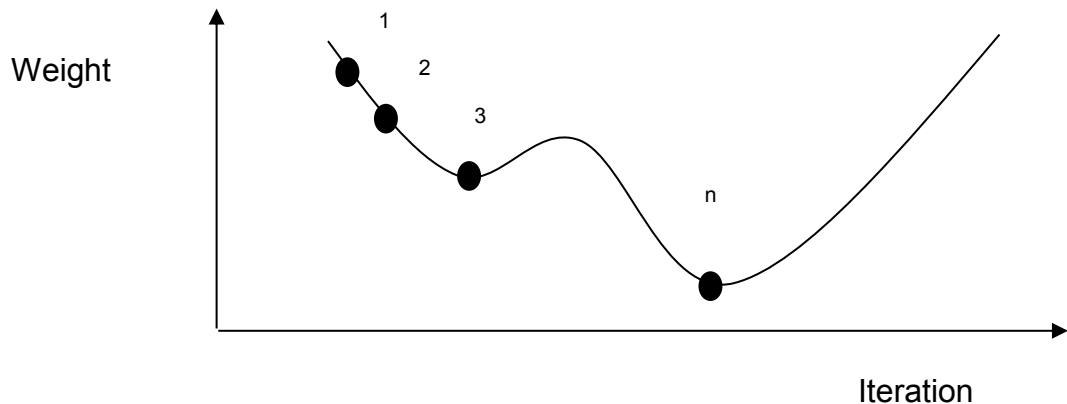


Figure 3-3: Local Minimums in Design.

If a board, for example, were being developed with certain characteristics (bending, torsion, mode shapes etc) and an initial design were proposed (iteration 1), by the time the design process has achieved iteration 3, this design would be assumed to be 'optimum' since further iterations will result in an increase in weight of the structure. However, as can be seen in Figure 3-3, this has simply found a local minimum. A better design is actually possible after n iterations. Techniques should be used to ensure that the true optimum design is actually found.

3.2 Prescriptive Models

Prescriptive Models try to force designers to adopt improved ways of working, to generate more efficient solutions. Many of these emphasise the need for analytical work to be carried out upfront in the design process. An example of a Prescriptive

model is shown in Figure 3-4. Previous work by the author has referred to this as CAE led designs.

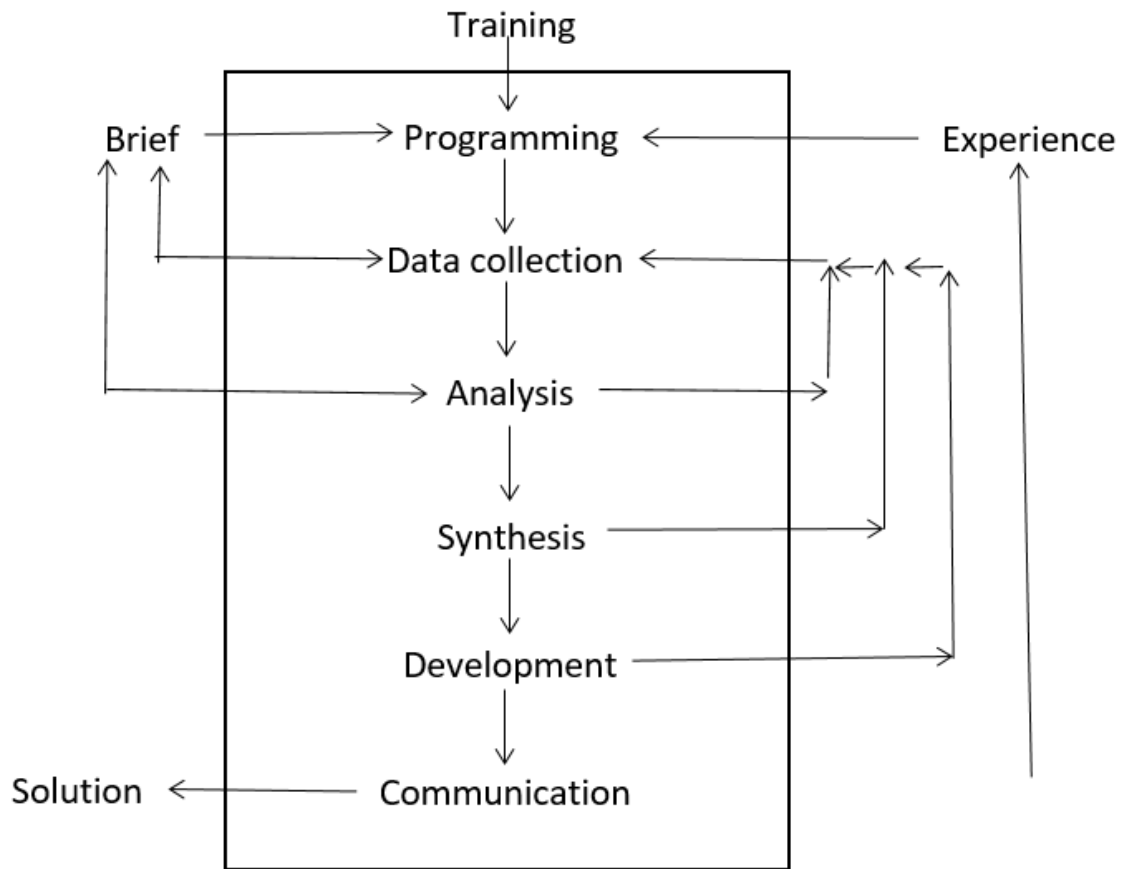


Figure 3-4: Archer's Model of the Design Process (60).

Archer's model was developed in 1984 and he identified six types of activity;

1. Programming – Establishing crucial issues. For components such as those discussed in this paper, this may include deciding critical bending or torsion values or natural frequencies.
2. Data Collection – collect, classify and store data. This may include obtaining material properties.

3. Analysis – identify subproblems; prepare performance (or design) specifications; Reappraise proposed programme and estimate. This phase would involve correlation of CAE models for sub-system responses.
4. Synthesis – Prepare outline design proposals. This should involve optimisation techniques to ensure that a ‘local minimum’ is not found.
5. Development –Develop prototype design(s). By using steps 1-4, this phase should be significantly reduced.
6. Communication – preparing manufacturing documentation (e.g. ply books etc.).

The fundamental difference between Descriptive Models and Prescriptive models is that Prescriptive models enable the use of the latest analytical techniques such as optimisation in order to improve the design process, this gives Prescriptive models a fundamental advantage, after all, as stated by Fernandes and Margarida (62), “Design tools, in order to truly support and accomplish the needs of the design process, must embrace change”, this means using the latest tools available to them. Fernandes and Margarida (62) then go further to state that the advent of computers have the opportunity to change the design process in a fundamental way not seen since the invention of paper, if the latest tools such as Generative design (a type of optimisation discussed in section 3.3.2.1) are used. If not, CAD tools are simply being used as a more efficient version of the traditional paper-based approach.

3.3 Optimisation in the Design Process

In order to understand the fundamental principal of optimisation techniques, the example from Vanderplaats is shown in Figure 3-5 and can be considered;

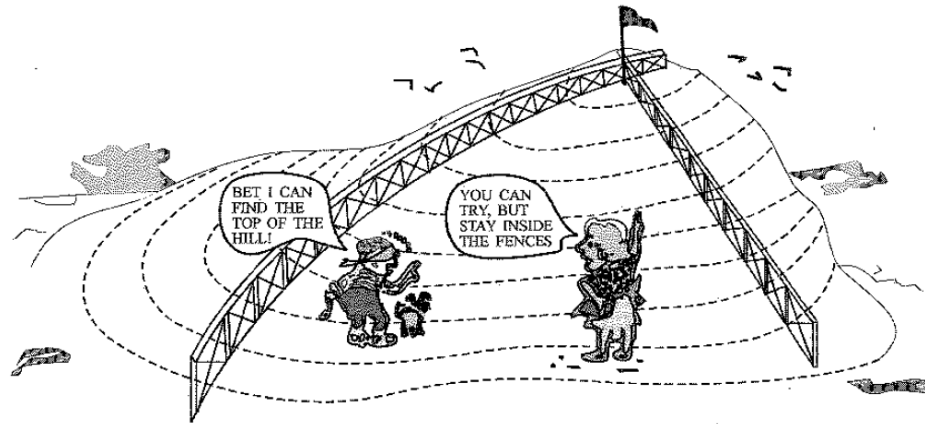


Figure 3-5: The Fundamental Principle of Optimisation (63).

In order to reach the top of the hill (the optimum point) the blindfolded boy can simply 'feel' his surroundings and move towards the top of the hill based on his findings of the surrounding gradient until he reaches his destination. Such 'gradient based' optimisation techniques, are still liable to converging on local minimum, and as such, many metaheuristic optimisation techniques have been developed to overcome this (64), such as;

- Genetic Algorithm
- Particle Swarm Optimisation
- Ant Colony Optimisation

Most metaheuristic algorithms are inspired by natural phenomena, for example mimicking animals' movement to find food sources, using a 'scatter gun' approach to improve the chances of finding the global minimum, rather than a local minimum.

Despite optimisation techniques being well understood, readily available and relatively cheap to implement, they are still remarkably underutilised in industry. In 2004 Vanderplaats stated that "In engineering, while development [of optimisation techniques] has been underway for over forty years, applications have lagged far

behind. The time has come for that to change” (65) and that “optimization technology has matured to the point where it can and should be used routinely for engineering design” (65). In fact, as far back as 1978 Vanderplaats “predicted that optimization would be commonplace by 1985” (66), but as he said in 2004, “Clearly, that did not happen and predictions are no longer made” (66).

Reducing waste has become a key element in lean thinking that has become prevalent in much of modern manufacturing. Typically, this focuses on reducing waste in the manufacturing process itself. However, product development is often overlooked despite its significant contribution to the cost of the product. In the design process, waste may be generated in two ways. Firstly, the process itself may be inefficient, with many iterative steps undertaken in an attempt to find the optimum design solution. Effectively, this makes the process inefficient and wasteful in terms of time and effort. Secondly, the output of the design process may well not be the optimum solution, leading to over-engineering and further waste in terms of material, manufacturing operations and so on. The remainder of this chapter illustrates how optimisation techniques can make the design process both more efficient and more effective and discusses the tools that can be used to do this.

The design process may be inefficient in several ways in terms of time, effort and costs, which will have significant consequences downstream in the product lifecycle. In terms of the costs incurred, the problem definition and the conceptual development stages are small, however, there are significant costs committed due to decisions made at this stage. This can be seen in Figure 3-6, reproduced from the work done by Agiyurdas (67).

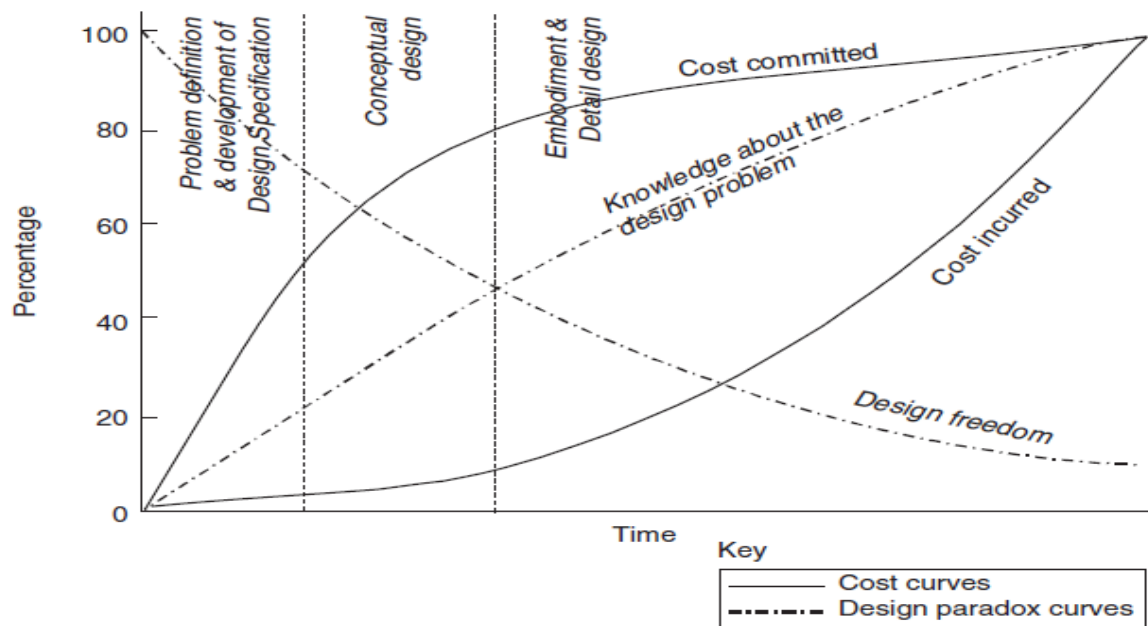


Figure 3-6: Superimposed Cost & Design Paradox Curves (67).

This is highlighted further by studies such as the Nafems Autosim project funded by the European Commission between September 2005 and August 2008. During this study, the “rule of ten”, was discussed, as follows;

“the cost of fixing a problem that designers should have corrected in the planning and concept phase increases 10 times if the company discovers it in the testing phase, 100 times if it finds the problem in the production phase, and 1000 times if the customer discovers it.” (68)

This statement, along with Figure 3-6, demonstrates the relative magnitudes of the incurred costs if the initial design is not correct. To ensure initial designs are right first time, simulation is now routinely used in the early stages of development. During a webinar to discuss the Autosim project and its findings, Gino Duffett from Herbertus, discussed how through the use of simulation upfront in the product lifecycle Ford had made significant savings in development of the Fiesta (69).

Not only is upfront development necessary to ensure that the correct product is produced, significantly reducing downstream development time, effort and costs, it is

also necessary to ensure that quality is assured throughout the project. This fact has been attributed to the significant advances in the Japanese Automotive industry as described by U. Sellgren in 1995 who stated that “It is now quite clear that the Japanese automakers, very early, had a clear knowledge in the fact that the basic quality of a product mainly is determined in the early stages of the design process” (70).

The importance of getting the correct product to market through front loading development is therefore clear in terms of the cost and quality, thus reducing wasteful activities downstream. However, this must also be done in a timely manner. This is highlighted in various pieces of research, such as that of U. Sellgren, who discussed the work done by McKinsey & Co, stating that “A high-tech product that, even on budget, reaches the market six months late, will earn 33% less profit over a period of five years. The same product is finished on time, but 50% over budget, will reduce the company’s profit by only 4%” (70). Hence to reduce waste in terms of cost and time, both within the development of a product and downstream to ensure a successful product, this development must happen quickly and efficiently upfront in the design process.

The second main area where waste may be generated in the design process, is if the designs that are generated are not optimum. This leads to over-engineering, excessive use of materials, waste in terms of manufacturing operations, and once the product is in service, additional usage of resources to maintain or use the product or through fuel usage for example. To ensure that optimum designs are produced, designs should be CAE led (Prescriptive), rather than simply using CAE as a verification tool (Descriptive). In particular, this can be achieved by using optimisation which Vanderplaats has shown “can be used with minimal added cost or effort to significantly reduce energy

consumption” (71). For example, in terms of the Automotive industry in 2006, Vanderplaats showed that “a ten percent mass reduction will increase fuel economy by about seven percent” (65), and he went on to say that given the economic climate at the time there were also significant benefits since a “one percent economy improvement [in fuel consumption] will save nearly three billion dollars per year in the U.S. at the pump” (65). This obviously also has an environmental impact that has a large and growing importance.

This work demonstrates how frontloading product development with CAE led design, ensures that designs downstream are produced efficiently. Further, to demonstrate the effectiveness of CAE led designs, a study is detailed to compare designs produced by a sample of students and engineers with that of a solution derived using optimisation techniques. This shows the variability in CAD led designs which rely on engineering judgement and experience, and how ineffective they are compared to the optimum solution.

However, Vanderplaats stated that “despite the widespread availability of this technology [optimisation techniques], it is seldom taught as a design tool by universities and remarkably underutilised by industry” (65), this is now starting to change. For example, in February 2009 Swansea Metropolitan University (now UWTSD) became the first University in the U.K. to include the practical application of optimisation tools in Genesis in one of its courses, thus ensuring future engineers have the knowledge to implement such tools.

There are however, examples where optimisation is used successfully. This work now summarises some of the FEA techniques that are available and then details the applications of these techniques in the optimisation of large, thin-walled, high-pressure, die cast Magnesium alloy components for the Automotive industry by a first and second tier supplier to Automotive manufacturers. Finally, how CAE led designs fit into the product lifecycle, successfully reducing development times and costs, are discussed and demonstrated using this example. It should be noted that simply using gradient based optimisation techniques does not guarantee that the global minimum is found. As seen in Figure 3-3, a local minimum may still be found. As such, when using optimisation techniques, it should be ensured that a local minimum is not found. One way of doing this is to use a ‘scatter gun’ approach, that is, to start in many places and then optimise all the designs in parallel and the design with the lowest objective is the ‘best’ design. This does not guarantee that the global minimum is found, however this increases the chances of this happening. Obviously, this has a computational consequence as many models are being run in parallel.

3.3.1 FEA Optimisation Processes

In order that the examples given are clearly understood, the optimisation techniques that were used will now be discussed.

3.3.1.1 Topology

Topology is used to define where the most efficient structural loadpaths are within a structure, and as stated by Leiva is a “specialization of material optimization, forcing zero/one answers” (72). A package volume is needed within which the part must fit and then loadcases are applied to the package volume, to determine these loadpaths. This is achieved by varying the modulus of elasticity and the density of the material in the

designable region. As described by Leiva, “In Topology optimization, the design variables correspond to the element volume fractions. The volume fraction designs simultaneously the material properties E (modulus of elasticity) and ρ (density) with the purpose of getting a 0-1 answer to identify the key elements to keep and the rest to discard” (73). For example, consider the following scenario where the variable is the package space, the objective is to minimise strain energy and the constraint is to only use 30% of the material by mass. An initial mass fraction may be set to 30%, and as such the modulus of elasticity and density of every element in the designable region is set to 30% of the material that they are made from. This package space that is made from this imaginary material is then loaded and the strain energy for this iteration is measured in every element and then summed for the total strain energy in the structure. In order to minimise the amount of strain energy in the total structure for this load-case, the modulus of elasticity and density are increased in any highly strained elements for the next iteration. However, in order to only use 30% of the material overall, if the density of some elements have been increased in highly strained areas, then the density of elements in areas that are not strained must be reduced. By doing this, the software migrates the material towards highly strained areas and thus defines the major load-paths within in structure.

Topology can successfully be used to design the a-surface of a part (i.e. what the main shape of the part needs to be), as well as being used to determine how an existing part needs to be reinforced to be as efficient as possible, for example in a casting to define where ribbing should be.

The results of a Topology optimisation are somewhat 'organic' in appearance and these need to be interpreted into a manufacturable engineering design. In many ways this is a limitation of Topology. As described in 2005 by Schramm "the interpretation of the results is still a mostly manual process" (74). Although contemporary manufacturing techniques such as 3D printing are changing this.

Topology is generally used as the first step in optimising a design. Vanderplaats describes how it identifies "load paths and provides a very good starting point for shape and sizing optimization." However, he then goes on to say that "topology optimization seldom produces a final part, even though manufacturing constraints are used. This is because topology optimization normally does not include stress and other constraints" (65). Topology optimisation has been shown by Niemann et al to be successfully applied in the optimisation of lattice composite aircraft fuselage structures (75).

3.3.1.2 Shape Optimisation

Once the major loadpath locations have been established, the cross-sectional size of those loadpaths can be determined by using shape optimisation. As described by Leiva, loads are applied to the structure and the physical dimensions of these loadpaths are altered to meet the set criteria by using "scale factors of perturbation vectors" (73) as the design variables.

3.3.1.3 Topometry

Once the position and size of the loadpaths of a structure have been established and the optimum shape has been designed, Topometry can then be used to determine how much material is needed throughout the structure to ensure that specific loadcases are met.

Size optimisation is well established where a physical property (for example the thickness of a group shell elements) is varied. In his work from 2005, Schramm describes it as follows. “Sizing optimization has traditionally been the entry point to the use of optimization technology in design. Especially in designs that use a lot of sheet metal this is a useful tool. It is well established and implemented” (74).

Topometry is an evolution of size optimisation. To quote Leiva, “Topometry optimization is a generalization of sizing optimization. Unlike size optimization, where all elements associated to a property data entry are designed with the same values, in Topometry optimization each element is designed independently” (73). This can have significant advantages. Size optimisation can work well for example, as stated above in a fabricated structure. However, in order to run a size optimisation, the analyst must group the elements together that will be optimised thus influencing and limiting the results. Leiva explains how Topometry “allows performing sizing optimization on each element individually” (73) thus resulting in a more efficient structure. Casting has the huge benefit that a varying wallstock can be applied throughout the structure, thus ensuring that extra material need only be added where it is required. Thus, Topometry results can be interpreted well into a cast design.

3.3.1.4 Composite Optimisation

Components that are manufactured from structural orthotropic composite materials offer significant challenges and opportunities in terms of optimisation. In particular, there are many variables that can be optimised during manufacture, including fibre

orientation, stacking sequence and number of plys when laying up a composite structure.

3.3.2 Enablers to Optimisation

3.3.2.1 Advances in Hardware and Software

Recent advances in Software mean that Topology optimisation is now being introduced into several CAD packages in order to enable designers to identify loadpaths in the early stages of design. This is now branded 'Generative' design and is available in products such as Autodesk (76). This is not without controversy. Bagassi (77) argues that Generative design includes both "Evolutionary Design and Topological Optimization" and states that these "tools are powerful from a structural engineering point of view, however they are not fully capturing the creative intent of the designer". Historically the process of optimisation has been expensive and takes a high level of technical skill and understanding. This software is now available very cheaply and thus should be being used routinely as a means improve designs. For example, Fernandes and Margarida (62) show how "integrating Generative Design as a new stage in the design process dramatically simplifies the handling of changes". At the same time hardware improvements mean that the machines needed to run this software are now more affordable than ever. This is demonstrated in Figure 3-7, which has been reproduced from the work by Komorowski (78).

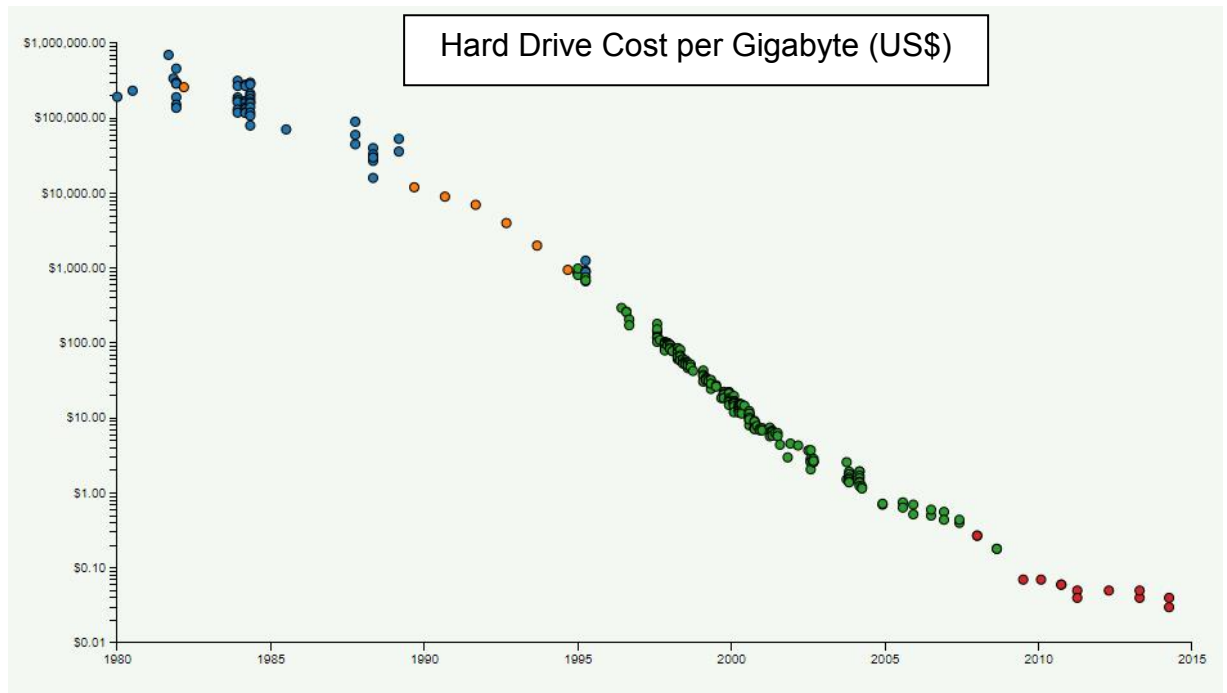


Figure 3-7: Storage Costs Over Time (78).

This shows that “over the last 30 years, space per unit cost has doubled roughly every 14 months” in terms of memory (78). SMEs can even access this hardware and software without the need to purchase it at all by using ‘Cloud’ based services as discussed by Leen et al (79). Ezell and Swanson (80) go so far to state that “Cloud computing – the provision of infinitely scalable computing resources as a service over the internet – is in the process of transforming virtually every facet of modern manufacturing”, such that “cloud computing democratizes access to [tools such a optimisation], and the use of these technologies by small manufacturers” acts to “level the playing field especially for small to medium-sized manufacturing enterprises (SMEs) that lack the financial wherewithal to purchase expensive IT equipment”. They also state that this of course has only been possible with the relatively recent access to robust, reliable, fast and widespread internet connectivity (80). The growth of this industry can be seen in Figure 3-8 which has been reproduced from Richman’s work which shows that global public purchases of Infrastructure as a Service (IaaS), Platform

as a Service (PaaS) and Software as a Service (SaaS) has gone from \$13 billion dollars in 2008 to \$87 billion dollars in 2015 and this was predicted to increase to \$236 billion dollars by 2020 (81). This is supported by Gartner, who reported in 2019 that this projection was correct and that this is predicted to grow to \$331bn in 2022 (82).

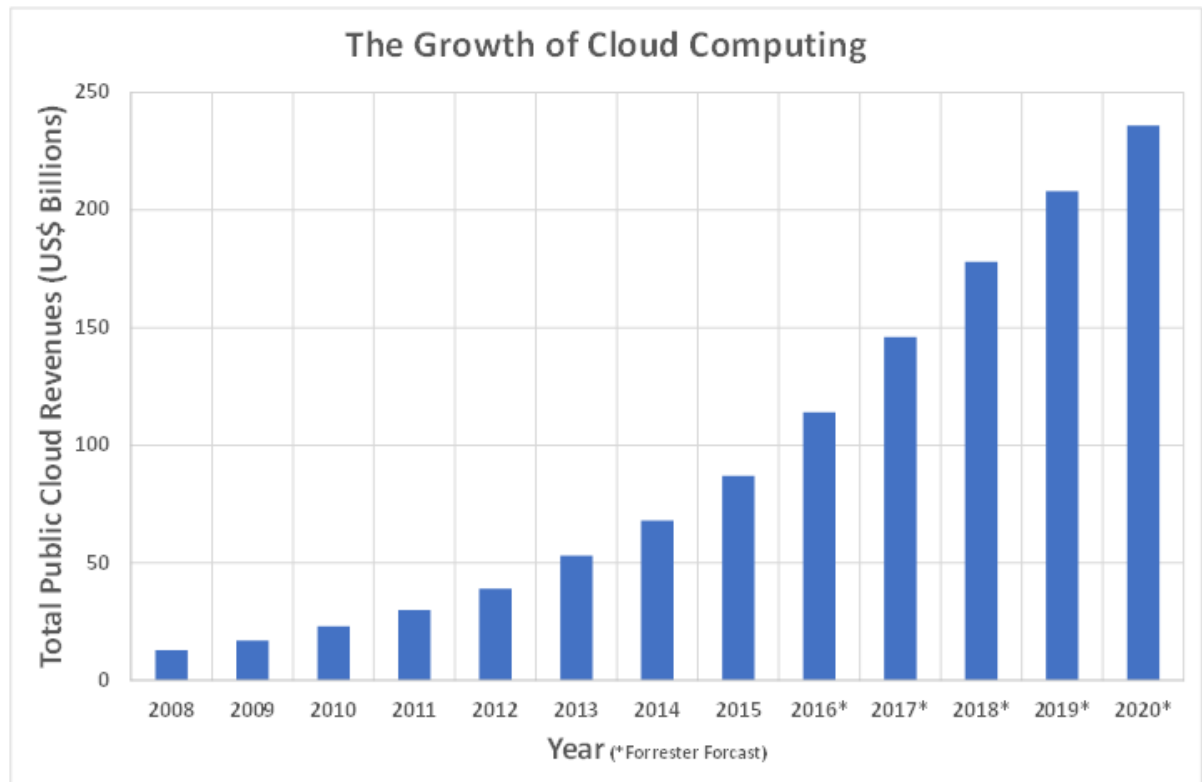


Figure 3-8: The Growth of Cloud Computing (81).

Leen et al (79) further demonstrate that ‘Entry-Level’ software and hardware can give similar results to ‘Professional’ systems. In this study, a component is reverse engineered and then a shape generation exercise is carried out and concludes that “what was once expensive (both in equipment and software costs) and [a] highly technical process, [is now] much more accessible to SME’s”. During this exercise the ‘professional’ scanner cost \$75,000 compared to the mobile phone alternative of \$350 whilst the professional software cost \$40,000 compared to the entry-level software that was \$560. In terms of the market of CAE, a study was undertaken by iDataLabs of

~310,000 companies and the market share of the software used was as shown in Figure 3-9 (which has been reproduced from this survey);

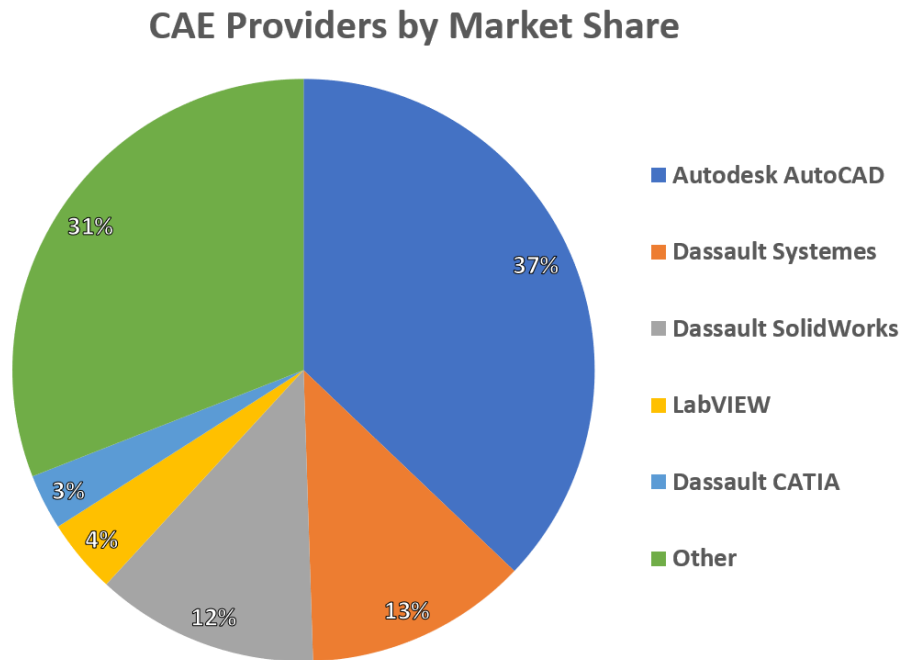


Figure 3-9: CAE Providers by Market Share (83).

This showed that the market leader with ~37% was Autodesk (it does not seem coincidental that Autodesk are integrating Generative Design). However, in terms of optimisation in the context of what is being discussed in this work, the market leader is Altair which is currently mainly used by the Automotive industry in the US by companies with more than 1000 employees (83). In 2017, Altair saw their shares increase in value by 41% on the first day of trading (84). Chief Executive James Scapa is reported to have said “We see a huge opportunity in simulation.....Math-based tools are driving everything now, so we’re in the right place at the right time with a strong model. This is only going to get a lot bigger”. In terms of the cost of software, according to Rosoff, this is now ~30% of what it was 20 years ago (85) across all software development, as seen in Figure 3-10.

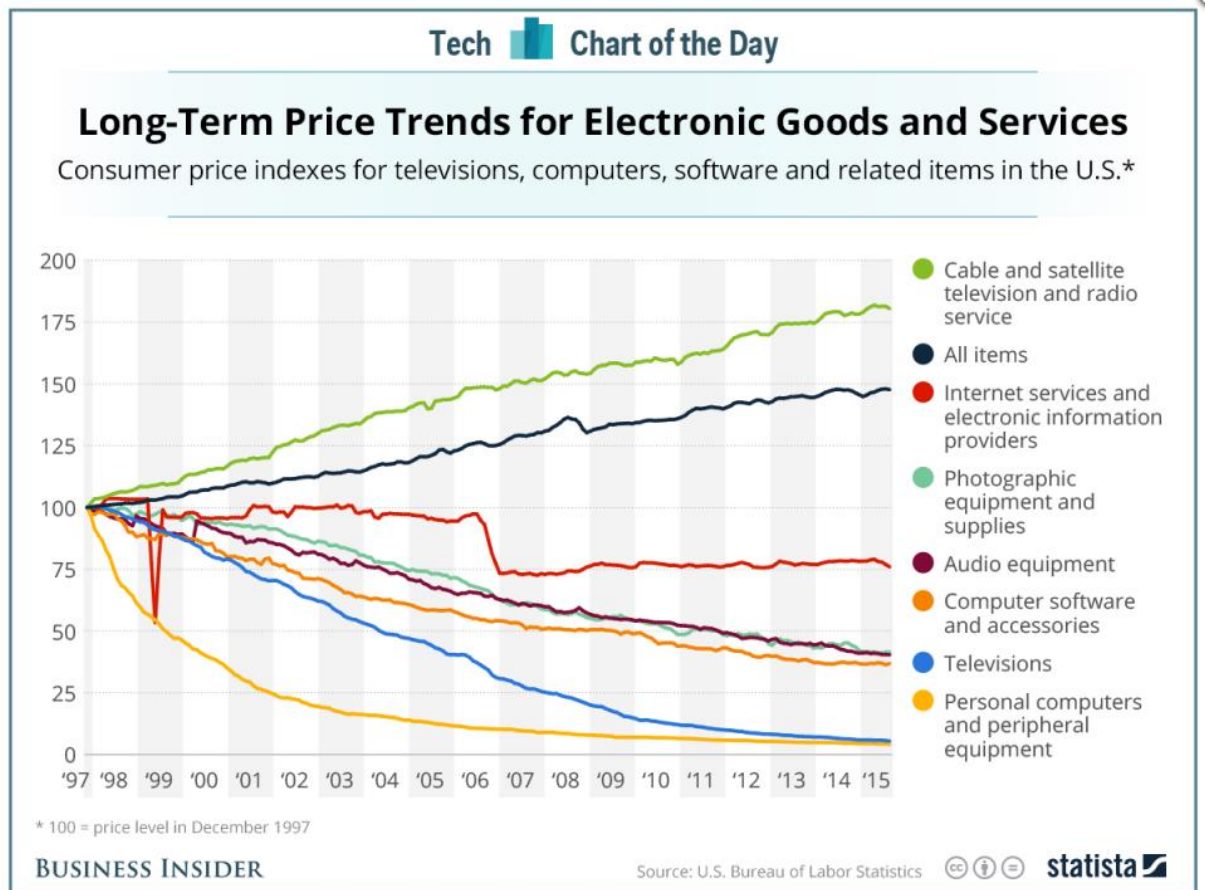


Figure 3-10: Price Trends for Electronic Goods & Services (85).

All of the above helps to demonstrate why SMEs should be applying optimisation to the design process.

3.3.2.2 Advances in Manufacturing Processes

Recent advances in additive manufacturing, or 3D printing, means that structures that are produced using tools such as Topology and Generative Design can now be manufactured easily and cheaply. Bagassi (77) discusses how “advanced components are [now] designed to be directly manufactured by an additive technique”. Airbus (86) demonstrated a very effective example of Generative Design in the development of the Airbus cabin partition designed in conjunction with Autodesk as seen in Figure 3-11.

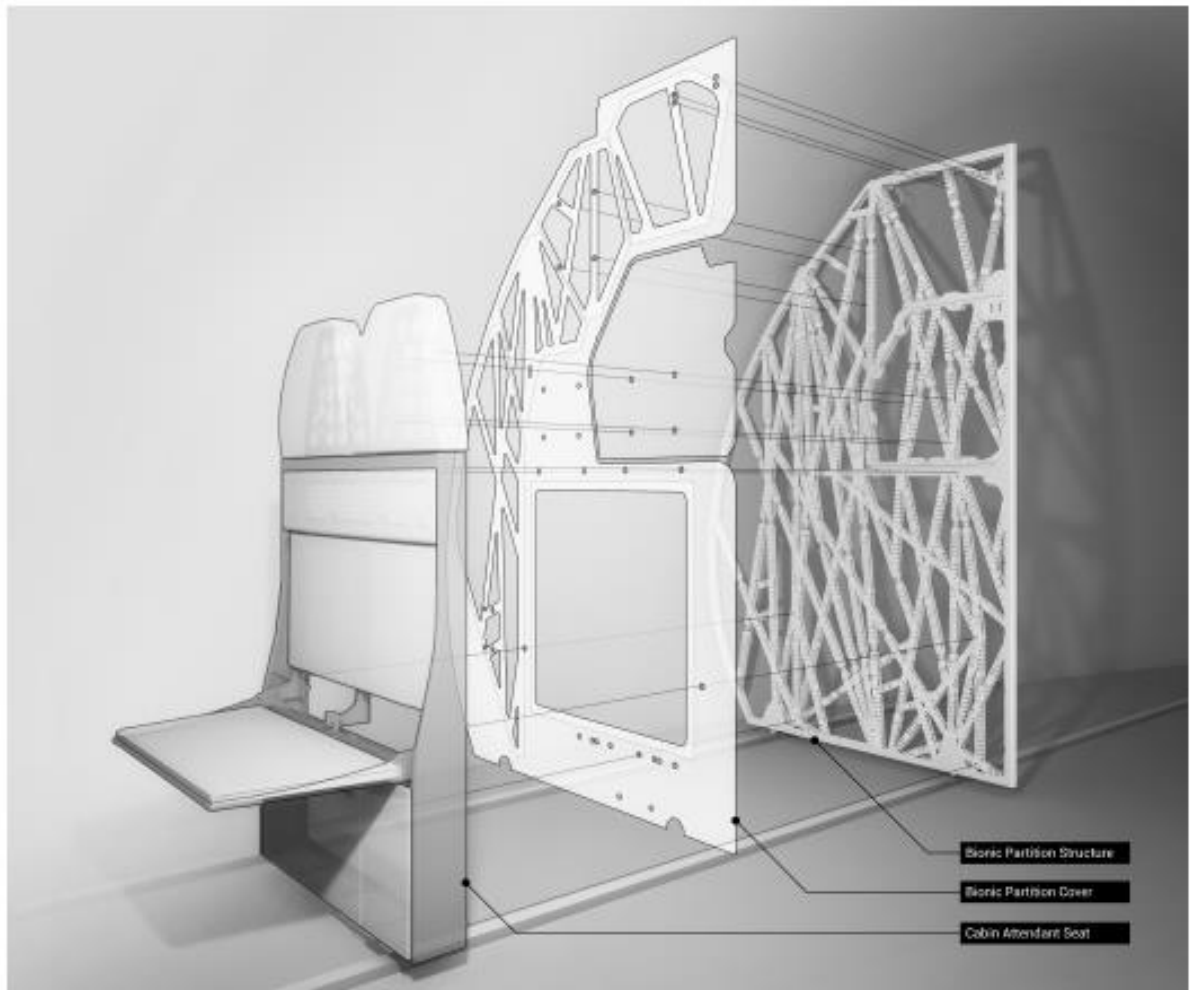


Figure 3-11: Novel Bionic Partition for Airbus A320 Cabin Interiors (77).

These structures that have been developed using Generative Design are designed to be manufactured using 3D printing resulting in a structure that is 30kg lighter than the current design. When applied to an A320 this will reportedly save 465,000 metric tons of CO₂ per year (86).

3.3.3 Examples of use by an Automotive Supplier

Structural optimisation in the Automotive industry is not a new idea. Vanderplaats quotes Henry Ford from 1923 who said “Saving even a few pounds of a vehicle’s weight means it could go faster and consume less fuel. Reducing weight involves reducing materials, which in turn, means reducing cost as well” (71). However, as

already stated, optimisation techniques are underutilised, although it is difficult to determine the real extent to which optimisation techniques are used. This is highlighted by Vanderplaats due to the fact that “most real commercial problems are proprietary and cannot be published” (65). Although, as discussed by Leiva, necessity has meant that the increase in use must grow as “Intensive competition in the market place today requires engineers to continually search for better and more economic designs” (73). Cases have been discussed, by Vanderplaats and large automotive suppliers, examples include;

1. A steering knuckle mass which was reduced by “five percent or more with no loss in strength” (66).
2. Optimisation was used to “increase the first frequency [mode] of a car body by ten percent for the cost of under one percent added mass” (66).
3. Delphi Automotive describe an Automotive dash cross beam in which they “reduce the mass by 33% and material cost by 39%” (71).

Previous work by the Author (1) (2) (3) has used the design process at Meridian Lightweight Technologies as an example of using CAE early in the Design process. Meridian design, engineer and manufacture Magnesium die cast components. CAE is used extensively throughout the development of their products and in these studies, this was highlighted as an example of good practice and specific examples were demonstrated using the optimisation tools that have already been described. The manner in which Meridian technologies successfully use CAE led designs both in conceptual designs and in development of these concepts, through to production once the business has been secured has been discussed.

Meridians components range from the structural part of a car dash board, through to car seats and grill reinforcements used at the front of the vehicle. These components are large, high pressure die castings (typically around 3mm wall thickness, the width of a car and 5kg in weight), which lend themselves to Topology, Topometry and Shape Optimisation. A customer may have defined the A-surface of a component and then require Meridian to develop that structure so that it meets the customers structural targets in Magnesium, in which case, Topology is used to define the ribbing required before Topometry is used to define the required wallstock of the structure.

Alternatively, a customer may come to Meridian with a package space and loading requirements for a structure. In this case, Topology is used initially to determine the A-surface shape of components. Once established, Topology can be used to determine the ribbing needed in these castings, shape optimisation is used to determine the final dimensions of the part and Topometry determines the required wall thickness of the castings.

These components are optimised to ensure no plastic deformation occurs in linear loadcases and there is no failure or plastic deformation in non-linear loadcases. In addition, Meridian's components are required to contribute to the vehicles in ways that include global stiffness and crashworthiness as well as achieving certain local stiffness requirements to provide stable platforms for the steering wheel assembly and airbags, whilst ensuring the designs are as light and cheap to produce as possible.

Meridians products are structurally quite complex and thus utilise many different types of optimisation tools where appropriate, as pointed out by Leiva when discussing Topology and Topometry "a complex structure can benefit from both [types of

optimisation], as some parts of it could only be designed by one type of optimization, whereas other parts could only be designed with the other type” (72).

To demonstrate how prescriptive design methodologies have been successfully applied by Meridian, three examples will now be considered. The first two show how prescriptive models can influence conceptual designs, whilst the third example illustrates how production design can also be influenced. The impact on the design process is then discussed.

3.3.3.1 Conceptual Design Example 1 – A rear cross car beam

A structure was required for a coupe convertible which had the following functions;

- Provide hinge points for the folding roof.
- Provide attachment points for the roll over hoops.
- Fit into the design volume provided by the customer.

Thus, the customer provided “designable” regions, i.e. the package space that was available and “non-designable” regions, i.e. the following attachment points;

- Attachments to the b-pillars.
- Attachments to the floor.
- Attachments to the hinge points for the folding roof.
- Attachments to the roll over hoops.

These regions can be seen in Figure 3-12;



Figure 3-12: Non-Designable & Designable Regions.

These two regions were then combined to form the complete package space model as seen in Figure 3-13. This was then used as the base for the initial Topology run.

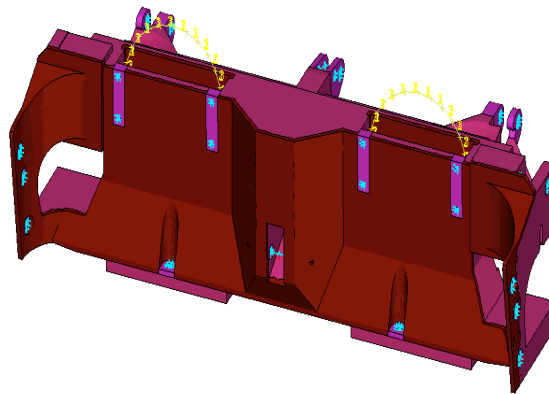


Figure 3-13: Complete Package Space Model.

A Topology run was set up with two loadcases that represented the rollover of the vehicle and the roof opening. There were two requirements for these loadcases, one that the structure did not fail in the rollover loadcase and two, that there was no fatigue failure in the roof opening loadcase. To ensure that no failure occurred in the rollover loadcase the amount of plastic strain seen by the Magnesium must be limited to 6% and to check no fatigue failure happens in the roof opening loadcase the stress seen in the Magnesium must be limited to 70MPa.

Optimisation techniques are currently available only in linear software (although coupling of optimisation with non-linear software is now available in software such as

Genesis). Hence to define the loadpaths in this design the non-linear rollover loadcase was approximated to a linear loadcase in the Topology run. Hence a Topology run was set up with a linear rollover loadcase and a linear roof opening loadcase with the designable region as shown in Figure 3-13 with the objective to minimise strain energy in the structure and the constraint of only using 30% of the material by mass. The results of this initial Topology run can be seen in Figure 3-14 with the major loadpaths highlighted. These Topology results were then interpreted into a castable structure as seen in Figure 3-15, in which major loadpaths from the Topology run can clearly be seen.

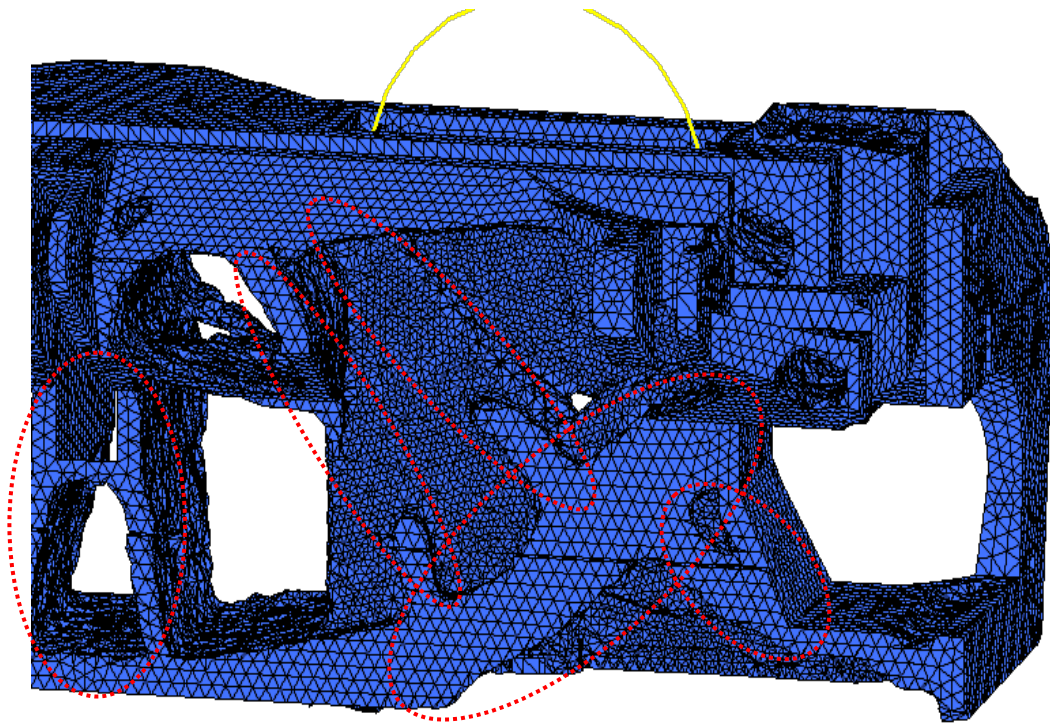


Figure 3-14: Topology Results for the Rear Cross Car Beam.

Once the loadpaths from the Topology run were interpreted into a castable design a shape optimisation study was then carried out to define the size of the sections in this structure, the results of which are seen in Figure 3-16 with the optimised sections highlighted.

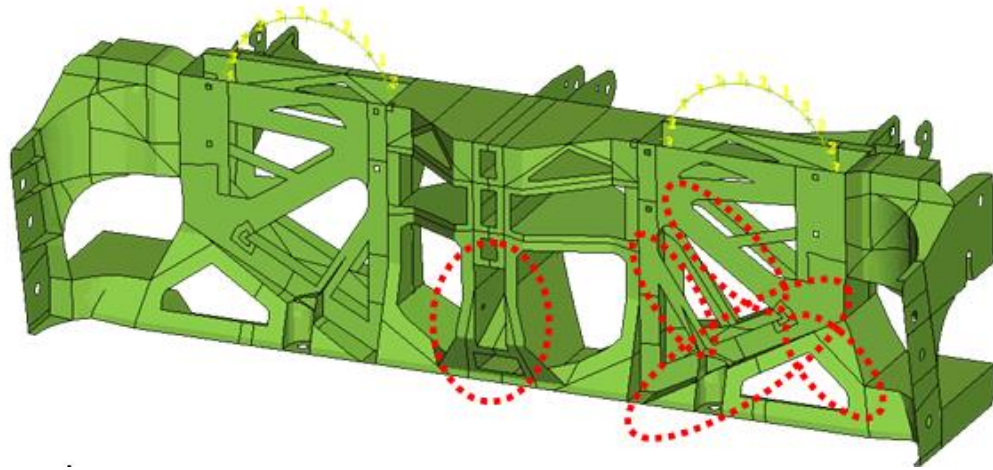


Figure 3-15: Interpretation of Topology Results.

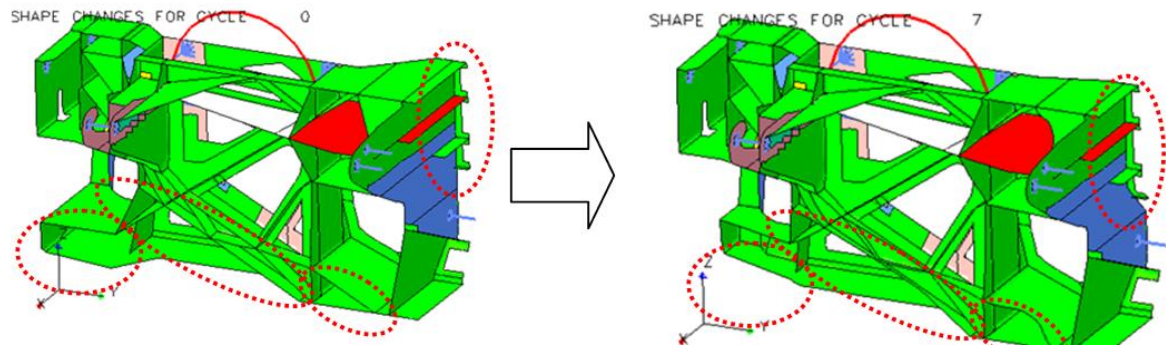


Figure 3-16: Shape Optimisation Results.

Finally, once the position and the size of the sections were established, a Topometry optimisation was then carried out to determine the required thickness of the wallstock whilst meeting the stress constraints. The minimum wallstock that could be cast for this component was 2.3mm. Thus, this was set as the minimum and the structure was then optimised to minimum mass. The results of this can be seen in Figure 3-17.

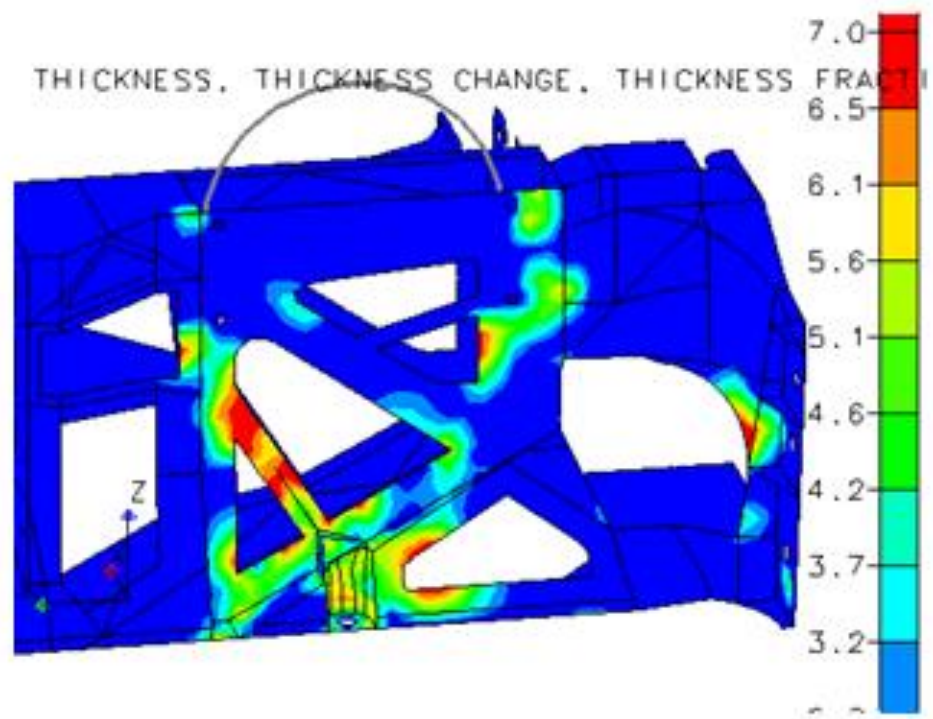


Figure 3-17: Topometry Results.

The Topometry results show the optimum thickness distribution for the structure this then had to be interpreted into a castable structure. The final design can be seen in Figure 3-18.

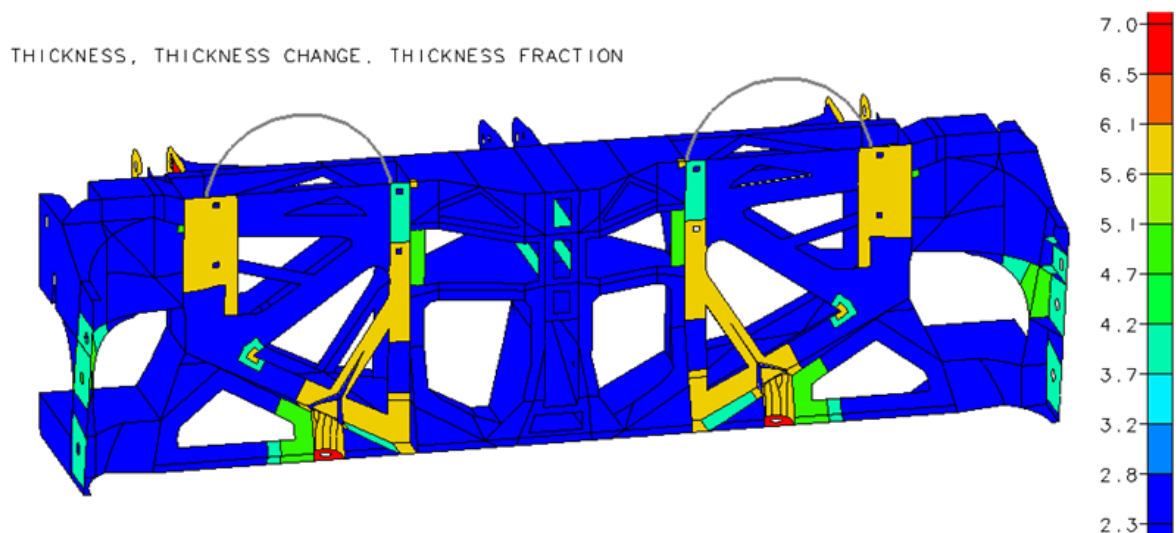


Figure 3-18: Final Optimised Rear Cross Car Beam.

To ensure that that the design met the two loadcases a linear verification model was then set up to ensure that 70MPa was not exceed in the roof opening loadcase and a non-linear model was set up to ensure that 6% plastic strain was not exceeded. The results of these two runs can be seen in Figure 3-19.

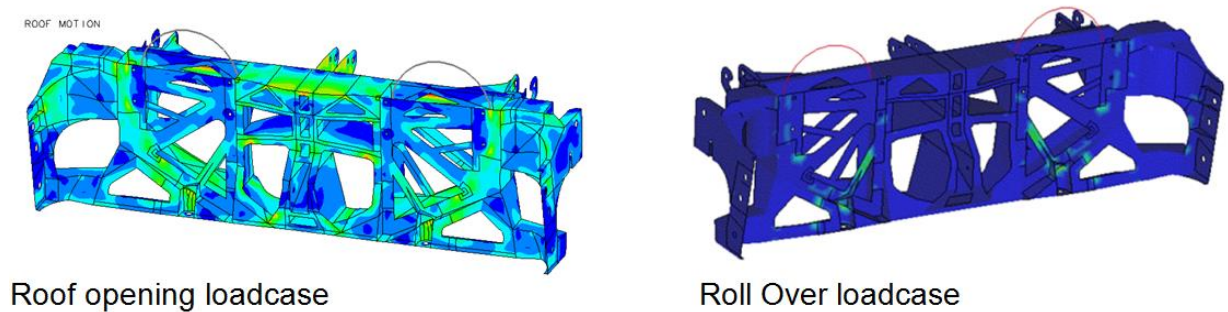


Figure 3-19: Verification Runs.

In the roof opening loadcase the majority of the structure is stressed suggesting a very efficient structure, whilst in the roll over loadcase the diagonal members that were upgauged by the Topometry run are clearly stressed.

The final CAE model is seen in Figure 3-20. This can be delivered to the customer to prove the structural performance of the component and additionally a surface CAD model of the part can be produced within 10 minutes from the CAE model which can also be delivered to the customer for an initial package check.

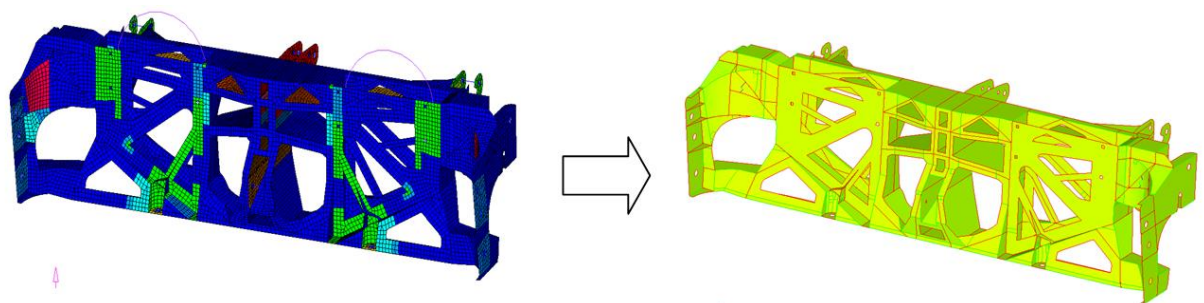


Figure 3-20: The Final CAE Model & Surface CAD Model.

3.3.3.2 Conceptual Design Example 2 – A Front End Carrier

In this example the customer had defined the a-surface of the component but required Meridian to develop the structural performance of the component. The A-surface model of this component, as supplied to Meridian is shown in Figure 3-21.

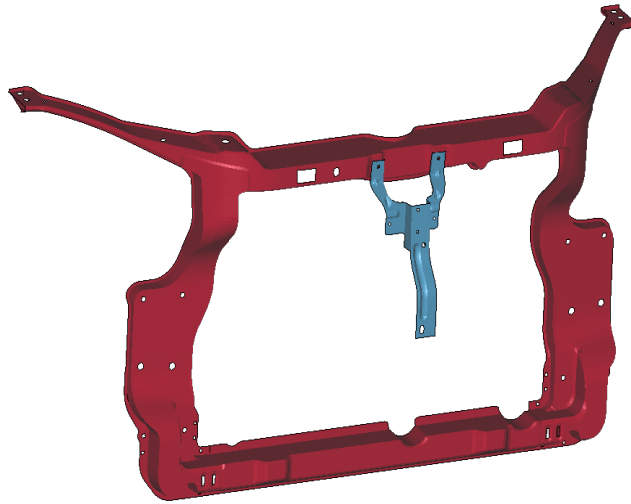


Figure 3-21: Model Supplied to Meridian.

This structure bolts onto the vehicle longitudinals and has the following functions;

- Support the headlamps.
- Support the Horn.
- Support the radiator.
- Provide support to the bonnet.
- Provide vehicle torsional stiffness.
- Withstand low speed impacts.

In addition, a NVH target was set for the structure. Thus, a Topology run was set up with multiple loadcases and the back of the structure filled with designable material to define where the major loadpaths were and thus where the ribbing should be. The multiple loadcases and the designable volume which fills the back of the structure can

be seen in Figure 3-22. The requirement was once again set up to minimise strain energy whilst only using 30% of the available material and a manufacturing constraint was set up so that the ribs could be cast in the longitudinal direction.

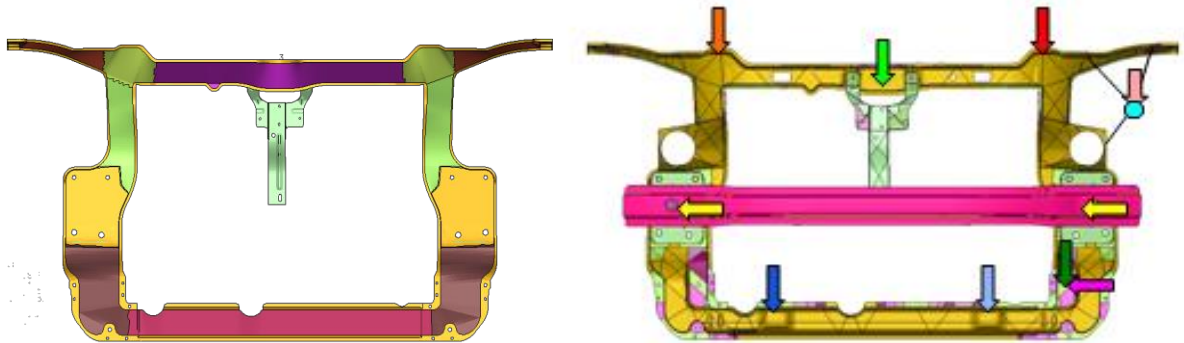


Figure 3-22: Setting up the Topology Run.

The results of this Topology run and the interpretation of these results into a rib pattern can be seen in Figure 3-23.

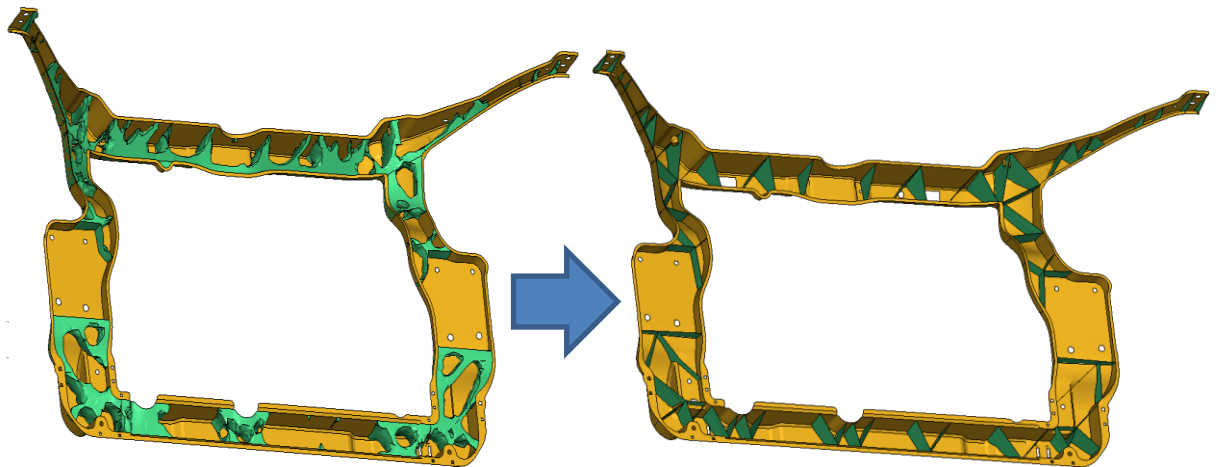


Figure 3-23: Results of the Topology Run & their Interpretation into a Rib Pattern.

Once the ribbing had been defined, a Topometry run was then set up to determine the thickness of the wallstock throughout the structure as seen in Figure 3-24.

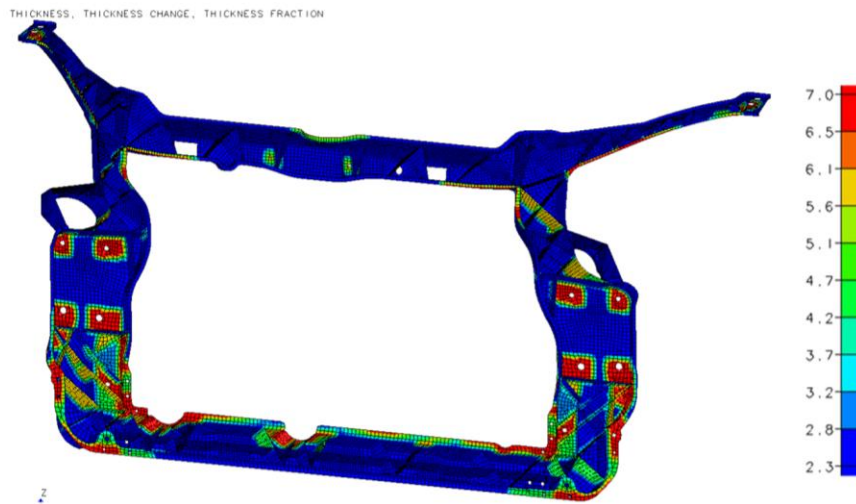


Figure 3-24: Topometry Optimisation.

3.3.3.3 Production Design Example 3 – A Cross Car Beam

The previous two examples shown were both of conceptual designs used by Meridian to try to secure future business. Once Meridian have secured the business, the same optimisation techniques are then used to ensure that the final design is as efficient as possible. A good example of this is the R56 beam as produced by Meridian for the second-generation BMW Mini. This is shown in Figure 3-25.



Figure 3-25: The R56 Cross Car Beam.

This design was an example where the A-surface was provided by the customer to Meridian and then Meridian developed the structural performance of the beam for several loadcases including the following;

- To provide a stiff platform for the steering wheel.
- To provide support to the passenger airbag.
- To provide a structural loadpath in side impact.
- Not to break in frontal impact.

The original Topology runs carried out by Meridian that led to the initial concept can be seen in Figure 3-26, this can then be compared to the final CAD model. The major loadpaths highlighted in the original Topology run can clearly be seen.

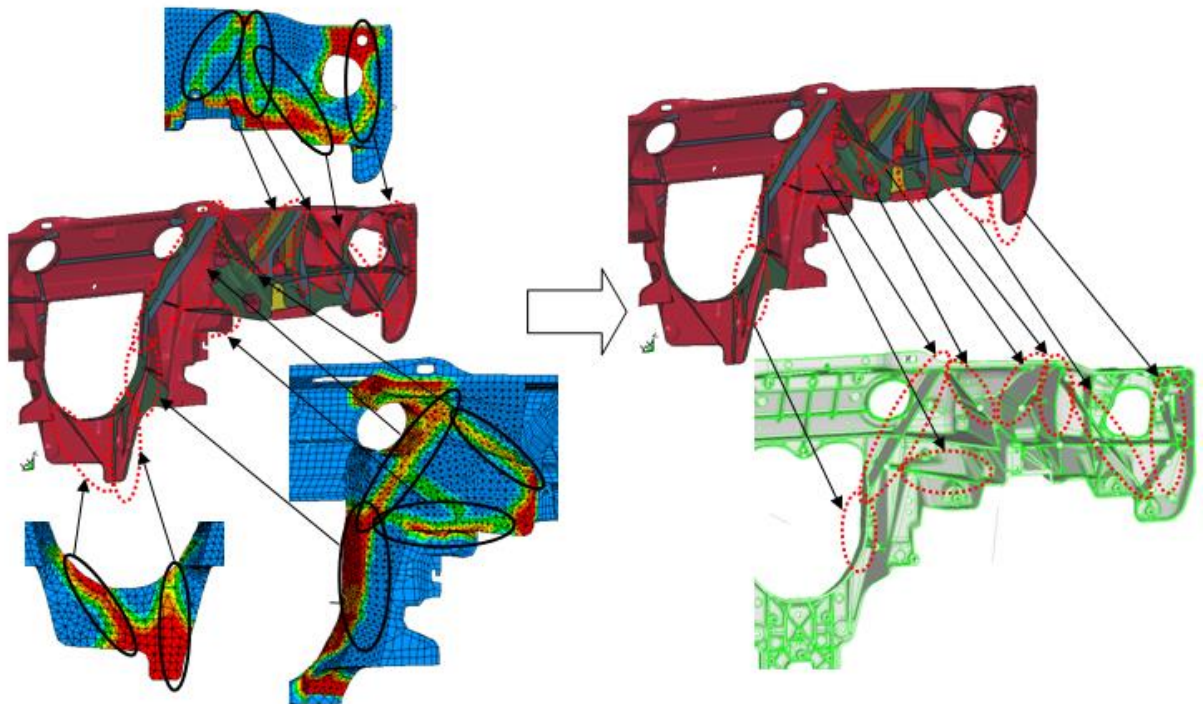


Figure 3-26: Initial Topology Runs done at Concept Through to Final Production.

The early use of Topology ensures that ribbing is added in the most efficient way and only where necessary. This can be compared to a previous generation Meridian beam

as seen in Figure 3-27 where ribbing was simply added in an 'X' structure to make the beam as stiff as possible in all directions, regardless of where the ribbing was actually needed, resulting in a beam not being as efficient as it could be.

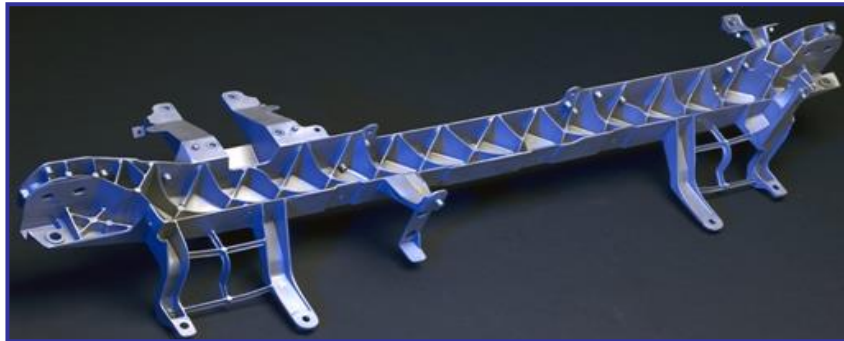


Figure 3-27: Previous Generation Meridian Beam.

Throughout development it is inevitable that as loadcases and the package space matures that the design will need to change. When this is done, the same optimisation tools can and should be used. For example, one such iteration is shown below;

The beam in an interim stage of development was 4.81kg and met all the required loadcases. An optimisation sweep was then carried out on this beam using the following steps;

- Local Topology runs were performed to redefine specific areas of ribbing as seen in Figure 3-28.
- A Topometry run was performed to highlight which areas needed to be reinforced or could be reduced in section as can be seen in the upper image of Figure 3-29.
- A Topometry run was performed to highlight any areas where lightening holes could be introduced as can be seen in the lower image of Figure 3-29.

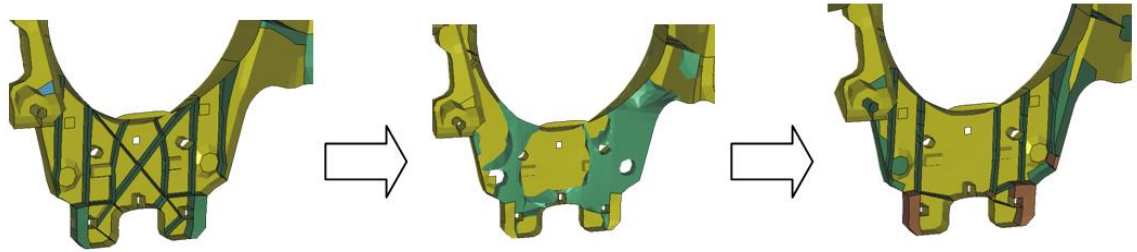


Figure 3-28: Example of a Local Topology Run.

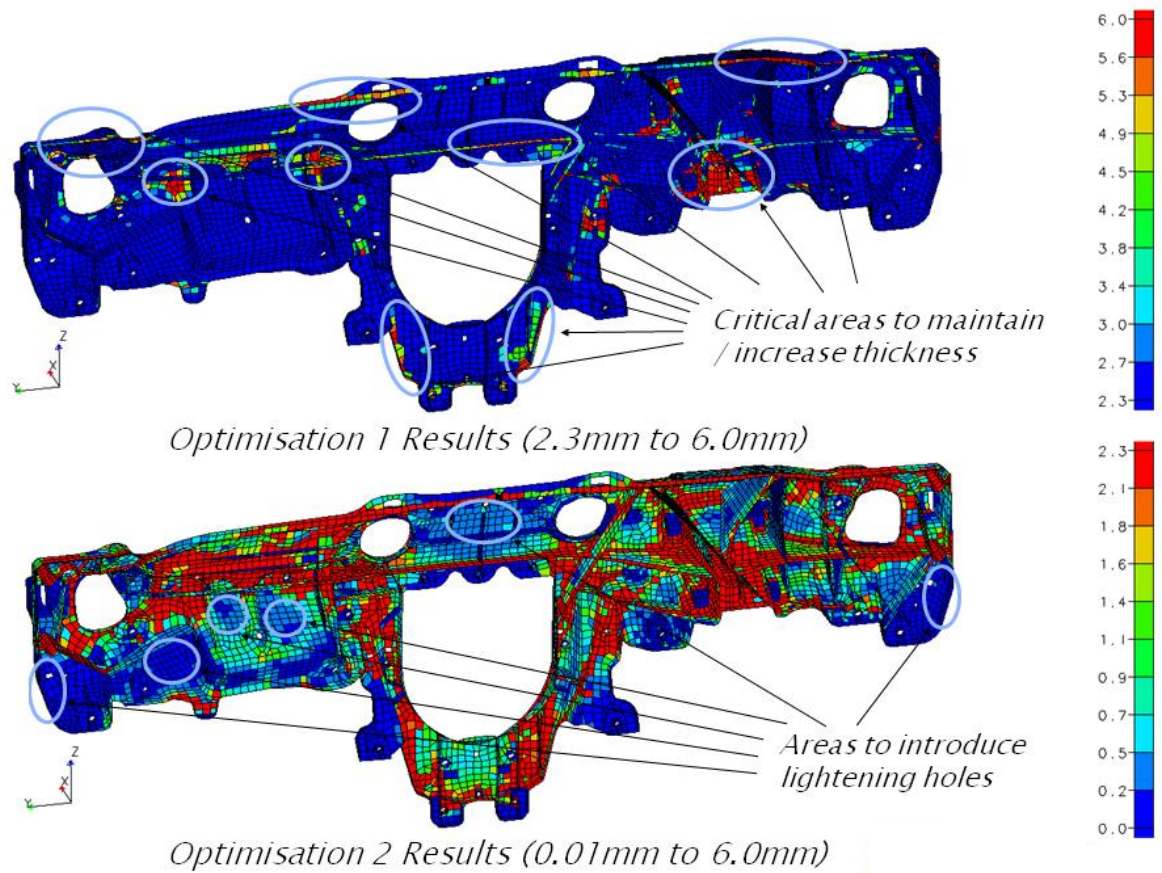


Figure 3-29: Topometry Runs.

The resulting beam still met all of the required loadcases but the mass was reduced to 4.49kg. Thus, the following was achieved by this interim optimisation sweep;

- Mass was reduced by nearly 7%.
- Crash performance remained unchanged.
- NVH performance remained unchanged.

3.3.3.4 CAE Led Design in the Product Lifecycle

The use of CAE early on in the development of Meridian's products has significant effects on the product lifecycle in terms of reducing development time and costs, of both concepts and final designs, particularly when compared to a more traditional CAD led approach. This is particularly important when developing concepts since it is becoming more and more common for OEMs to expect suppliers to offer conceptual designs when quoting for business carried out at the risk of the supplier. Also, once the business has been secured, by using these techniques development times are reduced, helping OEMs to deliver products to market quicker with final designs being as light and as cheap as possible. All of this results in cost savings for both the supplier, the OEM and ultimately, the final consumer.

These benefits already evidenced by Meridian are a direct result of the early adoption of optimisation. As stated by Vanderplaats "Optimization will improve the quality or reduce design for any system, subsystem or component to which it is applied" (66). All of this is achieved at very little expense as the "benefit will exceed the cost" (66) providing a competitive edge over companies that do not employ such techniques as "the companies that strongly embrace this technology will have a clear competitive edge for the betterment of all" (66).

To fully understand this, a comparison will now be made between traditional CAD led designs (Descriptive) and CAE led designs (Prescriptive). A representation of the processes can be seen in Figure 3-30. (This is a reproduction of Figure 2-4 shown here for clarity).

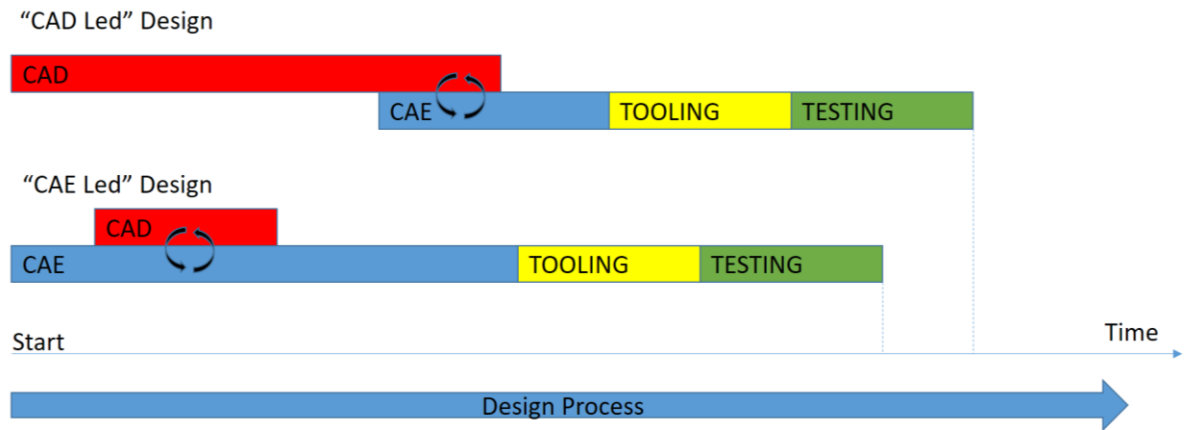


Figure 3-30: Comparison of a CAD Led Design to a CAE Led Design.

In a traditional CAD led design, a concept is first produced in a 3D CAD package. Once an initial design is developed, CAE is used as a verification tool, as Vanderplaats states, “we normally use computer analysis to judge the quality of our designs” (71), not to actually design the parts. Results of this analysis are fed back to the designer, the CAD is updated and a new CAE model is built with the process being repeated until a final design is reached and verified. Once the structural performance of the design has been proved, prototype tooling is produced, and then prototypes are manufactured and tested. Although the use of CAE in judging designs has significantly reduced the product lifecycle time by minimising the number of prototypes required. This is highlighted by Schramm who stated that “Physical experiment has been widely reduced or eliminated to be replaced by computational methods” (74). There are, however, significant opportunities to reduce the time taken to develop a design further by utilising optimisation techniques. This is particularly true for structural orthotropic composite materials in high-performance systems due to the fact of the large number of variables that can be changed to alter the structural performance, e.g. fibre orientation, or the number of plies. This has progressed significantly over the last few years and has been

reported by Zhou et al to be partially driven by the decision from leading aircraft OEMS to design and manufacture composite commercial airlines (87).

Leiva summarises CAD led designs when he says “The design of many of these objects has been done over the years based on judgement, experience of history. Structural analysis to study the behaviour of parts has only been used in the last few decades, Optimization has been used even less” (88). It is by introducing optimisation techniques that the change from CAD led designs to CAE led designs can be made. When describing current design processes Schramm stated that “Simulation can now keep up with design changes and therefore drive design decisions” (74), but by introducing optimisation techniques the CAE model should not only keep up with design changes, but should actually be in front of the design.

In a CAE led design such as the type used by Meridian the design process can be summarised as follows.

- The initial design is developed in CAE using optimisation techniques thus ensuring the major loadpaths are used.
- A CAE model can be built and thus the design can be verified, all of this before any CAD needs to be created.
- Once an initial concept has been created and proved, CAD can then be produced.

Inevitably, there will still be iterations once the initial concept has been developed as the design matures, hence there is still a feedback loop. This results in a verified final design before prototype tools are cut and prototype parts produced and tested. As can be seen in Figure 3-30 this results in a shortening of the design process time.

The benefits of CAE led concepts also give significant benefits to Meridian. To demonstrate this, consider the concept design process. In order to show the impact on timing and cost, the following assumptions have been made. These were typical at the time of these studies.

- Timing Assumptions;
 - The time taken to produce the initial CAD in a 3D CAD package takes 6 weeks which was typical given the complex nature of Meridians parts which have draft, fillets and ejector bosses etc.
 - To produce an initial concept in the CAE environment including a Topology study, Topometry study and structural assessment, takes 2 weeks.
 - To carry out a structural assessment of design takes 1 week.
 - A CAD led approach requires 3 iterations to get structural performance working, which is typical given the complex nature of Meridians products.
- Cost Assumptions;
 - All CAD and CAE is outsourced.
 - The hourly rate of a CAE analyst is £45/hour.
 - The hourly rate of a CAD analyst is £35/hour.
 - A working week is 40 hours.

Using these assumptions, it is now possible to compare the cost and timings of a CAD led design and a CAE led design as shown in Figure 3-31.

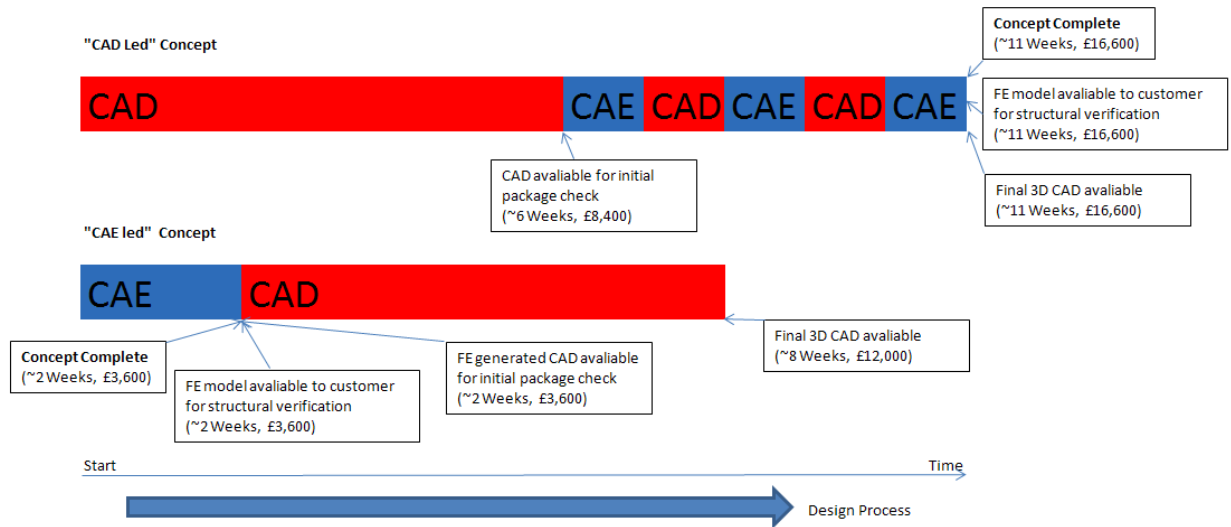


Figure 3-31: CAD Led Concept v CAE Led Concept in the Design Process.

In the CAD led concept, initially a 3D CAD model is produced which takes 6 weeks with a cost of £8,400. This initial CAD concept can be provided to the customer for an initial package check. The design is then assessed in CAE and then goes through two additional iterations before the concept is complete, resulting in the following; (Note that the majority of the deliverables are at the end of the concept).

- CAD is available for an initial package check after 6 weeks and £8,400.
- The final FE model is available to prove the structural performance after 11 weeks and £16,600.
- The final 3D CAD is available and the concept is complete after 11 weeks and £16,600.
- The geometry and weight of the component is known and can thus be costed after 11 weeks at a cost of £16,600.

In comparison, the CAE led design creates a concept purely in the CAE environment within two weeks, at a cost of £3,600. From this, a simple surface CAD model can be produced and delivered to the customer for an initial package check. In addition, the FE model can be delivered to the customer to prove its structural performance. Once the

customer has seen this concept and if they are still interested in the concept, then the expense and time of producing a 3D CAD part is incurred, resulting in the following deliverables;

- The initial concept is complete after 2 weeks at a cost of £3,600.
- The FE model is available to the customer to prove the structural performance of the concept after 2 weeks at a cost of £3,600.
- A simple surface model is available to the customer for an initial package check after 2 weeks at a cost of £3,600.
- The geometry and weight of the component is known and can thus be costed after 2 weeks at a cost of £3,600.
- The Final 3D CAD to the customer after 8 weeks at a cost of £12,000.

As can be seen previously in Figure 3-31, by using a CAE led design as opposed to a CAD led design, the following can be achieved;

- Conceptual designs;
 - Reduced cost and timing by approximately;
 - 80% when 3D CAD is not required.
 - 30% when 3D CAD is required.
 - Designs developed using the CAE led approach are likely to be lighter and cheaper and have more potential to secure the business
- Production designs;
 - Are likely to be lighter, cheaper and more efficient.

In addition to the quantifiable benefits highlighted above there are also the following unquantifiable benefits.

- Improved customer perception of Meridian Technologies advanced engineering capabilities.
- Improved customer confidence in Business Development quotes when quoting for business.
- Improved customer perception of Magnesium and its capabilities.

In a year when Meridian were aggressively pursuing new business they undertook up to 40 conceptual studies and of these approximately one quarter were of enough interest to potential customers to warrant the generation of full 3D CAD models. Thus, these CAE led concepts would cost as follows;

CAE studies	40 x £3,600	= £144,000
3D CAD	$\frac{1}{4} \times 40 \times £8,400$	<u>= £84,000</u>
TOTAL		£228,000

If these studies were all traditional CAD led designs then the cost would be as follows;

CAD led studies	40 x £16,600 = £664,000
------------------------	--------------------------------

Thus, a direct saving of over 65% or over £430,000 can directly be attributed to using optimisation techniques. This is a significant saving, particularly as these concepts are nearly entirely unfunded by the customer.

3.3.4 Making the Design Process more Effective

In order to demonstrate how optimisation can make the design process more effective, a survey was undertaken of 149 full time and part-time students at Swansea Metropolitan University (now UWTSD). The sample of students surveyed was taken from undergraduate and postgraduate programmes in Automotive Engineering, Mechanical and Manufacturing Engineering and in Industrial Automotive Design, where design was a key element of study. Students on these courses came from a variety of

backgrounds including students whose route to university was straight from school, through to mature students who had worked in industry and were returning to education, and students who were still working in industry for large engineering firms such as Ford and Tata Steel and were studying on a part-time basis.

The students were given a 10-minute talk to introduce them to the design problem and then 10 minutes to complete the exercise based on their judgement and engineering experience.



The Baseline Structure



Possible Placements for Reinforcements

Figure 3-32: The Design Problem.

The students were introduced to the baseline structure shown in Figure 3-32. The structure was assumed to be two dimensional and linear. The bottom left hand corner of the structure is assumed to be fixed in all six degrees of freedom and a load is applied to the right-hand end as shown in the picture. The students were then shown how the structure could potentially be reinforced as shown in Figure 3-32.

There were potentially up to 12 reinforcements that the student could use, with the aim to maximise the stiffness to weight ratio. This meant that there were potentially

$\sum_{n=0}^{12} {}^{12}C_n = 3896$ combinations. It can be seen from the above example, how a very

simple design problem very quickly has a huge variety of variables that the designer

must consider. Most structural components such as those highlighted so far, have much more complicated three-dimensional geometry, are subjected to multiple loadcases and many more variables which may include wall thickness, ribbing, constraints and materials. It can clearly be seen that if a design problem is solved using engineering judgement and experience, a sample of engineers will come up with a sample of designs which are not necessarily the optimum design.

In order to find the optimum design, a simple Topology run was set up in Genesis. The FE model had two materials. The first forms the baseline structure seen in Figure 3-33. Inside this there is designable material. The loads and boundary conditions were applied to the baseline structure and the objective was set up to minimise strain energy whilst only using a specified fraction of the designable material for the loadcase described. The results of this analysis and its interpretation can be seen in Figure 3-33.

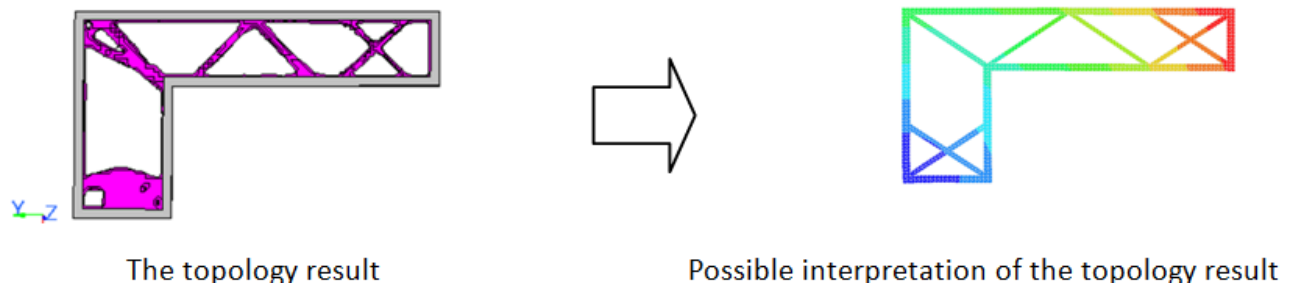


Figure 3-33: Topology Optimisation.

The stiffness to weight ratio of the baseline structure was calculated to be 51.49Nkg/mm. This was normalised to 1. The normalised stiffness to weight ratio of the interpretation of the Topology result shown in Figure 3-33 was calculated to be 5.76. That is to say, that the stiffness to weight ratio of the baseline structure could be increased by a factor of 5.76 if the reinforcements that the Topology result was suggesting were used. The purpose of this survey was to see how much variation

there was in the designs suggested by the population surveyed in terms of the stiffness to weight ratio, and in addition to that, how these designs compared to the design suggested by the optimisation software.

The results of the survey showing stiffness/weight, against the frequency of the response can be seen in Figure 3-34. The mean (3.51) and median (3.32) values of the stiffness/weight ratio were close, but the modal value (2.98) was significantly lower. With a standard deviation of 1.04 the data showed considerable variation. The results are shown, along with a superimposed curve of a normal distribution with a mean of 3.51 and a standard deviation of 1.04.

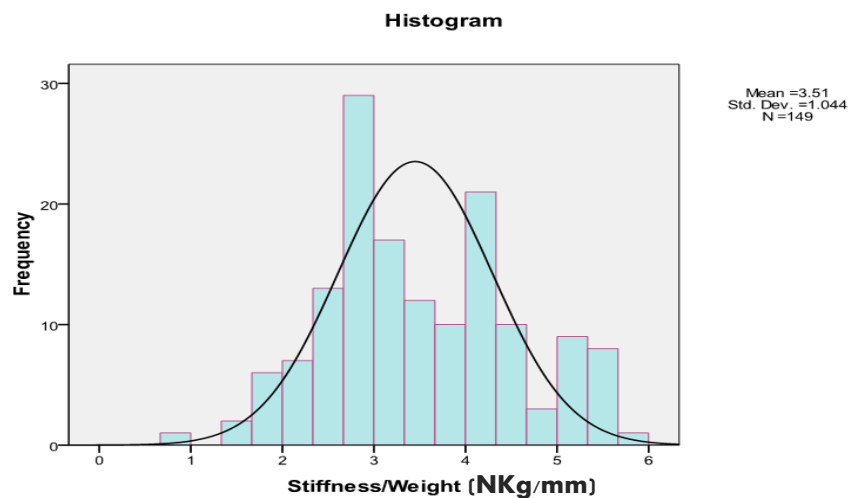


Figure 3-34: Test Results with Superimposed Normal Distribution.

As can be seen, visually there are 'gaps' under the curve and spikes above the curve suggesting the data is not normally distributed as would potentially be expected. In order to test this, the Shapiro-Wilk test was used, which is used for sample sizes of up to 2,000. For the data to be normally distributed the Sig. (p) value must be greater than 0.05. The result of this indicated that in this sample the Sig. (p) = 0.006, thus showing that the data does not fit the normal distribution.

Whilst the results do not follow a normal distribution, if you take the mean value of 3.51 and the standard deviation of 1.04 you can see significant variation in the results. Of the 3896 potential combinations that the sample of 149 could choose, 110 different designs were chosen with a normalised stiffness to weight ratio ranging from 0.95 to 5.76. Interestingly, one of the students came up with a design with a normalised stiffness to weight ratio comparable with the design suggested by the Topology run, although the design was not the same. This demonstrates that although Topology must be used as a tool to aid design, its results are open to interpretation.

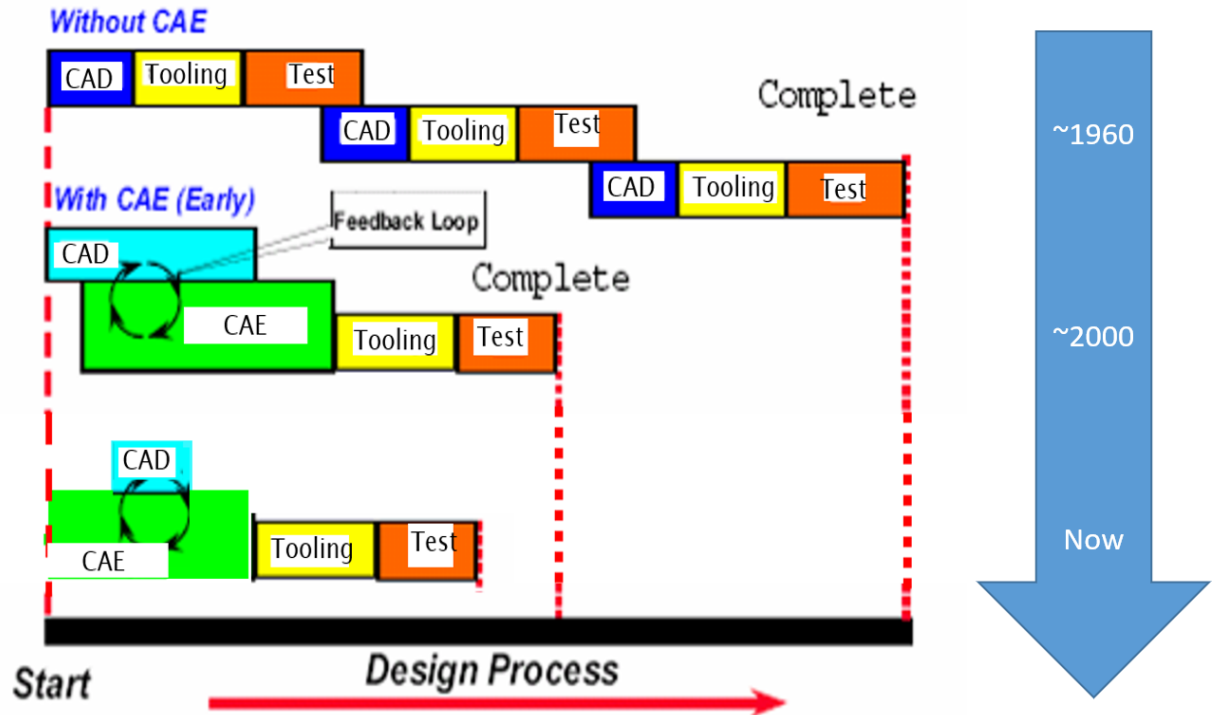
If the mean value of 3.51 is taken, any design below this value may be considered to be 'inefficient' and any designs above this value may be considered to be 'efficient'. Of the sample of 149 taken 44% could be considered to be inefficient and 56% could be considered to be 'efficient', however, only 0.7% of the sample produced a design as efficient as the design suggested by the optimisation software.

The distribution of results shows some tendency to bimodality. It is not clear why this is so, but possible causes include differences in the level of relevant experience possessed by different groups in the sample, or it may be due to one strut which is significant in terms of increasing the stiffness-to-weight ratio of the structure which figures in the choice of a significant proportion of the sample, but not in the selection made by others.

3.3.5 Comparing the Automotive Industry to the Kiteboard Industry

The previous sections in this chapter have provided evidence that there are clear advantages to multinational companies when applying this methodology to their products. This work suggests that SMEs that produce components from orthotropic materials that are subjected to complex loading would also benefit from using this methodology. How the implementation of this methodology can be compared to the Automotive industry can be seen in Figure 3-35.

- Automotive industry



- Composite Industry Example

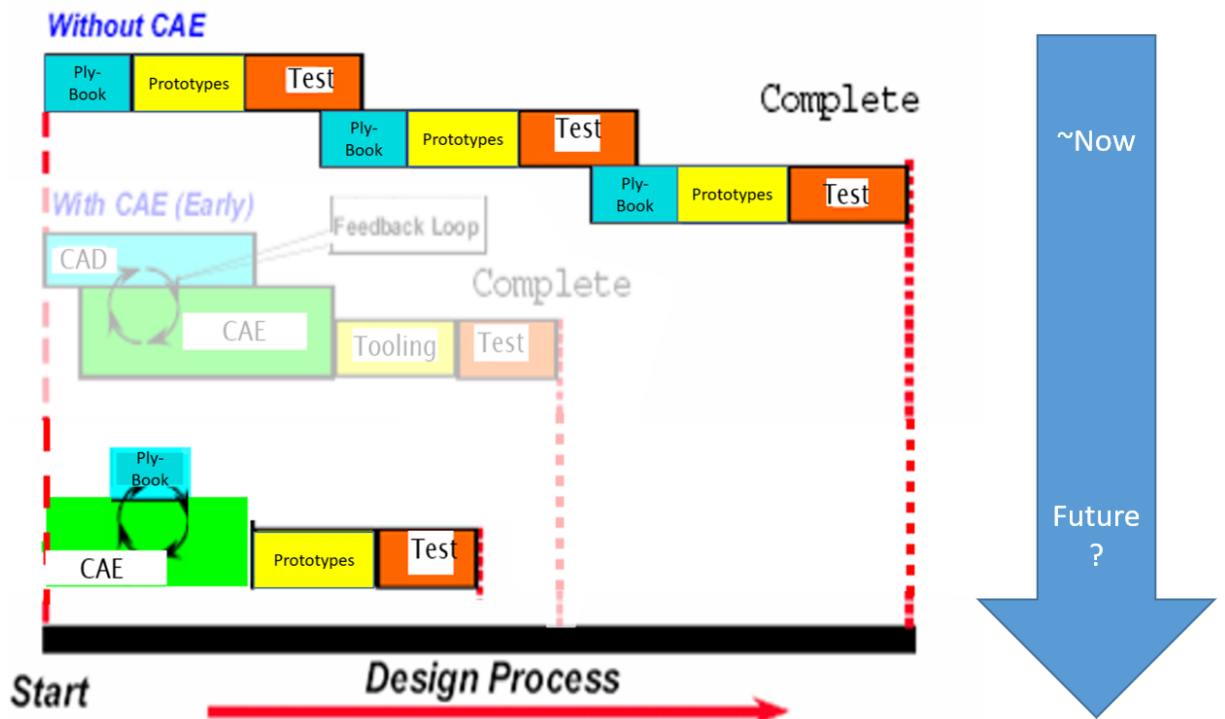


Figure 3-35: The Methodology used by the Automotive Industry Compared to the Kitesurf Industry.

Figure 3-35 shows the methodologies previously discussed for the Automotive industry in the top half of the image and in addition has added approximate time scales to the right of this. As can be seen, development without any CAE was commonplace up until approximately 1960. By 2000 many manufacturers were using CAE to validate their designs, and have since moved on so that CAE is leading the design process using optimisation techniques. The image on the top has been modified so that it can be applied to kiteboard manufacturers using orthotropic materials such as CFRPs (this can be seen in the bottom half of Figure 3-35). As can be seen in this image (on the right), these manufacturers still appear to be using a methodology similar to those used by the Automotive industry in the 1960s, where an iterative approach is used. Instead of producing CAD, a ply-book is suggested, prototypes are then produced and tested, before (inevitably) the ply-book is modified and the process is repeated. Due to the advances in hardware and software discussed in section 3.3.2.1, industries such as these do not need to use the CAD led approach used by many OEMs at the turn of the century (and still being used by many manufacturers). However, this methodology does not lend itself to these products since the CAD remains the 'same' and instead, the ply-book is simply updated. Instead, these industries can move directly to the CAE led approach that has been implemented by the Automotive industry.

3.4 Current Design Methodology

The development of an existing kiteboard may be summarised by the descriptive model shown in Figure 3-36 with the stages of *conceptual design*, *embodiment of schemes* and *detailling* from French's model (previously discussed in section 3.1) identified.

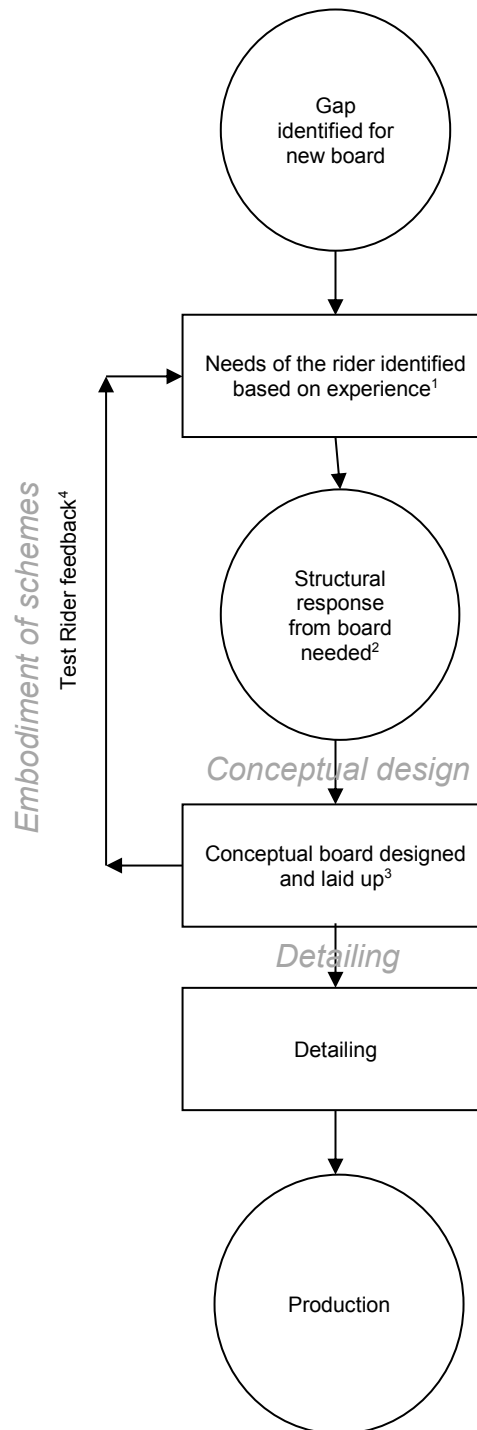


Figure 3-36: Descriptive Design Process of a Kitesurf Board.

Note that this design process has the following flaws;

1. Needs of the rider are not quantified/published so are based on perceived subjective requirements.
2. This is then cascaded down into the structural response of the board.
3. The initial concept is designed and laid-up using experience and judgment rather than employing analytical techniques such as CAE optimisation.
4. Many iterations may be necessary due to the fact that the design is not 'right first time' and even when an 'optimum' design is reached it may be a 'local' minimum.

3.5 Proposed Design Methodology

In contrast to Figure 3-36, Figure 3-37 provides a modified alternative of Archer's model (previously discussed in section 3.2), illustrating how programming, data collection, analysis, synthesis, development and communication can be incorporated into the design process for a component such as a kitesurf board.

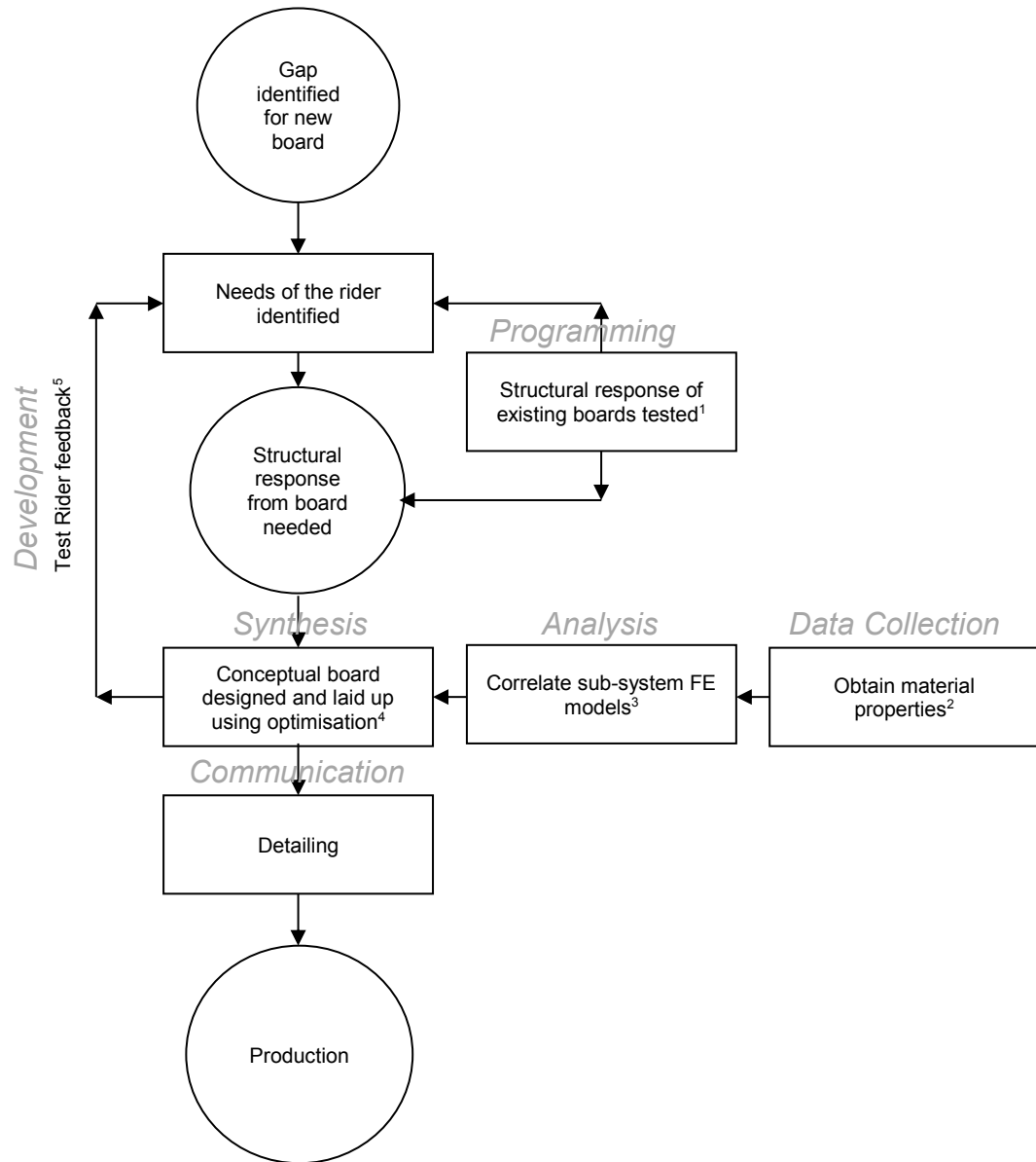


Figure 3-37: Prescriptive Design Process of a Kitesurf Board.

Note the following regarding this proposed design methodology;

1. The Programming stage includes testing of existing structures to see what are the desired characteristics of the component being designed. In this case this would include bending and torsional stiffness' as well as modes of vibration.
2. The data collection stage involves obtaining accurate material information. This could either be via material testing or obtained from material suppliers. In this case this may include material properties such as the stiffness of the carbon fibre and wood used in the production of these boards.
3. The analysis stage would involve obtaining confidence in the modelling of the material properties obtained in the data collection stage so that this can be used in the synthesis stage – i.e. to optimise the lay-up of the structure. This may need to be obtained at a material level and a subsystem level.
4. The synthesis stage should involve optimisation techniques to ensure that an efficient lay-up of the board is achieved.
5. A development stage is still needed to refine the initial design, however, the number of iterations (and thus time) required to do this should be significantly reduced. In addition to this, the final design should be more efficient in terms of weight since a local minimum should not be found.

3.5.1 Pilot Study

In order to demonstrate and evaluate the rationale behind the proposed design methodology, a pilot study was undertaken to demonstrate the potential benefits from incorporating CAE into the design process of this type of structure. The example of a generic skateboard deck was optimised in Genesis as shown in Figure 3-38.

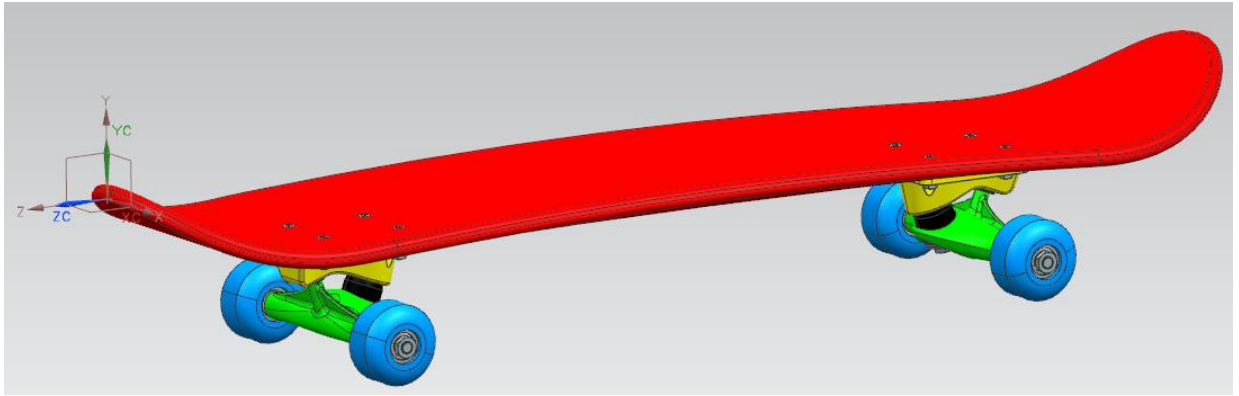


Figure 3-38: Geometry of a Generic Skateboard Deck.

Firstly, a board was constructed using an arbitrary lay-up of unidirectional carbon fibre with typical material properties as seen in Table 3-1;

Young's Modulus in the longitudinal direction	132 GPa
Young's Modulus in the lateral direction	10.8 GPa
Poissons Ratio	0.24
Inplane Shear Modulus	5.65 GPa
Transverse Shear Modulus	5.65 GPa
Mass Density	1540 kg/m ³
Allowable longitudinal stress in tension	1513 MPa
Allowable longitudinal stress in compression	1000 MPa
Allowable transverse stress in tension	43.4 MPa
Allowable transverse stress in compression	43.4 MPa
Allowable inplane shear stress	43.4 MPa

Table 3-1: Summary of Material Properties used for the Unidirectional Carbon Fibre.

It was assumed that the longitudinal (fibre) direction ran lengthways along the structure. This material was laid up in the arbitrary sequence seen in Table 3-2, assuming that each layer was 0.1mm thick. The resulting structure was therefore 20 layers thick, thus 2mm;

Fibres orientated at -90 degrees	4 Layers
Fibres orientated at -45 degrees	4 Layers
Fibres orientated at 0 degrees	4 Layers
Fibres orientated at 45 degrees	4 Layers
Fibres orientated at 90 degrees	4 Layers
TOTAL	20 Layers

Table 3-2: Lay-up Sequence of the Initial Design.

The first resulting mode shape can be seen in Figure 3-39.

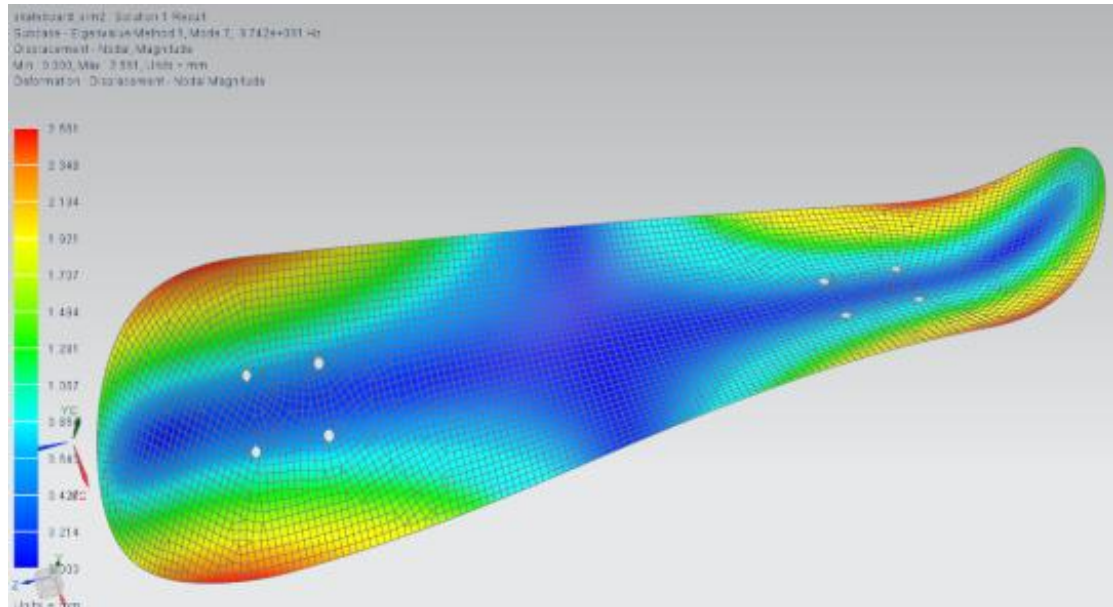


Figure 3-39: First Mode of the Skateboard Deck. Torsion at 41.3Hz.

This was taken as the target frequency of the first mode. This lay-up sequence was then optimised. Initially each orientation was set to 0.1mm and the following requirements were set;

- A first mode of at least 41.3Hz to match the arbitrarily laid up structure.
- The objective was to minimised mass.
- Discrete variables were set so that up to 10 layers could be used in each orientation.

Optimisation was performed out on desktop PC (with an Intel i7 3.4GHz processor with 16GB RAM) and converged on a solution in approximately 2 minutes. The results of this optimisation are as shown in Table 3-3;

Description	Thickness (mm)	Plys Needed
Ply one (-90 degrees)	0.1	1
Ply one (-45 degrees)	0.4	4
Ply one (0 degrees)	0.6	6
Ply one (45 degrees)	0.4	4
Ply one (90 degrees)	0.1	1

Table 3-3: Summary of Optimised Ply Orientations.

The result of this optimisation suggests that this structure should be laid up as shown in Figure 3-40 using 16 layers, giving a thickness of 1.6mm.

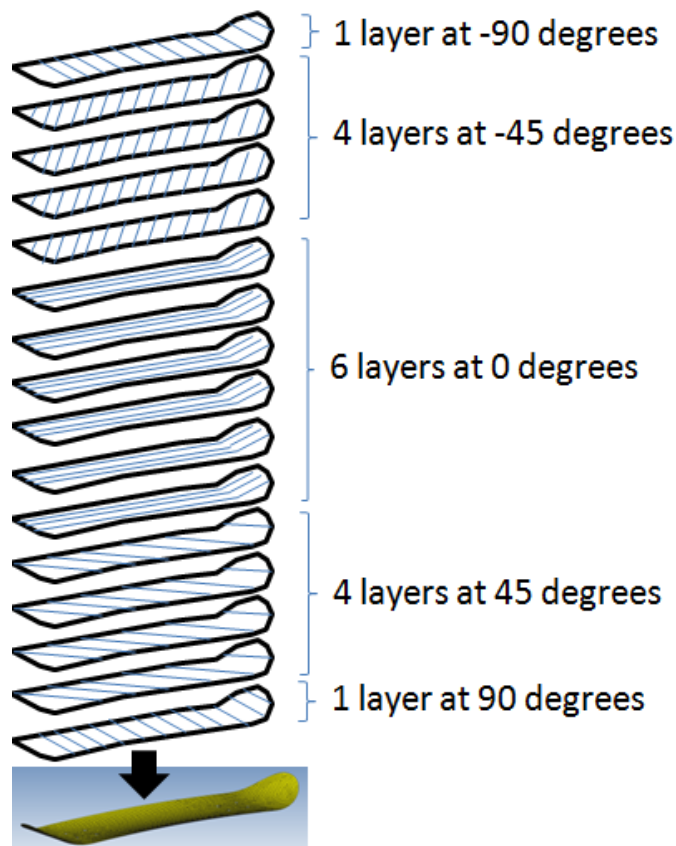


Figure 3-40: Optimised Lay-up with a 1st Mode of the Same Frequency.

A comparison of the performance of an arbitrary laid up board against a board laid up using CAE in the design process is given in Table 3-4.

Design	Thickness (mm)	Weight (kg)	First mode (Hz)
Arbitrary Lay up	2	0.41	41.3
Lay up using CAE	1.6	0.32	42.6

Table 3-4: Summary of the Analysis Carried Out.

Notice that the frequency of the optimised design is slightly higher than the baseline design since discrete layer thicknesses had to be used. Hence a similar structural response is seen in terms of the natural frequency with a weight saving of over 20%. This is particularly interesting when weight saves of 20% have been reported by Reinforced Plastics Magazine through the use of Carbon fibre over a traditional wooden board (44). This suggests that if CAE is incorporated into the lifecycle, in

terms of a skateboard deck, an additional 20% weight save could potentially be achieved over an arbitrarily laid up board.

Obviously, the example presented in this pilot study is relatively simple, with one loadcase and each layer is assumed to go over the entire length on the board. In reality, as shown by the work considered in section 2.2, the board will almost certainly be required to be stiffer in the centre sections, hence additional layers would need to be added around the bindings and the board would need to meet various stiffness, frequency and stress constraints. Additionally, the other materials used in the construction of these boards needs to be modelled, such as the core material and potentially the rails and the bindings also. This example clearly demonstrates the potential advantages of incorporating CAE into the design cycle. It must be noted that this process is accepted to such an extent for components such as this, that Altair even give a very similar example in one of their optimisation tutorials (89).

3.6 Conclusions

Meridian has been used as an example to show how optimisation can and should be routinely used in the design process. Meridian have successfully implemented the tools of Topology, Shape Optimisation and Topometry into their design process. These tools are used right from the start of the design process for conceptual designs and then throughout the development of any products right through to production. It has been shown that these tools reduce the time taken to get fully matured designs. When considering conceptual designs, costs are reduced by up to 80% which is particularly relevant when OEMS are increasingly expecting suppliers to provide conceptual designs at their own cost. These conceptual designs result in more efficient and lighter

products faster, and are more likely to result in securing new business. As stated by Schramm, “Improvements to the design process are driven by the need for shorter time to market, innovation, and product performance” (74).

The optimisation tools used by Meridian are routinely available, yet remain remarkably underutilised. Vanderplaats said that “the key to their use is that we must change the way we do design. The changes are not big, only necessary” (71). This statement is particularly relevant when you consider the recent advances in enablers to optimisation as discussed in section 3.3.2, which mean this change in the design methodology must not only be routinely used by large industries such as the Automotive and Aerospace, but also by SMEs.

4 Material Analysis

4.1 Background

In order to be able to simulate the structural response of a composite component, firstly the mechanical properties of the material and the construction of that composite material must be understood. Composite lamina can be constructed in various ways. The reinforcing agent, in this case fibres, can be woven together, to form sheets to which the matrix is added. Some typical ways in which fibres can be woven together are shown in Figure 4-1 from the work by Vasiliev and Morozov (90).

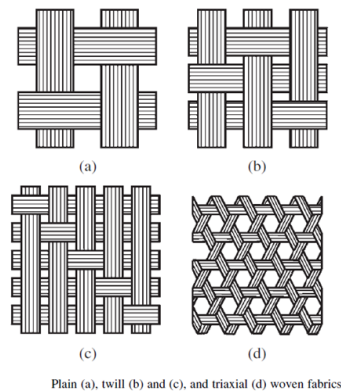


Figure 4-1: Typical Ways in Which Fibres can be Woven Together (90).

The fibres are arranged into fabrics which are made of unidirectional rovings (a collection of untwisted (parallel) continuous strands) (91). These fibres are normally interlaced at 90° to each other in the warp and the weft (or fill) direction as shown in Figure 4-2.

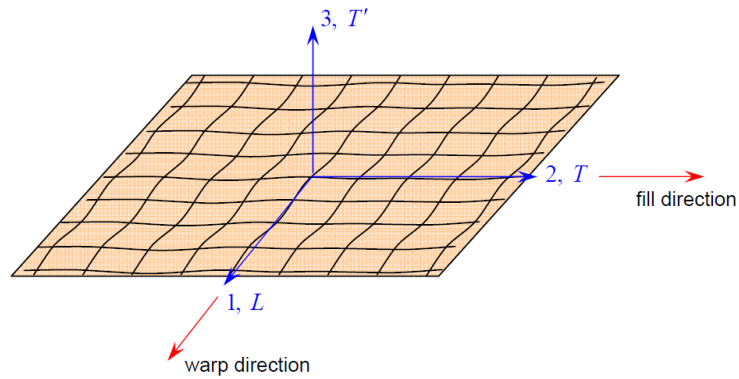


Figure 4-2: Layer of an Orthotropic Composite Material (92).

The longitudinal direction (shown as 1 or L in the figure) is known as the warp direction, the lateral direction (shown as 2 or T for ‘transverse’) is known as the fill (or weft direction) whilst the out of plane direction is denoted as 3 or T’.

An FE model in a readily available piece of optimisation software (Genesis) is now used to simulate the structural response and then to optimise the composite in the design methodology. In order to simulate a composite in Genesis, an orthotropic material (known as “mat8”) was used. In order to define this material, the following properties were needed;

1. Youngs modulus in the longitudinal direction.
2. Youngs modulus in the lateral direction.
3. Poissons ratio.
4. Inplane shear modulus.
5. Transverse shear modulus (1-Z plane).
6. Transverse shear modulus (2-Z plane).
7. Mass density.
8. Allowable longitudinal stress in tension.
9. Allowable longitudinal stress in compression.

10. Allowable transverse stress in tension.
11. Allowable transverse stress in compression.
12. Allowable inplane shear stress.

These values change dependent on the strain value of the fibres (how the fibres move relative to each other) that make up the composite, as it is draped over the mould of the component during manufacture. Previous testing by Lemmerling (93) showed that the fibres in the pre-preg used in this work can potentially strain up to 32 degrees as seen in Figure 4-3.

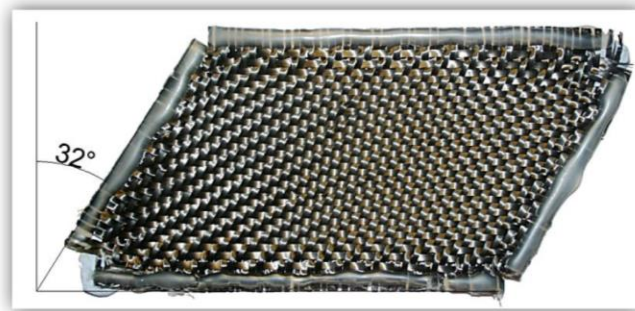


Figure 4-3: Straining of Composite Material During Draping (93).

Therefore, for a highly curved component, testing must be carried out with samples that have been prepared with varying degrees of strain values such that these values can be entered into the draping software and the mechanical properties of the individual elements are adjusted accordingly. However, since the components considered in this study had relatively small curvature, the influence of draping on the mechanical properties was assumed to be negligible. This significantly reduced the amount of testing needed since material properties could simply be found with zero strain.

4.2 Macromechanical vs Micromechanical Methods

When calculating the mechanical properties of woven composite materials, there are two approaches that can be used, Macromechanical and Micromechanical.

Micromechanical methods, such as those outlined by Vasiliev and Morozov (90) use the mechanical properties of the constituent parts (for example the fibres and the epoxy) of the composite and then use geometric techniques to understand the influence of these properties once the fibres are woven into the fabric. Once this is known, the properties of the composite can then be calculated. For example, if a fabric were to be constructed with the geometry shown in Figure 4-4 (from the work by Vasiliev and Morozov (90)) and the material properties of the constituent parts were known then, the Young's modulus of the fabric, E_a , could be calculated as shown in Equation 4-1.

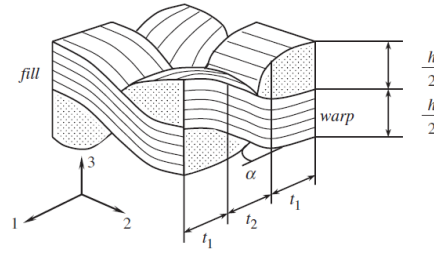


Figure 4-4: Micromechanical Analysis of a Composite Structure (90).

$$E_a = \frac{E_2}{2} + \frac{E_1(4t_1 + t_2)}{4 \left\{ 2t_1 + t_2 \left[\cos^4 \alpha + \frac{E_1}{E_2} \sin^4 \alpha + \left(\frac{E_1}{G_{12}} - 2\nu_{21} \right) \sin^2 \alpha \cos^2 \alpha \right] \right\}}$$

Equation 4-1: Young's Modulus of a Woven Fabric Using Micromechanical Methods.

Obviously, this technique requires the composite material to be broken down into its constituent parts (fibres and epoxy for example), the mechanical properties of these materials need to be established via testing and then the geometry of the assembled composite needs to be analysed so that the material properties of the final composite material can be estimated. This also demonstrates the impact that draping will have on a composite such as the ones being considered in this study as α would clearly change during the manufacture of the product. In contrast to this, in order to calculate the

Macromechanical properties of a composite, the fully assembled composite can be tested and the results of these tests used to establish the macroscopic properties without the need to analyse the material at a microscopic level. For the purpose of this study and to simplify the testing that needed to be undertaken, the decision was made to consider the material properties using a Macromechanical approach.

4.3 Failure Criteria of Composites

Whilst needed in the material data requested by Genesis, this information did not influence the results seen in this study since the material only acted within the elastic region. However, the failure criteria was considered in the final proposed lay-up of the board to see how robust this design was in section 6.7. Various failure methods can be used when considering failure, including those listed by Nafems (94) shown in Equation 4-2 to Equation 4-10.

- Maximum stress criteria

$$\sigma_{f1} = \frac{\sigma_{1u}}{\cos^2 \theta}$$

$$\sigma_{f2} = \frac{\sigma_{2u}}{\sin^2 \theta}$$

$$\tau_{f12} = \frac{\tau_{12u}}{\sin \theta \cos \theta}$$

Equation 4-2: Maximum Stress Failure Criteria.

- Maximum strain criteria

$$\varepsilon_{1uT} = \frac{\sigma_{1uT}}{E_1}, \varepsilon_{1uC} = \frac{\sigma_{1uC}}{E_1}$$

$$\varepsilon_{2uT} = \frac{\sigma_{2uT}}{E_2}, \varepsilon_{1uT} = \frac{\sigma_{1uT}}{E_2}$$

$$\varepsilon_{12u} = \frac{\sigma_{12u}}{\sin\theta \cos\theta}$$

Equation 4-3: Maximum Strain Failure Criteria.

- Tsai-Hill criteria

$$f(\sigma) = \left(\frac{\sigma_1}{\sigma_{1u}} \right)^2 - \frac{\sigma_1 \sigma_2}{\sigma_{1u}^2} + \left(\frac{\sigma_2}{\sigma_{2u}} \right)^2 + \left(\frac{\sigma_{12}}{\sigma_{12u}} \right)^2$$

Equation 4-4: Tsai-Hill Failure Criteria.

- Tsai-Wu Criteria

$$F(\sigma) = F_1 \sigma_x + F_2 \sigma_y + F_6 \sigma_{xy} + F_{11} \sigma_x^2 + F_{22} \sigma_y^2 + 2F_{12} \sigma_x \sigma_y + F_{66} \sigma_{xy}^2$$

Equation 4-5: Tsai-Wu Failure Criteria.

Where

$$F_1 \frac{1}{\sigma_{1uT}} - \frac{1}{\sigma_{1uC}}, F_2 \frac{1}{\sigma_{2uT}} - \frac{1}{\sigma_{2uC}}, F_6 \frac{1}{\sigma_{12uT}} - \frac{1}{\sigma_{12uC}}$$

$$F_{11} \frac{1}{\sigma_{1uT} \cdot \sigma_{1uC}}, F_{22} \frac{1}{\sigma_{2uT} \cdot \sigma_{2uC}} \text{ and } F_{66} \frac{1}{\sigma_{12uT} \cdot \sigma_{12uC}}$$

Equation 4-6: Terms Used in Tsai-Wu Failure Criteria.

- Generalised von Mises Criteria

$$F_{12} = -\frac{\sqrt{F_{11} F_{22}}}{2}$$

Equation 4-7: Generalised Von Mises Failure Index Criteria.

- Cowin

$$F_{12} = \sqrt{F_{11}\sqrt{F_{22}}} - \frac{1}{2\sigma_{12u}^2}$$

Equation 4-8: Cowin Failure Index Criteria.

- Hoffman Criteria

$$F(\sigma) = F_1\sigma_x + F_2\sigma_y + F_{11}\sigma_x^2 + 2B_{12}^*\sigma_1\sigma_2 + F_{22}\sigma_y^2 + F_{66}\sigma_{xy}^2$$

Equation 4-9: Hoffman Criteria Failure Index.

Where

$$B_{12}^* = -\frac{1}{2\sigma_{1uT} \cdot \sigma_{1uC}}$$

Equation 4-10: Term Used in the Hoffman Criteria Failure Index.

As such, each of these failure criteria available in Design Studio was considered to see the failure index of the boards that were designed. These boards were subjected to the mock field testing that is proposed in section 5.4. If the failure index seen was 1, then this would suggest failure according to the failure method being used. The results of this study can be seen in section 6.7.

4.4 Data Provided by Material Suppliers

Initially values for the material properties were taken and calculated from the data provided by the material suppliers and can be seen in Table 4-1.

Property	Supplier Values	Comments
Youngs Modulus in the longitudinal direction (E_1)	58.0 GPa	1
Youngs Modulus in the lateral direction (E_2)	58.0 GPa	2
Poissons Ratio (Nu_12)	0.04	1
Inplane shear modulus (G_12)	27.9 GPa	3
Transverse shear modulus (1-Z plane)(G_1z)	27.9 GPa	2
Transverse shear modulus (2-Z plane)(G_2z)	27.9 GPa	2
Mass density (Rho)	1.51 g/cm ³	4
Allowable longitudinal stress in tension (Xt)	641.2 MPa	1
Allowable longitudinal stress in compression (Xc)	605.8 MPa	5
Allowable transverse stress in tension (Yt)	641.2 MPa	2
Allowable transverse stress in compression (Yc)	605.8 MPa	5
Allowable inplane shear stress (S)	59 MPa	1
THICKNESS /laminate	0.26mm	6

Table 4-1: Supplier Material Properties for Initial FEA.

- Based on Tencate Composites (95) Multipreg E722:T300 (3K) 280g/m2 5HS carbon 0/90° configuration woven laminates, cured 1hr at 120°C (248°F), results normalised to 55% Vf
- Assumed to be the same as E1
- Based on the work by Vasiliev and Morozov (90)
$$G_{12} = \frac{E_x}{2(1+\nu_{yx})}$$

at 45 degrees if $E_1 = E_2$

$$G_{12} = 58e9/2(1+0.04) = \mathbf{27.9 \text{ GPa}}$$
- Using the rule of mixtures for 1cm³ of composite assuming no voids
Resin properties = 1.21g/cm³ based on Tencate Composites Multipreg E722
Fibre properties = 1.76g/cm³ based on Torayca T300 data sheet.
Volume of fibres is 55% based on Tencate Composites Multipreg E722
Mass of matrix, $m_m = \rho_m V_m = 1.21 \times 0.45 \times 1 = 0.54\text{g}$
Mass of reinforcement, $m_r = \rho_r V_r = 1.76 \times 0.55 \times 1 = 0.97\text{g}$
∴ Density of composite, $\rho_c = (m_m + m_r)/V_c = (0.54+0.97)/1 = \mathbf{1.51\text{g/cm}^3}$
- This value was quoted by Tencate composites (95) however this has been modified so that it is the same as the allowable longitudinal stress in tension (641.2MPa) so that the Hill failure criteria can be used.
- Based on average of 6 samples by Lemmerling (93)

The values supplied by the materials supplier were used in the initial analysis that was undertaken. Additional values were calculated using mechanics of materials techniques as outlined in comments 1-6 in Table 4-1.

4.5 Vibration Testing

Once the material properties were established from the materials supplier, the modelling of vibrations in this kind of structure were assessed to establish confidence in these material properties. The simulation and the assessment of vibrations in simple flat carbon fibre structures was undertaken. Testing was undertaken using a Laser Doppler Vibrometer (LDV) and a mechanical exciter with a pulse input to give an automated 'tap test' throughout the component.

4.5.1 Principles of the Laser Doppler Vibrometer

This system, as discussed by Pagano and Lake (96), is used for determining vibration velocity and displacement at a fixed point. The technology is based on the Doppler-shift effect; sensing the frequency shift of back scattered light from a moving surface.

Using an LDV, the laser projects a defined wave (f) equal to $f=c/\lambda$, where (c) will be the velocity which is equal to the speed of light in the case of a laser beam and (λ) the wavelength. If the surface is then excited, it would have its own velocity, displacement and acceleration. The source frequency will be subjected to a variation which has to be considered twice (as this variation is also affecting the reflected frequency (frequency shift)). Therefore (Δf) is related to the actual frequency (f) and the relative speeds of the source (v_s), observer (v_o), and the speed of the wave in the medium (v). This relationship can be seen in Equation 4-11.

$$\Delta f = \left[\frac{v \pm v_o}{v \pm v_s} \right] f$$

Equation 4-11: The Frequency Shift Seen due to a Moving Object.

The sign utilised depends on whether the observer and source are moving towards each other or away from each other. Δf is greater than (f) , if the source is moving towards the observer. Δf is less than (f) if the source is moving away from the observer. Δf is equal to (f) if the source is stationary with respect to the observer. Figure 4-5 gives a visual representation of how the waves stretch or compress depending on a stationary or a moving source as shown by Warwick University (97).

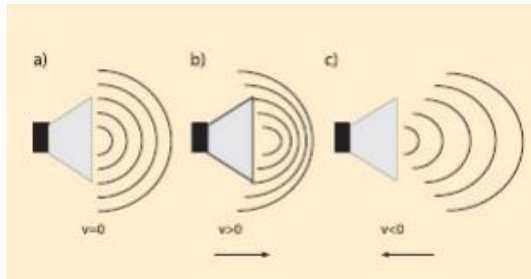


Figure 4-5: Doppler-Effect Representation (97).

This can be then correlated to the frequency shift within the laser beam which will be $\Delta f = 2 \cdot (v/\lambda)$, where v will be the object's velocity and λ is the wavelength of the emitted wave.

4.5.2 Test Procedure and Sample Manufacture

In terms of how to constrain the object whilst using the LDV, two options were considered, clamping the plate along one edge and suspending the plate with elastic bands to simulate a 'free-free' condition. Pictures of these set-ups can be seen in Figure 4-6 and Figure 4-7.



Figure 4-6: Vibration Testing Clamping the Plate Along One Edge.

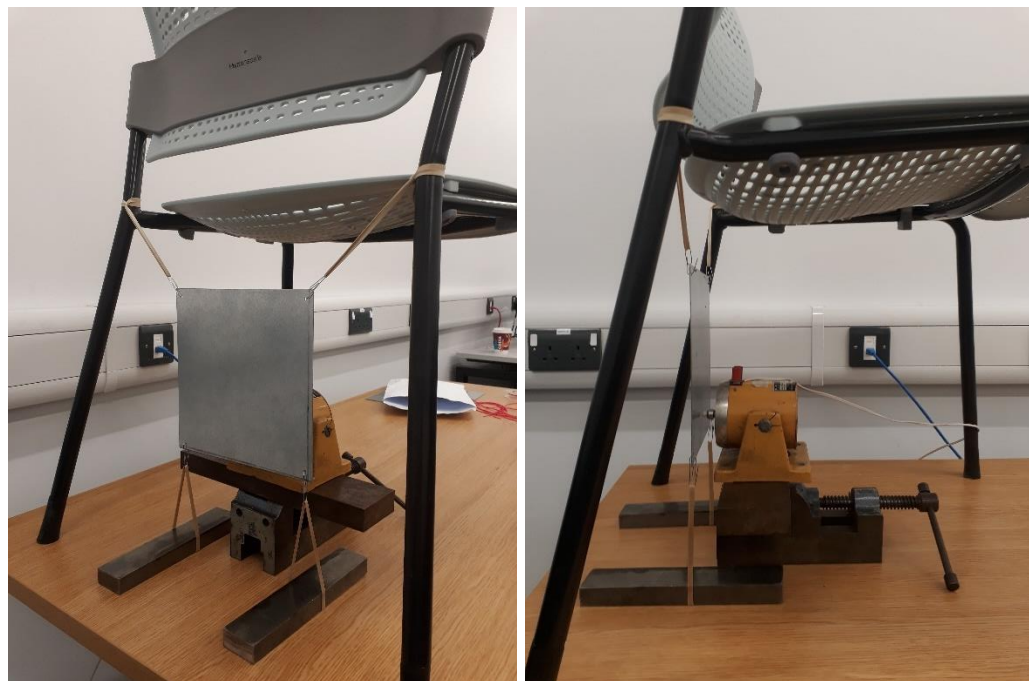
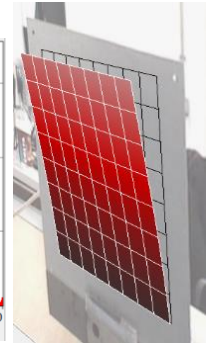
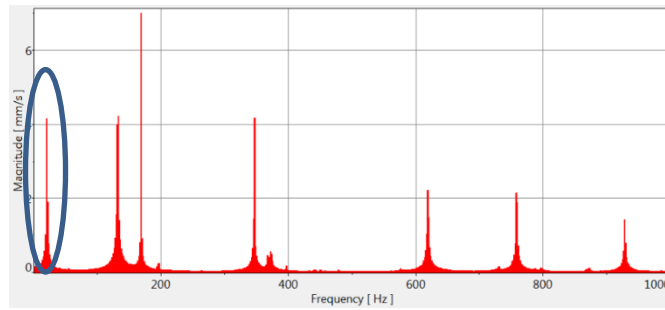
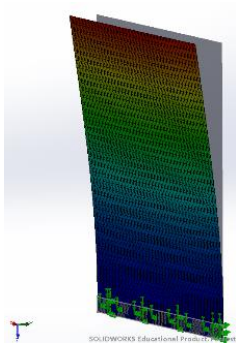


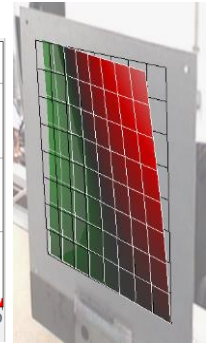
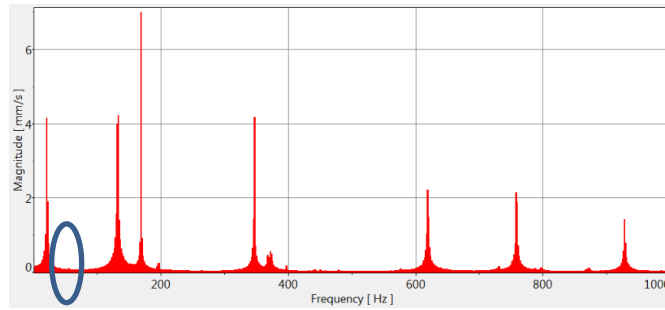
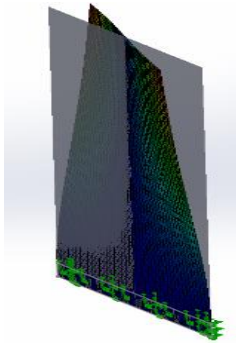
Figure 4-7: Vibration Testing in the 'Free-Free' Condition.

In order to gain confidence in the testing procedure, dummy tests were carried out on an isotropic material (Steel) of similar size and weight to the CFRP samples that would be made. This Steel plate had known properties so that tests could be correlated to equivalent finite element models. The results of this for the two testing scenarios considered can be seen on the right hand side of Figure 4-8 and Figure 4-9. The left hand side shows the equivalent mode in the FE model, whilst the FFT that has been calculated by the LDV is shown in the central column (note that the Velocity is plotted

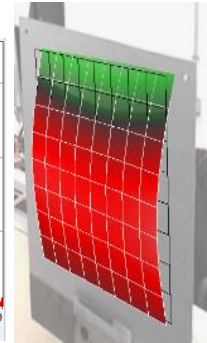
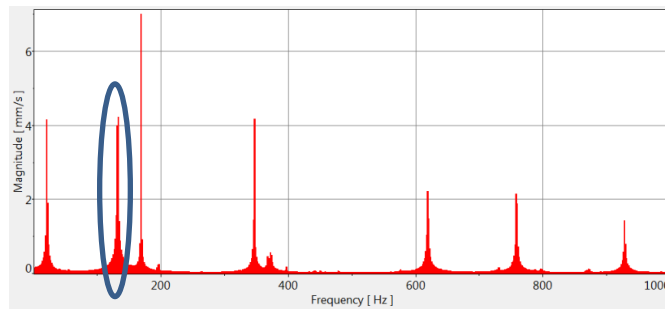
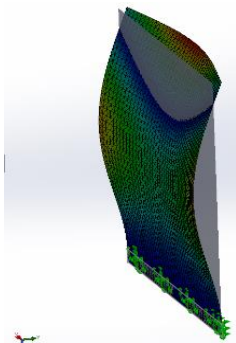
against the frequency that this is seen at, however the LDV software plots this velocity as “Magnitude”). The relevant mode is circled.



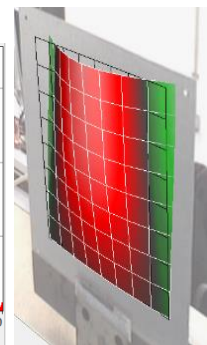
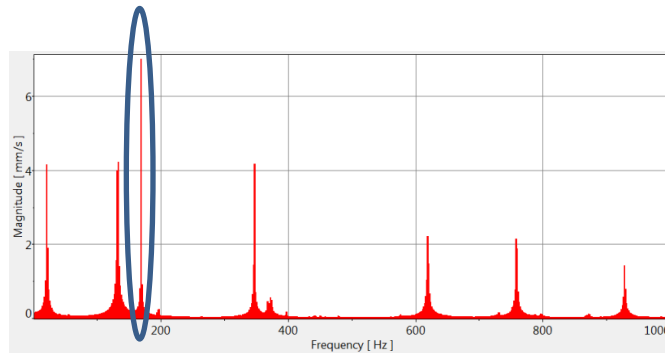
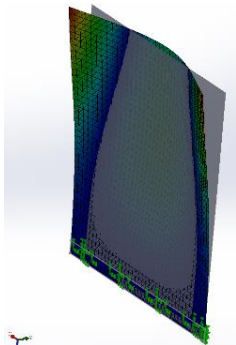
FEA: Mode 1 = 24.1Hz Test: Mode 1 = 20.3Hz



FEA: Mode 2 = 57.2Hz Test: Mode 2 = 54.7Hz



FEA: Mode 3 = 146.9Hz Test: Mode 3 = 132.8Hz



FEA: Mode 4 = 174.8Hz Test: Mode 4 = 168.8Hz

Figure 4-8: FEA v Testing of Steel Plate Clamped Along One Edge.

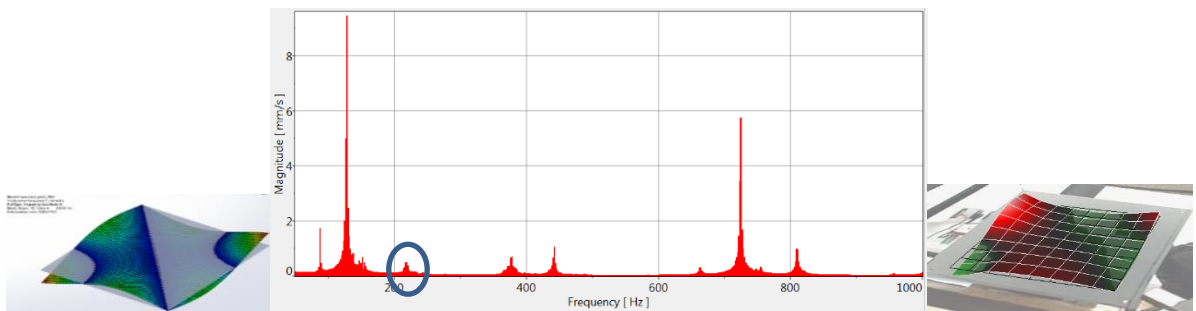
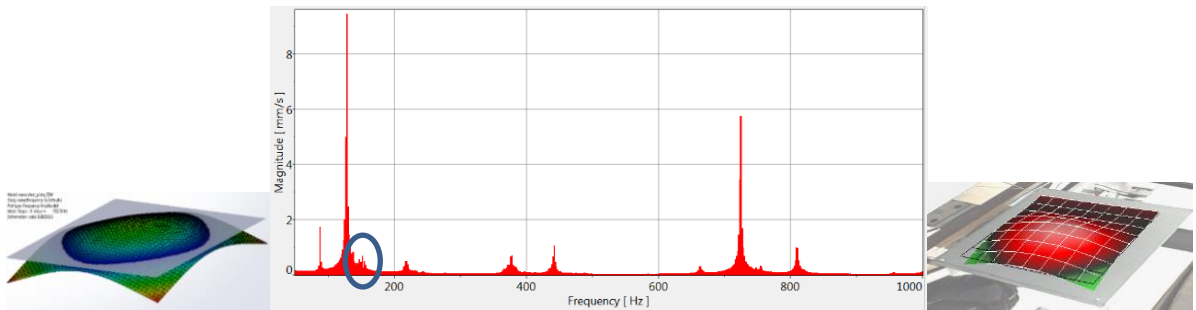
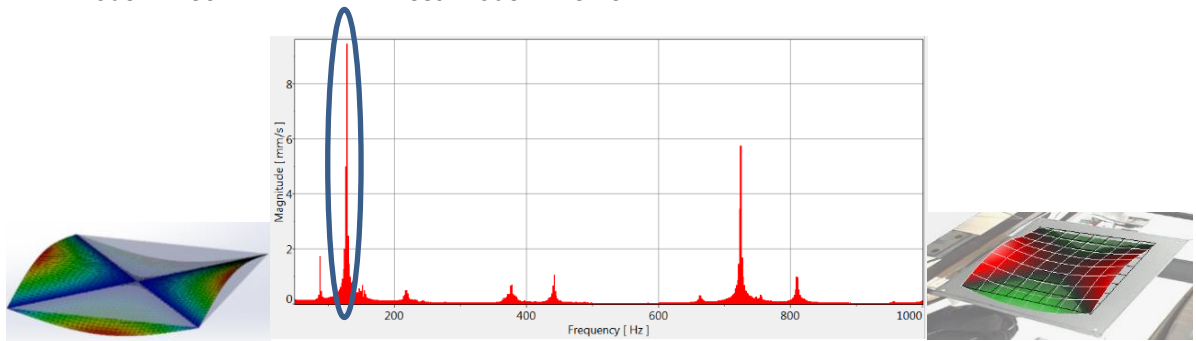
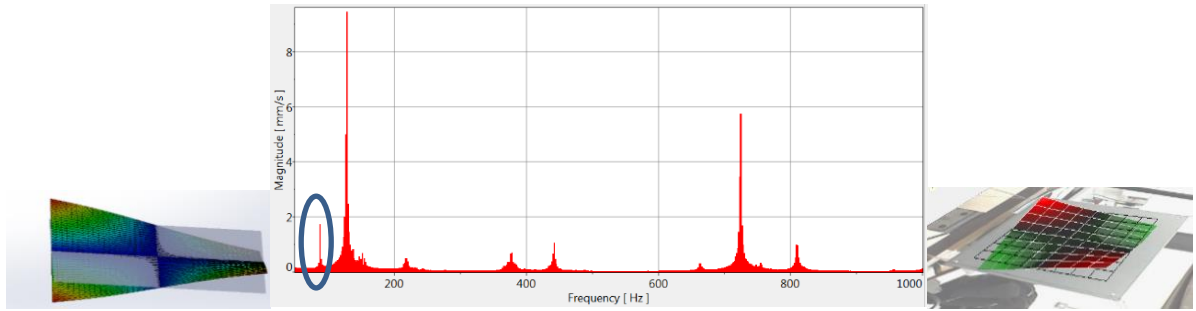


Figure 4-9: FEA v Test for Freely Suspended Steel plate.

As can be seen in Figure 4-8 and Figure 4-9, correlation can be achieved using both sets of boundary conditions, however in order to suspend the plate using the 'free-free' conditions, additional holes would need to be drilled into the samples. Thus to ensure simple repeatable and reliable testing, future tests were carried out with the plate clamped along one edge.

The samples were manufactured and modelled using the material properties established in section 4.4. Testing was based on the study carried out by Mishra and Sahu (51). The study by Mishra and Sahu used plates 0.24m x 0.24m with a thickness ranging from 3.2 to 5.7mm. Their study looked at the influence of the number of plies, fibre orientation and aspect ratio. In order to do this, they laid up plates as follows $[0]_8$, $[30/-30]_4$, $[45/-45]_4$. Their plates were then assessed in a 'free-free' environment. Rather than doing this, the decision was made to fix one edge of the plate to aid correlation to the FE models produced. The dimensions of the plate were also altered to 0.2m x 0.2m. The plate was fixed along the first 25mm of the edge. The physical test set-up of this is seen in Figure 4-10. The part was manufactured from E722, 3k, 2x2 twill 280g/m² using a pressure of 3.0T for 60 minutes and cured at 120°C as suggested by the material supplier data sheet (95).

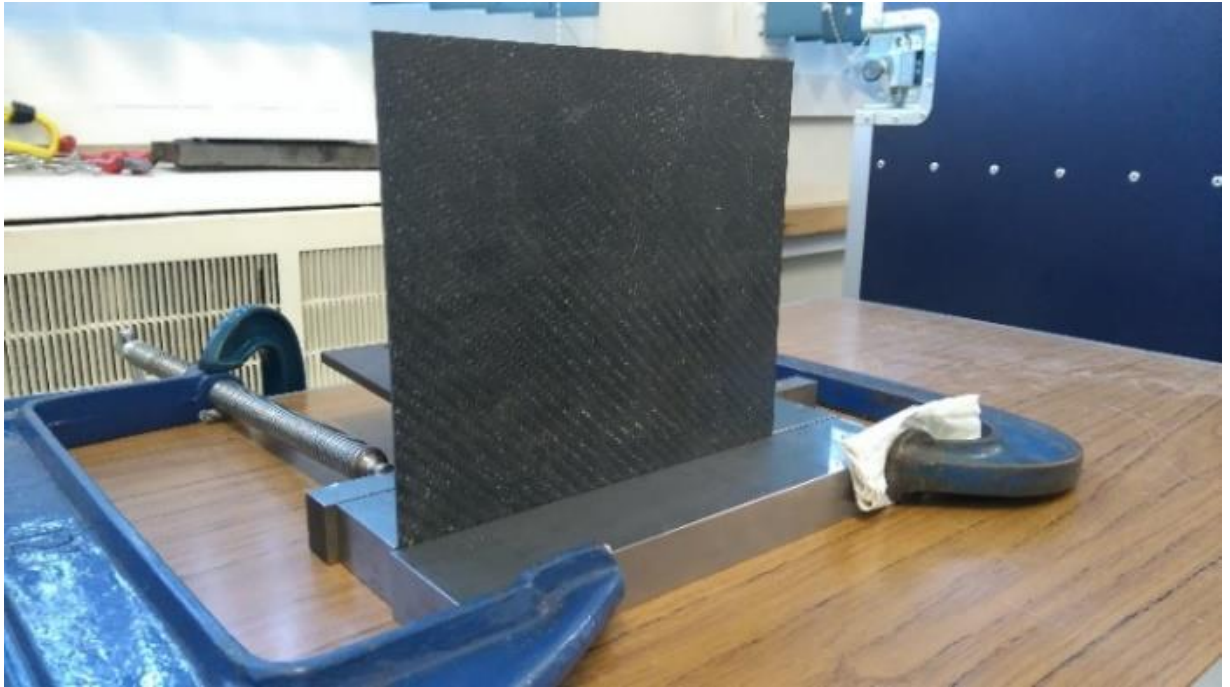


Figure 4-10: CFRP Plate Clamped Ready for Vibration Testing.

In addition, the $[0]_8$ plate was tested at 90 degrees. Thus, three plates of each configuration were made to the following specification meaning that 9 plates were manufactured in total.

- Plate dimensions = 0.2m x 0.2m
- Lay-ups $[0]_8$, $[30/-30]_4$, $[45/-45]_4$

The following conclusions were drawn by Mishra and Sahu (51);

1. Increasing the number of plies increases the natural frequency of the component.
This is to be expected as the stiffness of the structure increases. The decision was therefore made not to alter the thickness of the plates being assessed.
2. Increasing the angle of the fibre reduces the natural frequency of the sample by 11.5 – 13%. This is obviously vitally important in this application and as such, the samples were varied in the same way to see if similar results were observed as this will give confidence in the way that the material is modelled in the final structure.

3. Whilst aspect ratio obviously influences the stiffness of the plate the influence of aspect ratio will be dictated by the shape and the size of the board. Therefore, it was decided to leave the aspect ratio as 1:1.

Models were set up in Genesis using the material properties obtained in section 4.4. Section 4.5.4.1 to section 4.5.4.4 show the vibrational responses predicted for different lay-ups and compares them to the vibrational responses seen experimentally. In this testing the standard deviation has been quoted. The reason for quoting this standard deviation can be seen in appendix B.

4.5.3 Errors

It is important to understand the errors that are introduced during any testing. It is assumed that illegitimate errors (“blunders” or “mistakes”) have been spotted and reduced as much as possible. This includes errors, for example, in the orientation of the fibres during the manufacture or the positioning of any components in the jigs that they are being held in. However, any uncertainties that are inherently introduced by measurements due to the resolution of the measuring equipment should be displayed in any results that are presented. In addition, how these uncertainties combine in any resultant calculations as defined by Bevington and Robinson (98) (error propagation analysis) should be considered. The error propagation equation is given by Lindberg (99) using standard deviations as;

$$\Delta z^2 = \left(\frac{\partial f}{\partial w} \right)^2 \Delta w^2 + \left(\frac{\partial f}{\partial x} \right)^2 \Delta x^2 + \left(\frac{\partial f}{\partial y} \right)^2 \Delta y^2 + \dots$$

Equation 4-12: The Error Propagation Equation (Standard Deviations).

Lindberg then goes on to show that using simpler average errors that;

$$\Delta z = \left| \frac{\partial f}{\partial w} \right| \Delta w + \left| \frac{\partial f}{\partial x} \right| \Delta x + \left| \frac{\partial f}{\partial y} \right| \Delta y + \dots$$

Where $z = f(w, x, y, \dots)$

Equation 4-13: The Error Propagation Equation (Simpler Average Errors).

$$\Delta z = |\Delta x| + |\Delta y| + \dots \quad (\text{for addition and subtraction})$$

Equation 4-14: Error Propagation (Addition and Subtraction).

$$\frac{\Delta z}{z} = \frac{\Delta x}{x} + \frac{\Delta y}{y} + \dots \quad (\text{for multiplication and division})$$

Equation 4-15: Error Propagation (Multiplication and Division).

$$\frac{\Delta z}{z} = |m| \frac{\Delta x}{x} + |n| \frac{\Delta y}{y} + \dots \quad (\text{for products of powers})$$

Equation 4-16: Error Propagation (Products of Powers).

All equipment used during the testing had been calibrated by the supplier of that equipment. Where possible, the uncertainty, related to the equipment had been obtained from the literature associated with the measurement equipment or from previous research. Where this has not been possible it was assumed that the accuracy was to plus or minus one-half of the least count as suggested by Bevington and Robinson (98). The errors considered in the calculations for the vibration testing were therefore as follows;

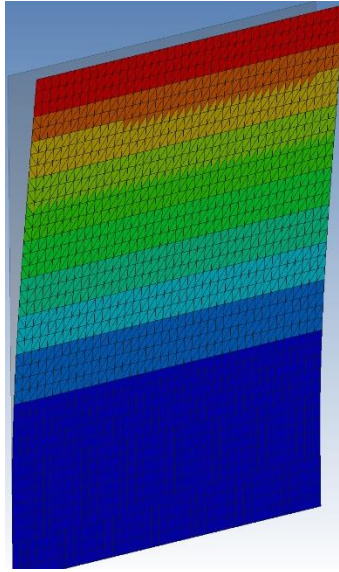
1. The Bandwidth of the LDV was 10KHz whilst the resolution of the Fast Fourier Transform (FFT) was 6400 over the 1000Hz range that was plotted. This means that the error in the frequencies that were quoted by the LDV were $(10000/6400)/2 = \pm 0.8\text{Hz}$.

These errors are shown in any results that are presented in this section.

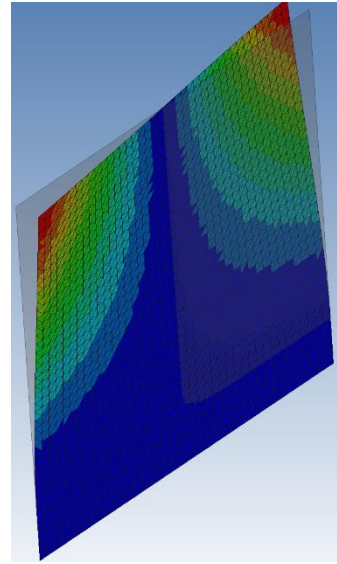
4.5.4 Results

4.5.4.1 $[0]_8$ Sample

The following responses were predicted by the FEA models.



First mode is 68.1Hz



Second mode is 173.1Hz

The LDV experimental results were as follows;

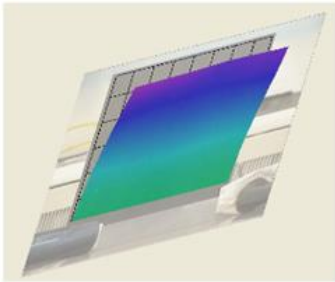
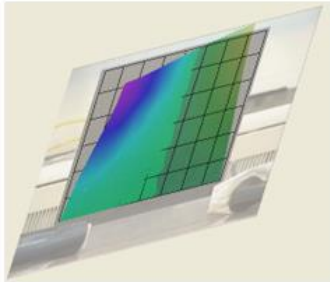
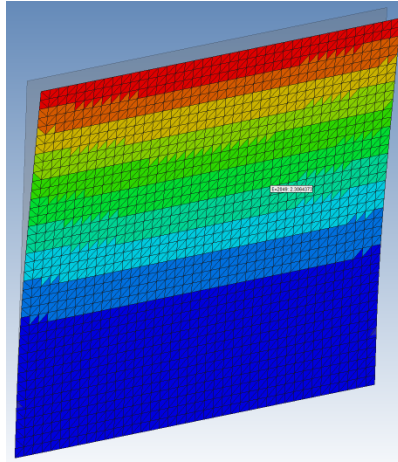
Test	Mode 1 (Hz)	Mode 2 (Hz)
		
1	37.5±0.8	60.9±0.8
2	43.8±0.8	62.5±0.8
3	43.8±0.8	60.9±0.8
Average	41.7±0.8	61.4±0.8
Standard Deviation	3.6	0.9

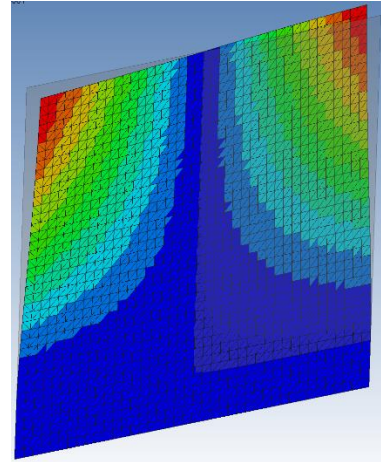
Figure 4-11: Vibrational Response of the $[0]_8$ Samples: FE Model v Experimental Test Results.

4.5.4.2 $[30/-30]_4$ Sample

The following responses were predicted by the FEA models.



First mode is 68.1Hz



Second mode is 173.1Hz

The experimental results were as follows;

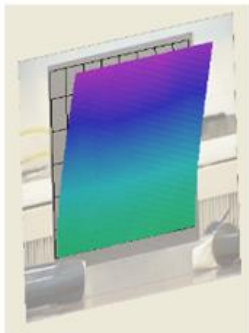
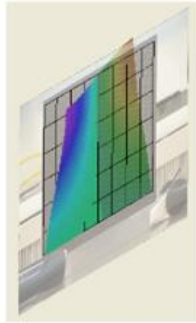
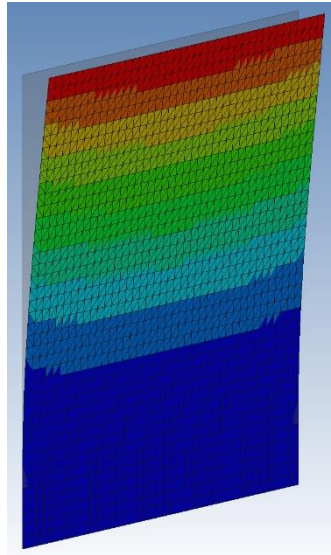
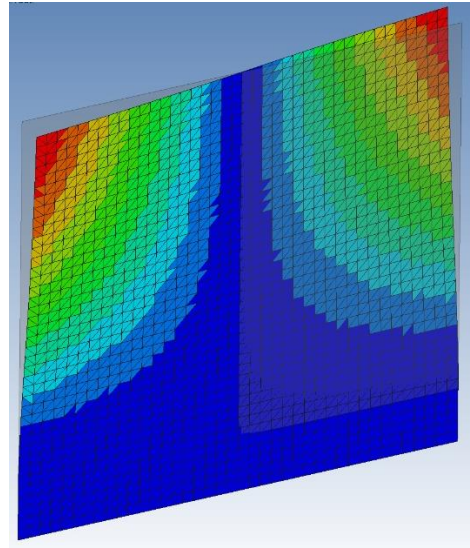
	Mode 1 (Hz)	Mode 2 (Hz)
Test		
1	32.8±0.8	103.1±0.8
2	42.2±0.8	107.8±0.8
3	39.1±0.8	106.3±0.8
Average	38±0.8	105.7±0.8
Standard Deviation	4.8	2.4

Figure 4-12: Vibrational Response of the $[30/-30]_4$ Samples: FE Model v Experimental Test Results.

4.5.4.3 $[45/-45]_4$ Sample



First mode is 68.0Hz



Second mode is 173.0Hz

The experimental results were as follows;

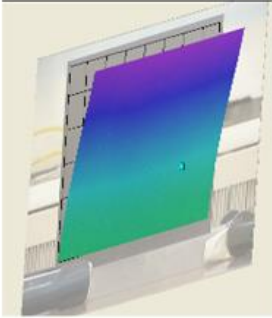
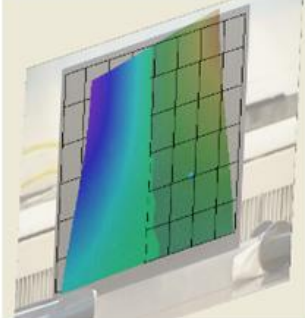
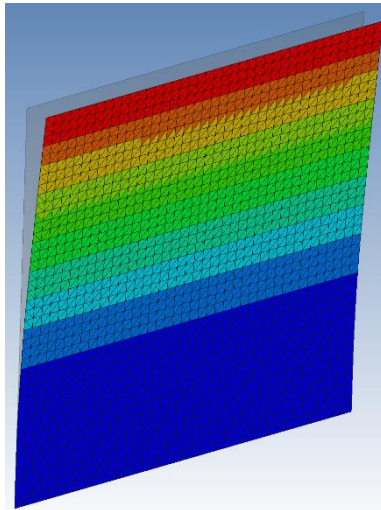
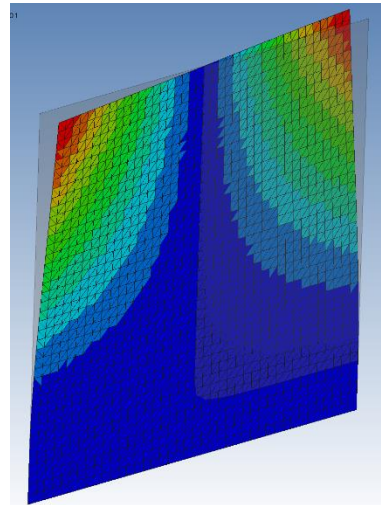
Test	Mode 1 (Hz)	Mode 2 (Hz)
		
1	35.9±0.8	106.3±0.8
2	34.4±0.8	110.9±0.8
3	35.9±0.8	115.6±0.8
Average	35.4±0.8	110.9±0.8
Standard Deviation	0.9	4.7

Figure 4-13: Vibrational Response of the $[45/-45]_4$ Samples: FE Model v Experimental Test Results.

4.5.4.4 $[90]_8$ Sample



First mode is 68.1Hz



Second mode is 173.1Hz

The experimental results were as follows;

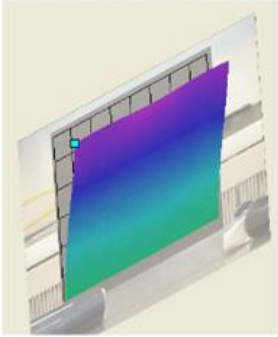
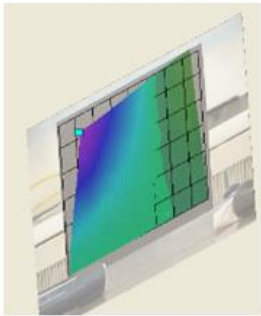
Test	Mode 1 (Hz)	Mode 2 (Hz)
		
1	40.6±0.8	67.2±0.8
2	46.9±0.8	65.6±0.8
3	50.0±0.8	68.8±0.8
Average	45.8±0.8	67.2±0.8
Standard Deviation	4.8	1.6

Figure 4-14: Vibrational Response of the $[90]_8$ Samples: FE Model v Experimental Test Results.

4.6 Correlation Through Scaling Material Properties

The results found in section 4.5 can be seen in Table 4-2. This shows the differences between the modes predicted with the FE models (containing the material properties provided by the material supplier), in terms of cycles per second and percentages.

		[0]8		[30/-30]4		[45/-45]4		[90]8	
		Mode 1	Mode 2	Mode 1	Mode 2	Mode 1	Mode 2	Mode 1	Mode 2
Test 1	(Hz)	37.5	60.9	32.8	103.1	35.9	106.3	40.6	67.2
Test 2		43.8	62.5	42.2	107.8	34.4	110.9	46.9	65.6
Test 3		43.8	60.9	39.1	106.3	35.9	115.6	50.0	68.8
Average		41.7	61.4	38.0	105.7	35.4	110.9	45.8	67.2
stdev		3.6	0.9	4.8	2.4	0.9	4.7	4.8	1.6
Predicted with SUPPLIER material properties	(Hz)	68.1	173.1	68.1	173.1	68.0	173.0	68.1	173.1
Difference		26.4	111.7	30.1	67.4	32.6	62.1	22.3	105.9
Difference		63.3	181.8	79.1	63.7	92.1	55.9	48.6	157.6

Table 4-2: Vibration Testing: FE Models v Experimental (Supplier Material Properties).

As can be seen, there are some differences between the modes predicted and the modes observed. This work now uses the values obtained from the material supplier in section 4.4 and then scales them in order to correlate vibrational testing to gain confidence in the simulated material properties, with the exception of the thickness of one ply and the density. These values can be obtained easily by measuring the thickness and mass of a sample that has been manufactured. This means that the stiffness due to the shape of the component, and the mass will be correct.

The material properties of the CFRP can be obtained relatively easily and then scaled as follows. It is clear that when the fibres are at 0 degrees (vertical) and the part is in pure bending (mode 1) that E_1 would be the biggest influence on this mode. This mode will obviously be defined by the flexural stiffness of the part (EI). If I is correct due to the geometry of the part being correct (the thickness of the plies has been

measured), then E can be correlated. When the fibres are at 90 degrees (horizontal) and the part is in pure bending (mode 1), E_2 is the biggest influence on this mode and in the torsion modes (all modes for the $\pm 30/45$ layups and mode 2 for the $0/90$ layups) the shear stiffness would have the biggest influence. (Since this mode will be defined by the torsional stiffness (GJ), again, since J is known to be correct (due to the geometry of the part), then G can be correlated. To test this hypothesis, these values were scaled to understand the sensitivity of these modes to these material properties and to get the best correlation. The results of these studies can be seen in Figure 4-15 to Figure 4-17 which show the percentage difference between the FE models and the test results for all modes as the material property is changed.

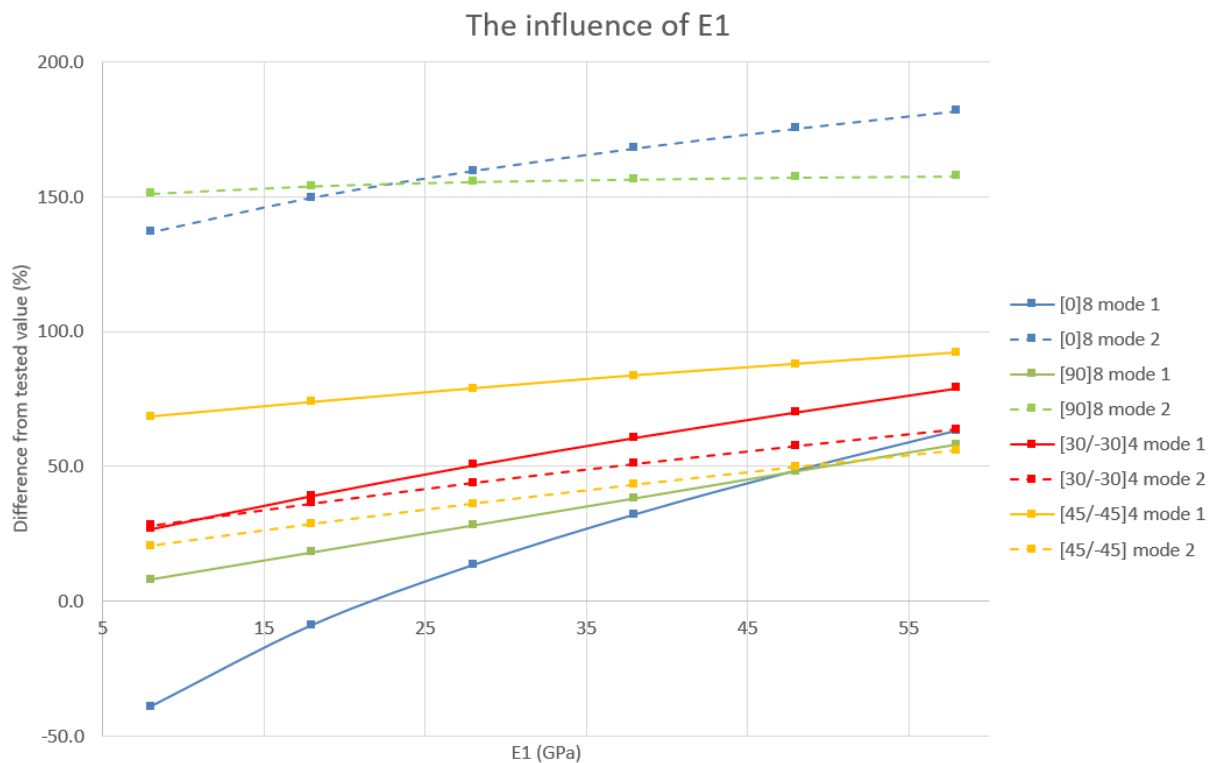


Figure 4-15: The Influence of Longitudinal Stiffness on the Vibration Modes.

E_1 can clearly be tuned such that it influences mode 1 on the $[0]_8$ plate. A 5th order polynomial was therefore fitted to establish that E_1 should be set to 22.0MPa. (The

rationale for using a 5th order polynomial can be seen in appendix C). E_1 was set to 22.0GPa and the exercise repeated to see the influence of E_2 , as shown in Figure 4-16.

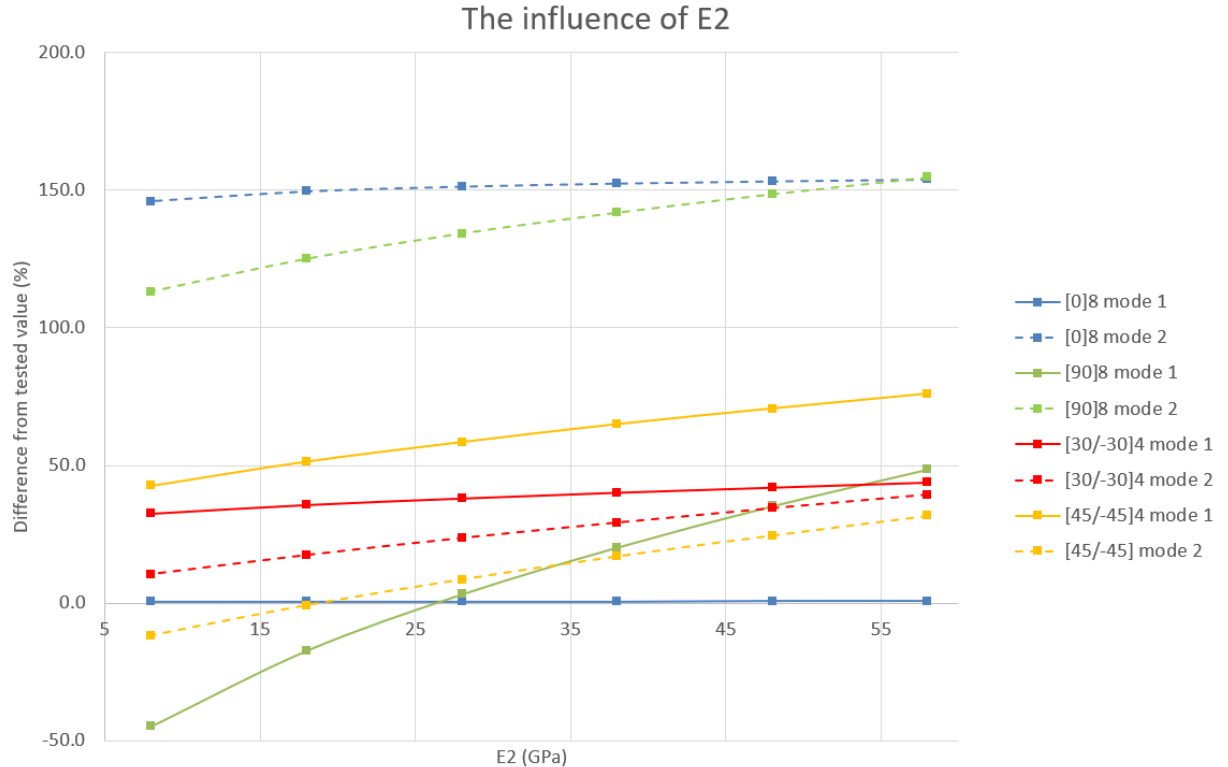


Figure 4-16: The Influence of Lateral Stiffness on the Vibration Modes.

E_2 can clearly be tuned such that it influences mode 1 on the $[90]_8$ plate. Also, as expected mode 1 on the $[0]_8$ plate now matches the expected value. A 5th order polynomial was therefore fitted to establish that E_2 should be set to 28.2MPa. (The rationale for using a 5th order polynomial can be seen in appendix C). The exercise was finally repeated to see the influence of G , assuming that G was the same in all planes, i.e. $G_{12} = G_{13} = G_{23}$, as shown in Figure 4-17.

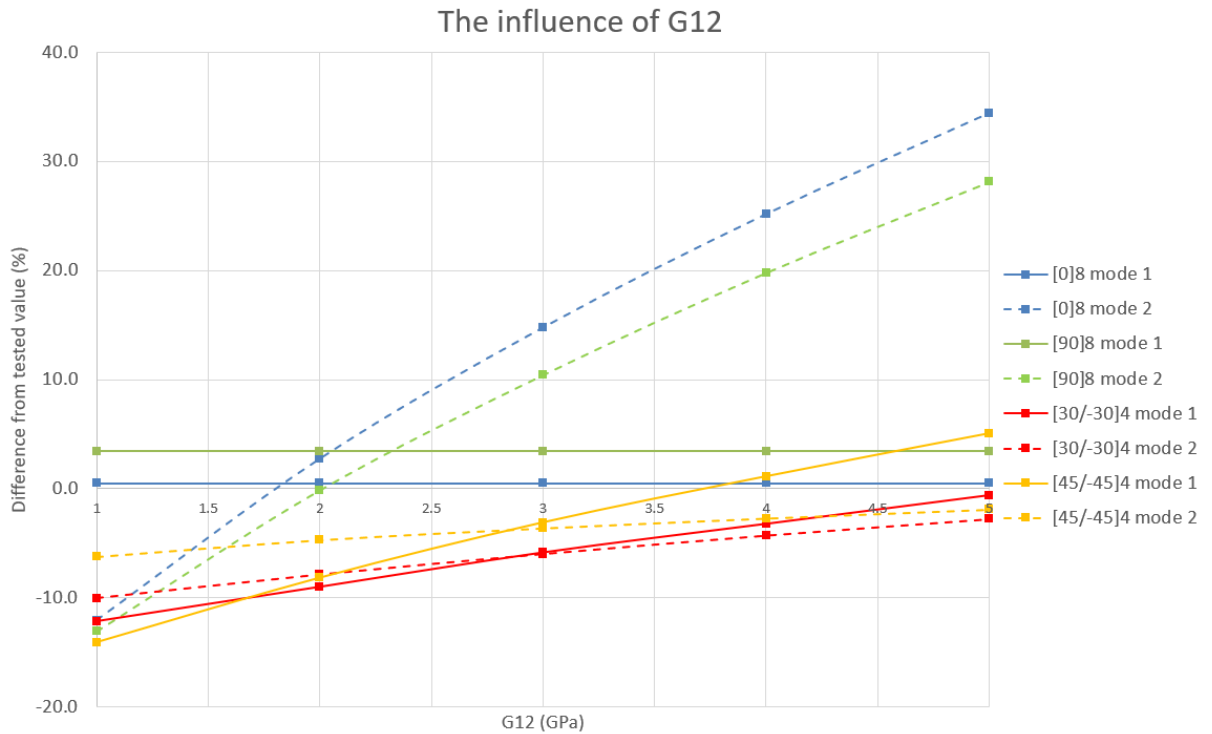


Figure 4-17: The Influence of Shear Stiffness on the Vibration Modes.

G influences all modes with the exception of the bending mode of the $[0]_8$ plate and the $[90]_8$ plates, in particular it influences the second (torsional) modes of these plates.

By adjusting the shear modulus to 2.4GPa to reduce the torsional modes and then scaling the longitudinal (23.0GPa) and lateral (23.0GPa) Young's Modulus reasonable correlation was obtained. (Note that in order to simplify the FE analysis E_1 and E_2 were assumed to be the same so that when the part is laid up the 90 degree plies are the same as the 0 degree plies, thus a value of 23GPa was used).

4.6.1 Correlation of Higher Order Mode Shapes

In order to gain further confidence in the material properties, it was decided to correlate higher order mode shapes from the FE model to those from the mechanical testing.

The comparison of these can be seen in Figure 4-18.

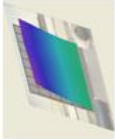
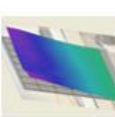
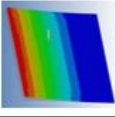
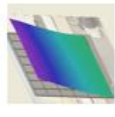
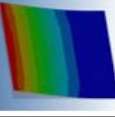
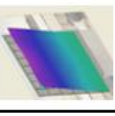
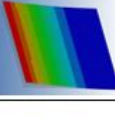
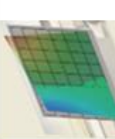
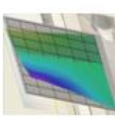
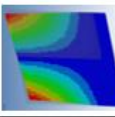
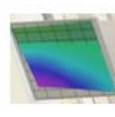
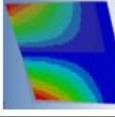
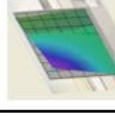
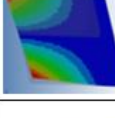
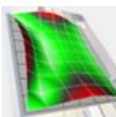
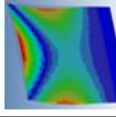
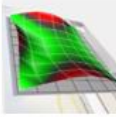
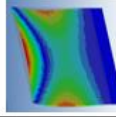
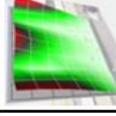
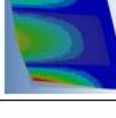
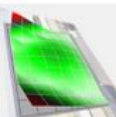
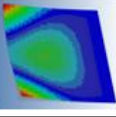
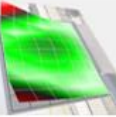
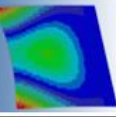
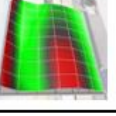
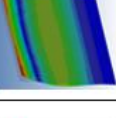
Lay-up											
[0]8			[30/-30]4			[45/-45]4			[90]8		
Test	FE model	Δ (%)	Test	FE model	Δ (%)	Test	FE model	Δ (%)	Test	FE model	Δ (%)
Mode 1											
	Ave. Freq (Hz) 41.7±0.8 3.6	2.6	38±0.8 4.8	35.3	-7.1	35.4±0.8 0.9	32.1	-9.3	45.8±0.8 4.8	42.8	-6.6
Mode 2											
	Ave. Freq (Hz) 61.4±0.8 0.9	8.8	105.7±0.8 2.4	95.5	-9.6	110.9±0.8 4.7	102.1	-7.9	67.2±0.8 1.6	66.8	-0.6
Mode 3											
	Ave. Freq (Hz) 272.4±0.8 6.3	-14.1	226.6±0.8 9.8	208.6	-7.9	197.4±0.8 7.1	186.8	-5.4	264.9±0.8 3.3	234.0	-11.7
Mode 4											
	Ave. Freq (Hz) 305.2±0.8 14.0	-12.2	288.6±0.8 9.1	254.3	-11.9	294.3±0.8 1.8	255.0	-13.4	293.2±0.8 14.1	268.0	-8.6

Figure 4-18: Correlation of the First Four Mode Shapes using the Scaled Material Properties.

As can be seen, although the first two modes are of primary interest in this study, modes 3 and 4 also correlate. The highest deviation seen from the predicted modes from the FEA models, when compared to the physical testing using the LDV, is 14.1% (mode 3 of the [0]₈ lay-up). This gives increased confidence in the scaled material properties.

4.6.2 Summary of Correlation Through Scaling of Material Properties

To improve material correlation further and for future studies, it is suggested that that this could be automated (using DoE) rather than doing this manually. The first two modes for all the lay-ups tested, have now been shown to have an error less than 10% as can be seen in Table 4-3;

		[0] ₈		[30/-30] ₄		[45/-45] ₄		[90] ₈	
		Mode 1	Mode 2	Mode 1	Mode 2	Mode 1	Mode 2	Mode 1	Mode 2
Test 1	(Hz)	37.5	60.9	32.8	103.1	35.9	106.3	40.6	67.2
Test 2		43.8	62.5	42.2	107.8	34.4	110.9	46.9	65.6
Test 3		43.8	60.9	39.1	106.3	35.9	115.6	50.0	68.8
Average		41.7	61.4	38.0	105.7	35.4	110.9	45.8	67.2
stdev		3.6	0.9	4.8	2.4	0.9	4.7	4.8	1.6
Predicted with SUPPLIER material properties		68.1	173.1	68.1	173.1	68.0	173.0	68.1	173.1
Difference		26.4	111.7	30.1	67.4	32.6	62.1	22.3	105.9
Difference	(%)	63.3	181.8	79.1	63.7	92.1	55.9	48.6	157.6
Predicted with SCALED material properties	(Hz)	42.8	66.8	35.3	95.5	32.1	102.1	42.8	66.8
Difference		1.1	5.4	-2.7	-10.2	-3.3	-8.8	-3.0	-0.4
Difference	(%)	2.6	8.7	-7.2	-9.7	-9.3	-8.0	-6.6	-0.6

Table 4-3: Vibration Testing: FE Models v Experimental (Scaled & Supplier Material Properties).

A summary of the scaled properties (correlated to the vibrational tests) can be compared to those established from the material supplier (in section 4.1.6) in Table 4-4;

Property	Supplier Values	Scaled Values
Youngs Modulus in the longitudinal direction (E_1)	58.0 GPa	23.0GPa
Youngs Modulus in the lateral direction (E_2)	58.0 GPa	23.0GPa
Poissons Ratio (Nu_12)	0.04	0.04
Inplane shear modulus (G_12)	27.9 GPa	2.4GPa
Transverse shear modulus (1-Z plane)(G_1z)	27.9 GPa	2.4 GPa
Transverse shear modulus (2-Z plane)(G_2z)	27.9 GPa	2.4 GPa
Mass density (Rho)	1.51 g/cm ³	1.60 g/cm ³
Allowable longitudinal stress in tension (Xt)	641.2 MPa	641.2 MPa
Allowable longitudinal stress in compression (Xc)	605.8 MPa	605.8 MPa
Allowable transverse stress in tension (Yt)	641.2 MPa	641.2 MPa
Allowable transverse stress in compression (Yc)	605.8 MPa	605.8 MPa
Allowable inplane shear stress (S)	59 MPa	59 MPa
THICKNESS /laminate	0.26mm	0.26mm

Table 4-4: Material Properties: Supplier v Scaled.

4.7 Ultrasonic Testing

Due to the differences in the material properties seen in section 4.6, it was decided to confirm these using Non-Destructive Testing (NDT) techniques. In particular, the use of Ultrasonic testing techniques were used in which the speed of sound was measured in a sample to establish the elastic material properties.

4.7.1 Principles of Ultrasonic Testing

Ultrasonic testing (UT) is used as a form of NDT. A sound wave is transmitted through an object and the received signal is used to determine the internal structure of the object that the sound wave has travelled through. Historically, through transmission signals were used as shown in Figure 4-19, however this technique has generally been replaced by backwall reflected signals as shown in Figure 4-20 due the fact that only

access to one side of the component is needed and the transmitter and receiver can be housed within the same hardware. However, through transmission signals do have the advantage that there is less attenuation of the sound energy (100). The size of the received signal and the timing of any received signal can be used to determine the size and the location of any defects within the structure. In addition to this, this technique can be used to measure the speed of sound through a sample of a known geometry, or if the speed of sound in the material being tested is known, then the component thickness can be calculated from the time of flight of the signal. The signal is generally transmitted at a frequency in the range of 0.1 to 20 MHz by piezo crystals that transform electric energy into mechanical energy in the form of sound pulses (101). Importantly, for this application, the speed of sound is related to the stiffness of the material and hence the material properties can be estimated.

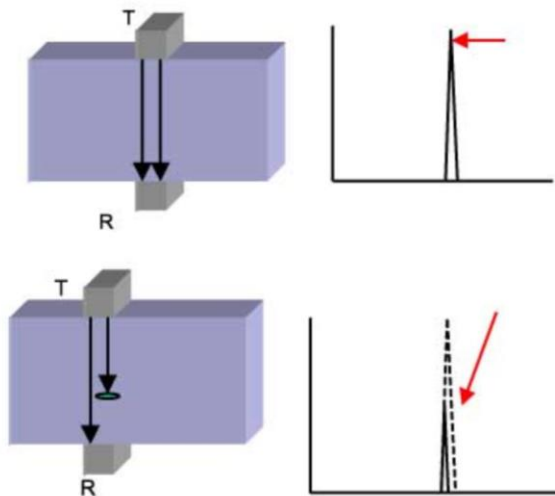


Figure 4-19: The Principle of Ultra-Sonic Testing (Through Transmission Signal) (100).

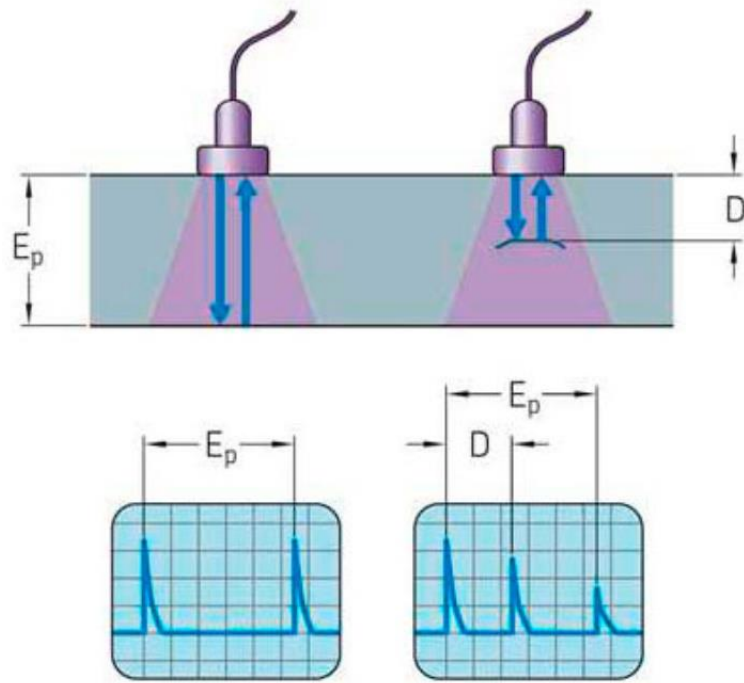


Figure 4-20: The Principle of Ultra-Sonic Testing (Backwall Reflected Signal) (101).

When sound propagates through a structure, this is mainly through a mixture of Longitudinal Waves and Shear Waves which are related to the motion of the particles within the structure that is transmitting the signal. In Longitudinal waves, the particle motion is parallel to the direction of the wave as it propagates. In shear waves, the particle motion is perpendicular to the direction of the wave as it propagates (102). These waves can be seen in Figure 4-21.

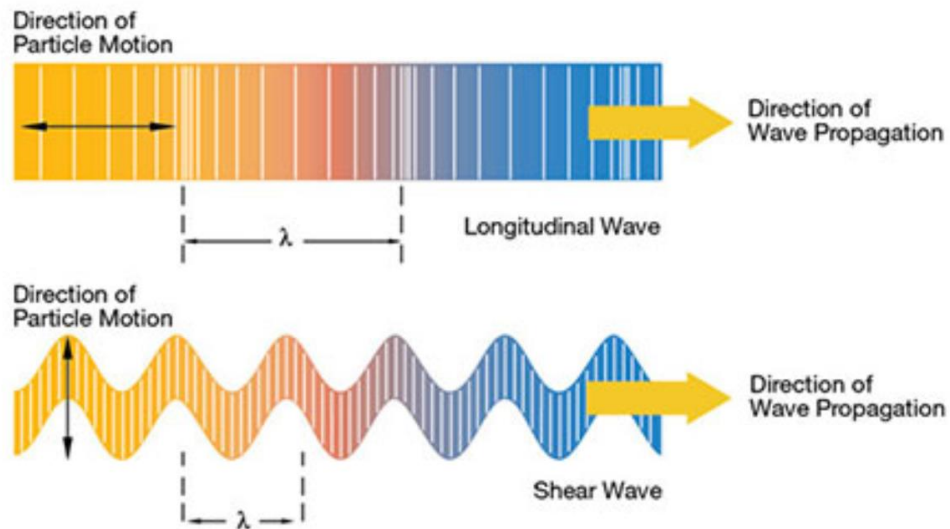


Figure 4-21: Longitudinal & Shear Waves (102).

4.7.2 Test Procedure and Sample Manufacture

In order to measure the speed of sound through a CFRP sample, the procedure outlined by Olympus (103) was carried out. Although this procedure has been developed to establish elastic properties for isotropic materials, it has been shown by the work of Grimberg et al (104) and Munoz et al (105) to be suitable for determining the elastic properties of CFRPs. This gave approximate material properties to see if the values established in section 4.6 were correct. The speed of sound of the Longitudinal Wave was measured through the sample in-plane with the plies that are laid-up in the CFRP and out of plane. An A-scan was carried out. The sample manufactured can be seen in Figure 4-22.

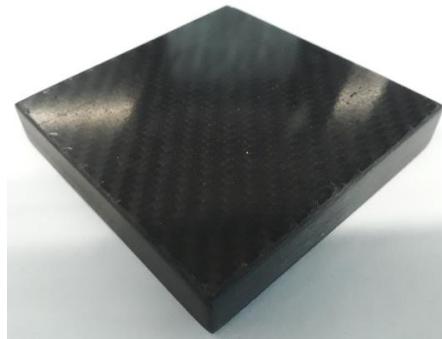


Figure 4-22: CFRP Sample for Ultrasonic Testing.

This sample was 90mm by 90mm by 16.5mm thick and was laid up with all plies at 0 degrees. This enabled the material properties to be established out of plane. A small section that was 19mm x 19mm was cut out of the corner of this sample so that the in-plane measurements could be taken and the material properties in this plane could be established. The part was manufactured from E722, 3k, 2x2 twill 280g/m² using a pressure of 3.0T for 60 minutes and cured at 120°C as specified by the material supplier data sheet (95).

When measurements were taken out of plane, a through transmission measurement needed to be taken due to the scattering of the signal as it passed through the sample. When taking measurements in-plane, the backwall reflected signal could be measured as this was subjected to less scattering. The experiment can be seen in Figure 4-23. 1MHz transducers were used.



Figure 4-23: Measuring of the Speed of Sound. Through Transmission (L) & Backwall Reflection (R).

It has been shown by the work of Paterson et al (106) that the velocity of the shear wave is 0.56 of the velocity of the longitudinal wave. Dorado and Moore (107) found an average of 0.52, whilst Grimberg et al (104) found a value of 0.69.

4.7.3 Errors

As discussed in section 4.5.3, the resolution of the measurement equipment needs to be considered in any results that are presented. In addition, how these uncertainties combine in any resultant calculations should be considered. The following errors are therefore considered and presented in the results in this section.

1. The sampling frequency of the Omniscan that was used to take the measurements was 100MHz. The error for the time measurements was thus $(1/100)/2 = \pm 5 \mu\text{s}$. From this the maximum/minimum velocities could be calculated and then applied to Equation 4-18 to Equation 4-22 to get maximum/minimum values for the material properties.
2. Note that Poisson's ratio is based on the ratio of the Transverse to Longitudinal speed of the waves travelling through the material. This resulted in no error since this value was assumed based on secondary research (as discussed in section 4.7.2).

4.7.4 Results

From the testing carried out, the average speed of sound of the longitudinal wave in plane was measured to be $5739 \pm 0.2 \text{ ms}^{-1}$. The transverse wave (taking an average of the literature quoted of 0.59 of the longitudinal wave) would result in a shear wave speed of $3386 \pm 0.1 \text{ ms}^{-1}$. The material properties could then be calculated using the following formulas taken from Olympus (103).

$$\text{Poisson's ratio } (\nu) = \frac{1 - 2\left(\frac{V_T}{V_L}\right)^2}{2 - 2\left(\frac{V_T}{V_L}\right)^2}$$

Where V_T = Shear (transverse) velocity and V_L = Longitudinal velocity

Equation 4-17: Poisson's Ratio Calculation.

This resulted in a value of 0.23 for the Poissons ratio in-plane.

$$\text{Young's Modulus (E)} = \frac{V_L^2 \rho (1+\nu)(1-2\nu)}{1-\nu}$$

Where V_L = Longitudinal velocity, ρ = Density and ν = Poisson's Ratio

Equation 4-18: Young's Modulus Calculation.

This resulted in a value of 45.2 ± 0.003 GPa for the in plane stiffness at zero degrees.

$$\text{Shear Modulus} = V_T^2 \rho$$

Equation 4-19: Shear Modulus Calculation.

This resulted in a value of 18.3 ± 0.001 GPa for the shear stiffness in plane.

The speed of sound of the longitudinal wave out of plane was measured to be $1941 \pm 0.02 \text{ ms}^{-1}$. The transverse wave (taking an average of the literature quoted of 0.59 of the longitudinal wave) would therefore result in a shear wave speed of $1145 \pm 0.01 \text{ ms}^{-1}$. As such the material properties could be calculated using the following formulas taken from Olympus (103);

$$\text{Poisson's ratio } (\nu) = \frac{1 - 2\left(\frac{V_T}{V_L}\right)^2}{2 - 2\left(\frac{V_T}{V_L}\right)^2}$$

Where V_T = Shear (transverse) velocity and V_L = Longitudinal velocity

Equation 4-20: Poisson's Ratio Calculation.

This resulted in a value of 0.23 for the Poisson's ratio out of plane.

$$\text{Young's Modulus (E)} = \frac{V_L^2 \rho (1+\nu)(1-2\nu)}{1-\nu}$$

Where V_L = Longitudinal velocity, ρ = Density and ν = Poisson's Ratio

Equation 4-21: Young's Modulus Calculation.

This resulted in a value of 5.2 ± 0.0001 GPa for the out of plane stiffness at zero degrees.

$$\text{Shear Modulus} = V_T^2 \rho$$

Equation 4-22: Shear Modulus Calculation.

This resulted in a value of 2.1 ± 0.00004 GPa for the shear stiffness out of plane.

The values of 5739ms^{-1} for the in-plane speed of sound and 1145 ms^{-1} for the out of plane speed of sound agree with the work by Yan et al. (108) which has been reproduced in Figure 4-24. In this work, the sample was 15mm thick and consisted of Uni-Directional (UD) plies stacked repeatedly at 0° , 45° , 90° and -45° with 1 layer of 2 x 2 twill on the top and the bottom surface to hold the UD plies together. However, the number of plies and the material properties are not stated. Despite this, the correlation between the UT results and the results obtained by Yan et al gives some confidence to the results obtained.

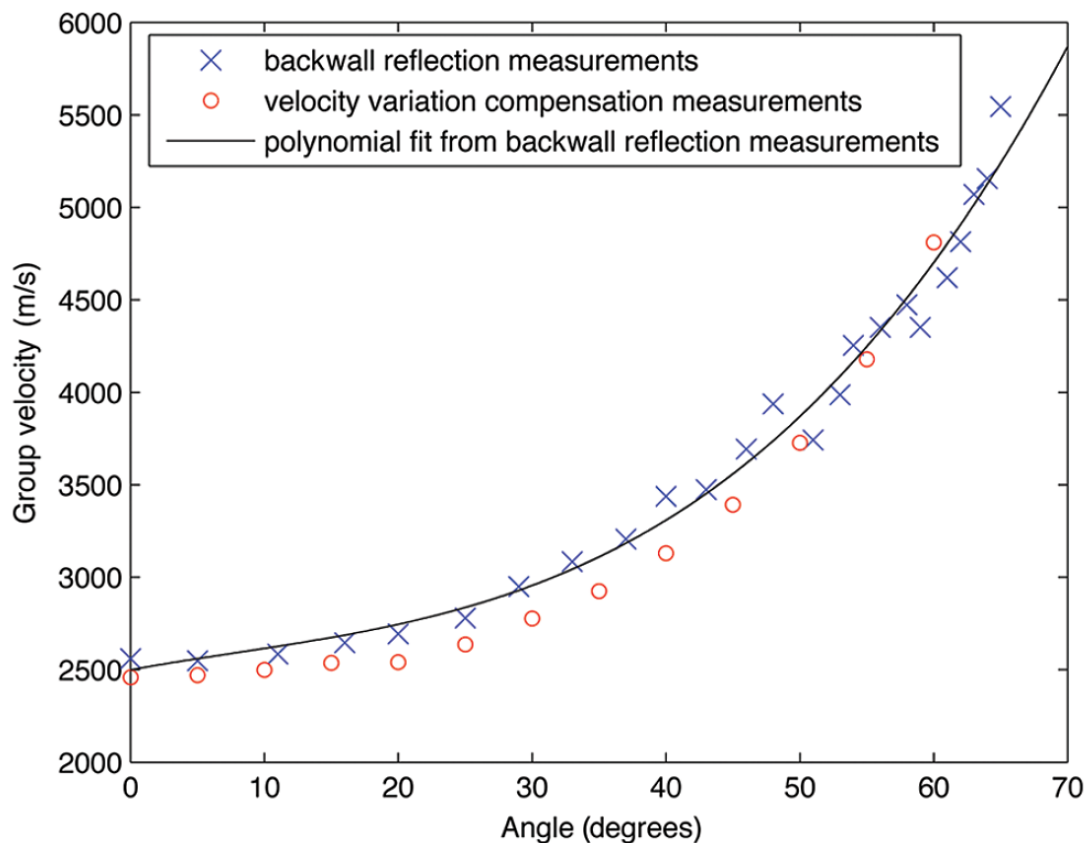


Figure 4-24: The Change of the Speed of Sound Through CFRP with Angle (108).

The material properties obtained from the UT testing can be seen in Table 4-5 where they can be compared to the Supplier values and the scaled values that have been obtained through correlation.

Property	Supplier Values	Scaled Values	UT Values
Youngs Modulus in the longitudinal direction (E_1)	58.0 GPa	23.0GPa	45.2GPa
Youngs Modulus in the lateral direction (E_2)	58.0 GPa	23.0GPa	45.2GPa
Poissons Ratio (Nu_12)	0.04	0.04	0.23
Inplane shear modulus (G_12)	27.9 GPa	2.4GPa	18.3GPa
Transverse shear modulus (1-Z plane)(G_1z)	27.9 GPa	2.4 GPa	2.1GPa
Transverse shear modulus (2-Z plane)(G_2z)	27.9 GPa	2.4 GPa	2.1GPa
Mass density (Rho)	1.51 g/cm ³	1.60 g/cm ³	
Allowable longitudinal stress in tension (Xt)	641.2 MPa	641.2 MPa	
Allowable longitudinal stress in compression (Xc)	605.8 MPa	605.8 MPa	
Allowable transverse stress in tension (Yt)	641.2 MPa	641.2 MPa	
Allowable transverse stress in compression (Yc)	605.8 MPa	605.8 MPa	
Allowable inplane shear stress (S)	59 MPa	59 MPa	
THICKNESS /laminate	0.26mm	0.26mm	

Table 4-5: Material Properties: Supplier v Scaled v UT.

Note that there is no out of plane direct stiffness (E_3) in the FE models that has been calculated as 5.2GPa using the UT methodology. This the reason that FE models that are based on classic laminate theory are inaccurate for relatively thick laminates as any three-dimensional effects such as normal deformations are neglected (109). Since the CFRP plys in this work are thin, these can be neglected has been demonstrated by the good correlation that can be achieved. This is supported by NASA (110) who compared various theories against the exact solution to calculate the deflection of a structure of varying thickness parameters (span/height). This is shown in Figure 4-25 which shows that as this ratio approaches 25/1, Classical laminated plate theory shows close approximation to the exact solution. Given that the ratio of span/height of the board is ~40/1 classical laminate theory can be used to model the structure with a high level of confidence in the results.

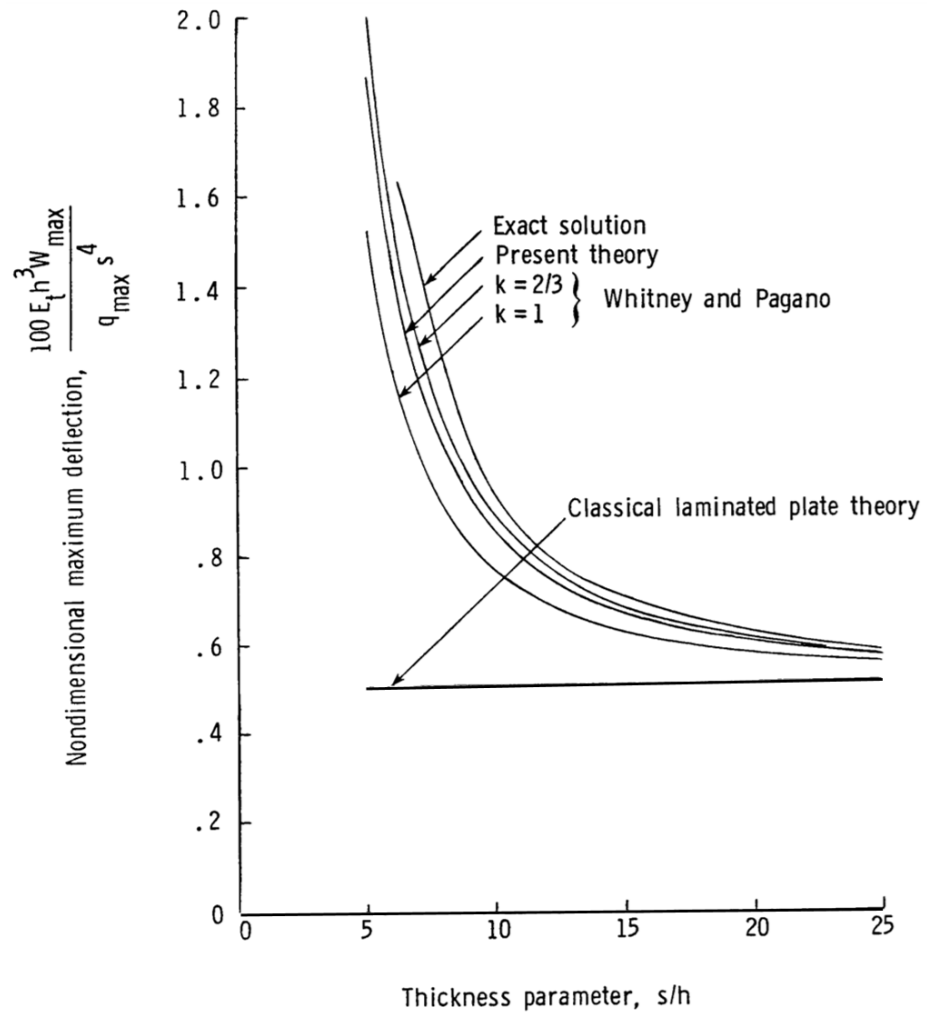


Figure 4-25: Classical Laminated Plate Theory Accuracy with Span/Height (110).

For completeness, the FE models of the vibration of the CFRP plates that was discussed in section 4.5 were re-run with the material properties obtained from the UT testing as shown in Table 4-5. The results of this can be seen in Table 4-6.

		[0]8		[30/-30]4		[45/-45]4		[90]8	
		Mode 1	Mode 2	Mode 1	Mode 2	Mode 1	Mode 2	Mode 1	Mode 2
Test 1	(Hz)	37.5	60.9	32.8	103.1	35.9	106.3	40.6	67.2
Test 2		43.8	62.5	42.2	107.8	34.4	110.9	46.9	65.6
Test 3		43.8	60.9	39.1	106.3	35.9	115.6	50.0	68.8
Average		41.7	61.4	38.0	105.7	35.4	110.9	45.8	67.2
stdev		3.6	0.9	4.8	2.4	0.9	4.7	4.8	1.6
Predicted with SUPPLIER material properties		68.1	173.1	68.1	173.1	68.0	173.0	68.1	173.1
Difference		26.4	111.7	30.1	67.4	32.6	62.1	22.3	105.9
Difference	(%)	63.3	181.8	79.1	63.7	92.1	55.9	48.6	157.6
Predicted with SCALED material properties	(Hz)	42.8	66.8	35.3	95.5	32.1	102.1	42.8	66.8
Difference		1.1	5.4	-2.7	-10.2	-3.3	-8.8	-3.0	-0.4
Difference	(%)	2.6	8.7	-7.2	-9.7	-9.3	-8.0	-6.6	-0.6
Predicted with UT material properties	(Hz)	60.0	140.0	59.4	140.0	59.4	140.0	59.7	140.0
Difference		18.3	78.6	21.4	34.3	24.0	29.1	13.9	72.8
Difference	(%)	43.9	127.9	56.2	32.4	67.8	26.2	30.3	108.3

Table 4-6: Vibration Testing: FE Models v Experimental (Scaled, Supplier & UT Material Properties).

As can be seen in Table 4-6, the errors are smaller with the scaled material properties. Due to this, the subsequent FE models in this work use these values for the material properties.

4.8 Conclusions

As has been shown, better material correlation can easily be obtained using vibration testing and simply scaling the material properties that have been provided by the materials supplier, without the need for costly UT testing, (although these confirmed the magnitude of the values being used). It has been shown that the material properties that influence the vibrational modes and the stiffness of the material are E_1 , E_2 and G . As such, these material properties can be taken from the material supplier and then scaled to achieve better correlation. By utilising a flat plate where the mode shapes can clearly be seen, the material properties that influence these modes shapes can be identified and scaled independently. It should be noted at this point that this has been

achieved via the LDV (which is a piece of Hardware that is prohibitively expensive to most SMEs). However, the same approach could be achieved using piezo transducers.

The other material properties that are needed for the numerical models (such as the allowable stresses in various directions) are used as supplied by the materials suppliers to check that the failure index is not exceeded on the final board. This is discussed in section 6.7 and compared to the values seen when using the material properties established in this chapter and related to the failure theories that have been discussed in section 4.3.

5 Board Analysis

The previous chapter considered the material that was used in the construction of the board. This chapter considers the overall system's response so that this response could be replicated in the proposed design methodology. For this study, two 'systems' were considered, which are aimed at expert riders and beginner riders.

5.1 Background

The 'Advisory' board by Blankforce is aimed at experienced riders as it delivers more 'pop' (this board is subsequently referred to as the 'Carbon' board in this work).

Blankforce state on their website (111) that the board has been developed to increase stiffness and to stop flex. The board is described as very firm, yet with control.

Blankforce describe the construction of this board as a wooden core surrounded with 100% carbon fibres, which have been orientated to produce a 'radical pop'. The first layer is made of unidirectional carbon in a generic +/- 45 degree form that crosses the board and resists the torsional mode. The following layers of carbon fibres are made of biaxial cloth laid-up at 45 degrees and of two 90 degree layers to reinforce the entire board.

In contrast, the 'Logic' is described by Blankforce (112) as an all-round freeride board with a comfortable ride, (this board is subsequently referred to as the 'Glass' board in this work). This board is described as having the right centre stiffness to the board for 'pop' and a better rider response. The thinner tip area offers shock absorption and better control. The Glass fibres are laid up at 45° as shown in Figure 5-1 to resist torsion.

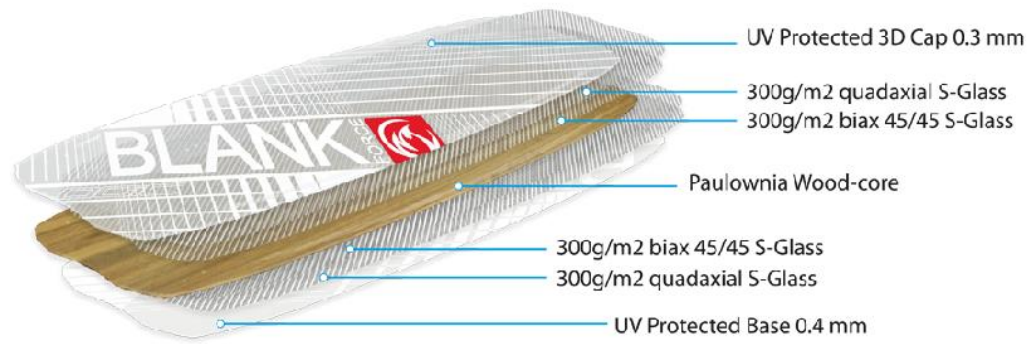


Figure 5-1: Lay-up Details of the Logic Board (112).

In terms of the materials used, Blankforce reportedly use Paulownia wooden cores laminated and orientated lengthways along the board (113). The Advisory board has more rocker than the Logic board meaning that the Logic board planes easier and earlier, however the Advisory board turns easier and releases easier from the water.

5.2 Anatomy of a Kiteboard

So that the descriptions of how the board was tested can be fully understood, Figure 5-2 shows a kiteboard and the regions described.

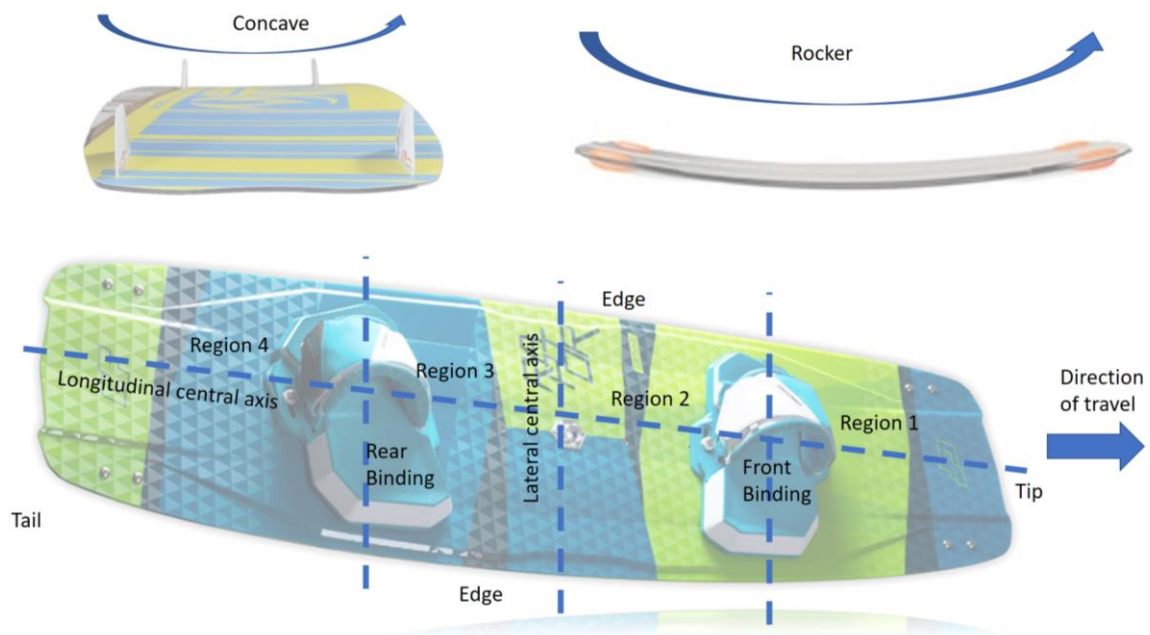


Figure 5-2: The Anatomy of a kiteboard (114) (115).

Region 1: The region of the board up to the centre of the front binding.

Region 2: The region of the board between section 1 and the lateral central axis.

Region 3: The region of the board between the lateral central axis and section 4.

Region 4: The region of the board beyond the centre of the rear binding.

Tip: The front of the board in the direction of travel.

Tail: The rear of the board in the direction of travel.

Edge: The sides of the board approximately parallel to the direction of travel.

Concave: The curvature of the board in the lateral direction.

Rocker: The curvature of the board in the longitudinal direction.

5.3 Component Testing

In order to test the kiteboard global characteristics, the methodology as used by Clifton (28) was modified. Clifton's work investigated the bending and torsional stiffness's of a Snowboard, hence this was modified for this work.

5.3.1 Bending

5.3.1.1 Test Procedure

In order to calculate the bending stiffness of the centre regions of the boards, testing was carried out as per Clifton (28), but with a few significant changes. The set-up used by Clifton can be seen in Figure 5-3.

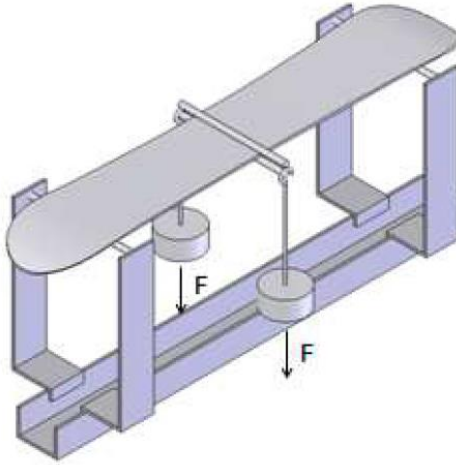


Figure 5-3: Bending Stiffness Arrangement for Regions 2 & 3 (28).

The rig used by Clifton had 20mm diameter rollers positioned 1500mm apart in order to support the board whilst 2 x 20mm rollers supported variable masses from 32kg to 50kg in the centre of the board, such that the bending stiffness could be calculated. This was repeated with 60lb (27.2kg) weights, whilst the board was supported on bars that were 25.28mm in diameter and 600mm apart. The set-up can be seen in Figure 5-4. This was set-up on the CMM machine, so that the position of the undeformed and the deformed boards could be measured easily.

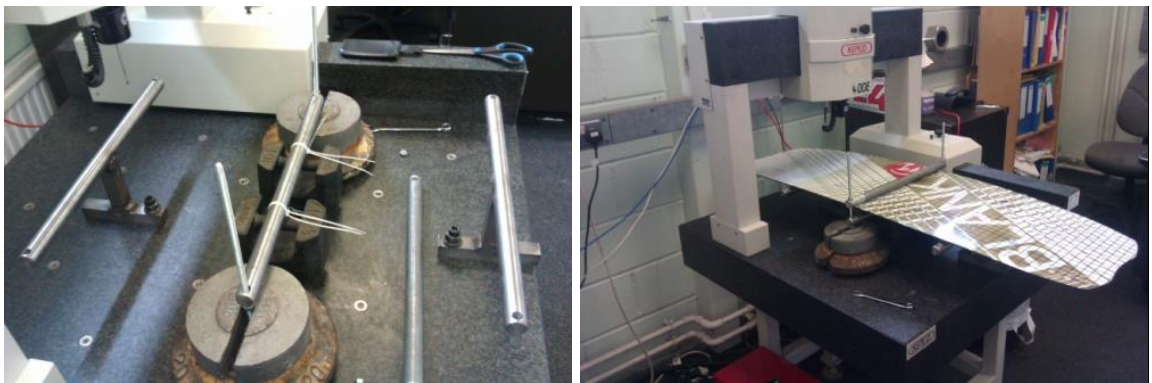


Figure 5-4: Measuring the Bending Stiffness (Regions 2 & 3).

In order to calculate the bending stiffness of the tip/tail regions of the boards, testing was carried out as per Clifton (28), but with a few significant changes. The set-up used by Clifton can be seen in Figure 5-5.

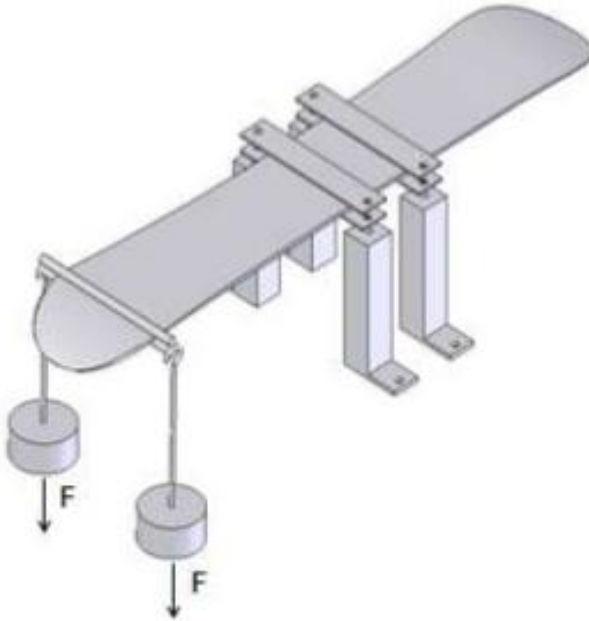


Figure 5-5: Bending Stiffness Arrangement for Regions 1 & 4 (28).

The rig used by Clifton had 40mm wide metal plates supporting the board whilst 22kg was suspended 100mm from the end of the board, such that the bending stiffness could be calculated. This was repeated with 40lb (18.1kg) weights, whilst the board was supported on bars that were 25.28mm in diameter and 300mm apart, whilst the weights were suspended 20mm from the end of the board. The set-up can be seen in Figure 5-6.



Figure 5-6: Measuring the Bending Stiffness (Regions 1 & 4).

5.3.1.2 Errors

As discussed in section 4.5.3, the resolution of the measurement equipment needs to be considered in any results that are presented. In addition, how these uncertainties combine in any resultant calculations should be considered. As such the following errors are considered and presented in the results in this section.

1. Despite the CMM being accurate to $\pm 0.001\text{mm}$ an error of $\pm 0.1\text{mm}$ was taken for the height calculation due to the textured nature of the board topsheets as suggested by Clifton (28). Thus, when the final position is subtracted from the initial position, this error is doubled to $\pm 0.2\text{mm}$. It should be noted at this point, that despite the CMM being used to obtain the deflections, this could be achieved by using simple DTI gauges.
2. In terms of position of the measurements, these were taken to be $\pm 0.5\text{mm}$.

5.3.1.3 Results

An average result was taken from either end of the board and it was assumed that the board was symmetrical about the lateral central axis. In addition, the errors discussed in section 5.3.1.2 have been added to the results. The results from the testing can be seen in Figure 5-7 and Figure 5-8. Notice that both boards display the same trend with

interruptions to the stiffness of the boards at approximately 100mm and 160mm which coincide with the locations of the steel inserts for the bindings.

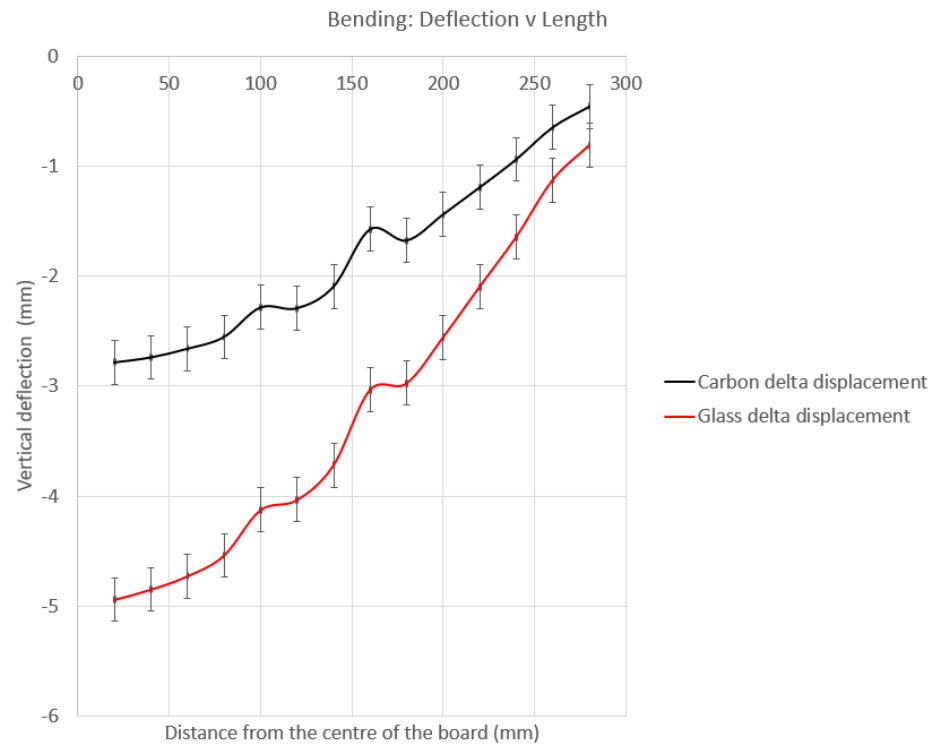


Figure 5-7: Deflections in Regions 2 & 3 (Centre) due to Bending.

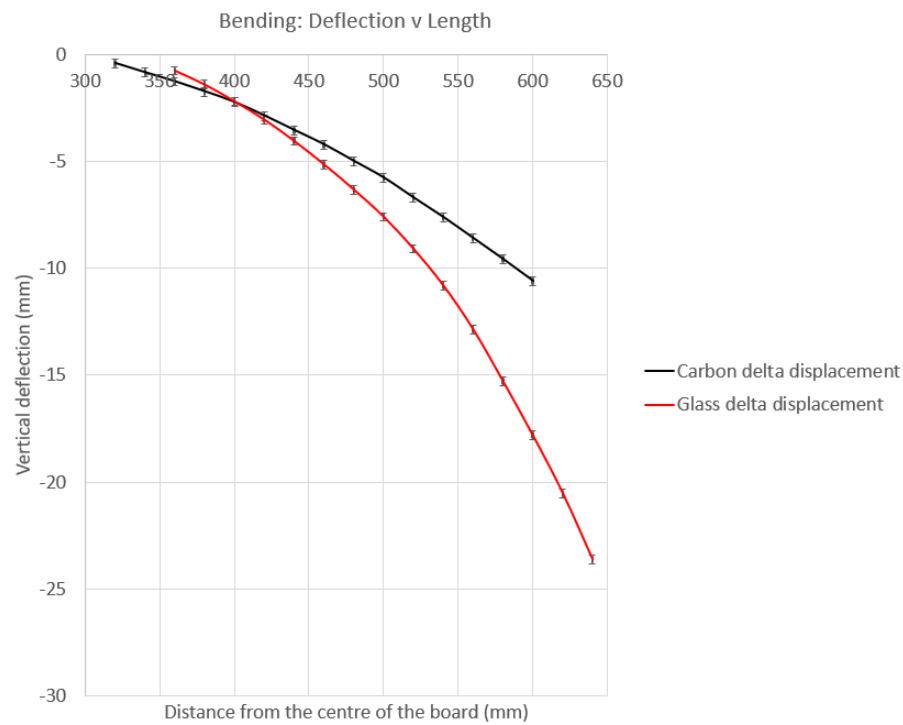


Figure 5-8: Deflections in Regions 1 & 4 (Tip/Tail) due to Bending.

The bending stiffness could then be assessed using the simple theory of elastic bending as shown by the work of Clifton, that is to say;

$$\frac{M}{I} = \frac{\sigma}{y} = \frac{E}{R}$$

Equation 5-1: The Simple Theory of Elastic Bending.

Thus, the flexural stiffness, EI , could be calculated by multiplying the moment, M , by the radius of curvature, R , i.e. $EI = MR$, and the radius of curvature is given by $R \approx \frac{L^2}{8\delta}$, as shown in Figure 5-9.

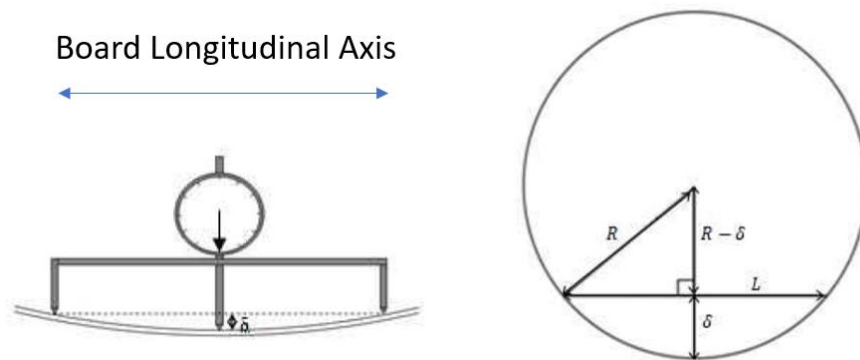


Figure 5-9: Calculating the Radius of Curvature (28).

Using Pythagoras,

$$R^2 = (R - \delta)^2 + \left(\frac{L}{2}\right)^2$$

$$R^2 = R^2 - 2R\delta + \delta^2 + \frac{L^2}{4}$$

$$8R\delta = 4\delta^2 + L^2$$

$$\text{Since } \delta^2 \approx 0$$

$$R \approx \frac{L^2}{8\delta}$$

The use of the simple theory of elastic bending makes certain assumptions as summarised by Hearn (116), however it enables an approximate value of the bending stiffness to be easily calculated. Hearn lists these assumptions as;

1. The beam is initially straight and unstressed.
2. The material of the beam is perfectly homogeneous and isotropic.
3. The elastic limit is not exceeded.
4. Young's modulus for the material is the same in tension and compression.
5. Plane cross-sections remain plane before and after bending.
6. Every cross-section of the beam is symmetrical about the plane of bending.
7. There is no resultant force perpendicular to any cross-section.

Whilst the above technique enables EI to be calculated relatively easily, it does introduce certain errors.

- Obviously, this method assumes that the board has a constant stiffness over the measured distance L . Measurements were taken every 20mm, thus the curvature would be measured over 40mm. Consider a section of the board that contains an infinitely stiff section which is less than 40mm, such as the red section shown in Figure 5-10.

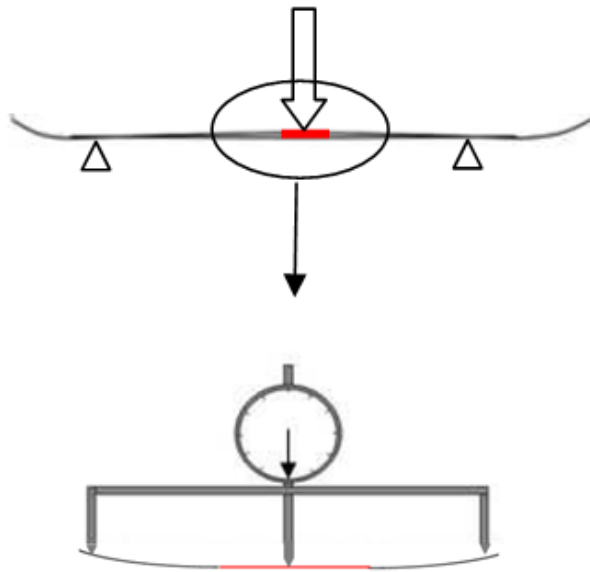


Figure 5-10: Demonstration of Average Stiffness Being Calculated.

Due to the fact that the measurement is taken over 40mm, this section appears to have a curvature and thus a finite stiffness, in reality, the red section is not curved at all due to the fact that it is infinitely stiff.

- Clifton assumed that the section being measured was initially straight and hence the stiffness of the board could be calculated from the curvature of the deformed board using ($EI = MR$), however, the undeformed boards are already curved due to their rocker. Clearly, using the technique used by Clifton would lead to significant errors when calculating the stiffness of a board with a large rocker such as a kiteboard. If the tips are considered, (supported as a cantilever) then the board would appear to have been deformed significantly more than it actually was from the loading due to the fact that it was already curved in its undeformed state. The stiffness of each board was measured in 2 regions, the tip/tail and the centre regions as outlined in Figure 5-4 and Figure 5-6 A representation of these measurements can be seen in Figure 5-11.

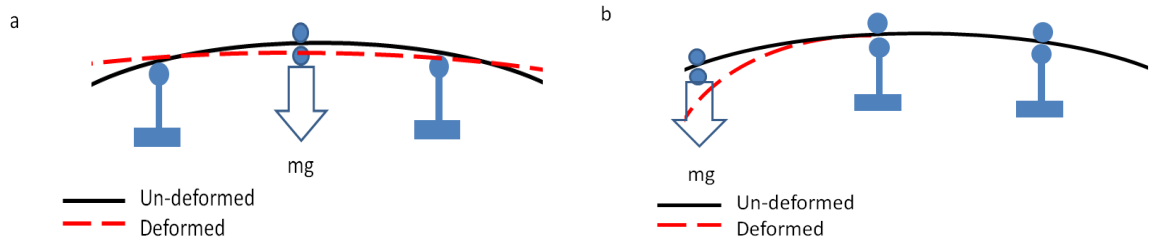


Figure 5-11: Measuring the Bending Stiffness of the Board (a) Centre region (b) Tip/Tail.

Hence when measuring regions 1 (tip) and 4 (tail) as shown in Figure 5-11 (b) the radius of curvature would decrease, and when measuring regions 2 and 3 (the centre) the radius of curvature would increase. However, since the boards already have curvature in both of these regions due to the rocker of the board, then the final curvature cannot simply be attributed to the moment being applied to the board.

- The alignment of the structure in the jig during testing has been shown by Truong, Brousseau and Desniens (117) to significantly influence the measured stiffness of similar structures, thus this would introduce errors which would propagate through the calculations.
- Due to the fact that a kitesurf board is slightly concaved, when this is clamped between two flat surfaces then the second moment of area, I , is reduced. This is significant since it is EI that is being measured. In order to minimise this, the clamping load was reduced until the board was just touching the clamping mechanism in 3 places, hence I would not be influenced. However, in all of the calculations it was assumed that if the board is clamped then there is no rotation about the lateral axis, this is clearly not the case if the clamping load is so small.

Due to the reasons discussed, rather than plotting EI along the length of the board, the deflection was simply plotted so that the FE model could be optimised against this.

To demonstrate the design methodology, the FE simulation results were optimised to replicate the performance of the Carbon board. The reason for this decision, is that the material being used may not be flexible enough to give the response of the glass board. If the methodology works for the carbon board then the same process could be achieved with the glass board, however more flexible materials would need to be used potentially.

5.3.2 Torsion

5.3.2.1 Test Procedure

In order to calculate the torsional stiffness of similar boards (GJ), the set-up in Figure 5-12 was used as developed by Clifton (28) where the board was clamped and two masses apply a torque – one simply pulling the board down and the second pulling the board up via a pulley system.

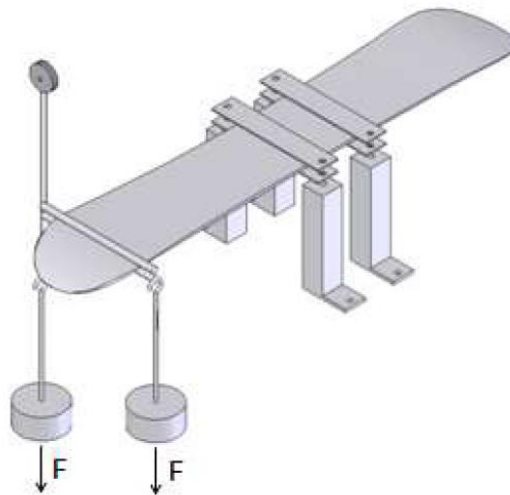


Figure 5-12: Torsional Stiffness Arrangement (28).

In order to measure the torsional stiffness in regions 1 and 4, a torque of $(9.07 \times 9.81 \times 0.225 \times 2) \text{ 40Nm}$ was applied to the board and measurements were taken every 20mm. The board was constrained at the front binding (300mm) and the lateral central axis and a torque was applied to the tip. This can be seen in Figure 5-13.



Figure 5-13: Measuring the Torsional Stiffness of the Boards.

In order to measure the torsional stiffness in sections 2 and 3, the board was constrained across the lateral central axis and the rear binding and then a torque was applied to the front binding.

5.3.2.2 Errors

As discussed in section 4.5.3, the resolution of the measurement equipment needs to be considered in any results that are presented. In addition, how these uncertainties combine in any resultant calculations should be considered. As such the following errors are considered and presented in the results in this section.

1. Despite the CMM being accurate to $\pm 0.001\text{mm}$ an error of $\pm 0.1\text{mm}$ was taken for the height calculation due to the textured nature of the board topsheets as suggested by Clifton (28).

2. The distance from the centre of the board to the edges where the measurements were taken was 140mm. This means that the error was $\tan^{-1}\frac{0.1}{140} = \pm 0.04^\circ$. Thus, when the final position is subtracted from the initial position, this error is doubled to $\pm 0.08^\circ$.
3. In terms of position of the measurements, these were taken to be $\pm 0.5\text{mm}$.

5.3.2.3 Results

The results from the testing can be seen in Figure 5-14 and Figure 5-15.

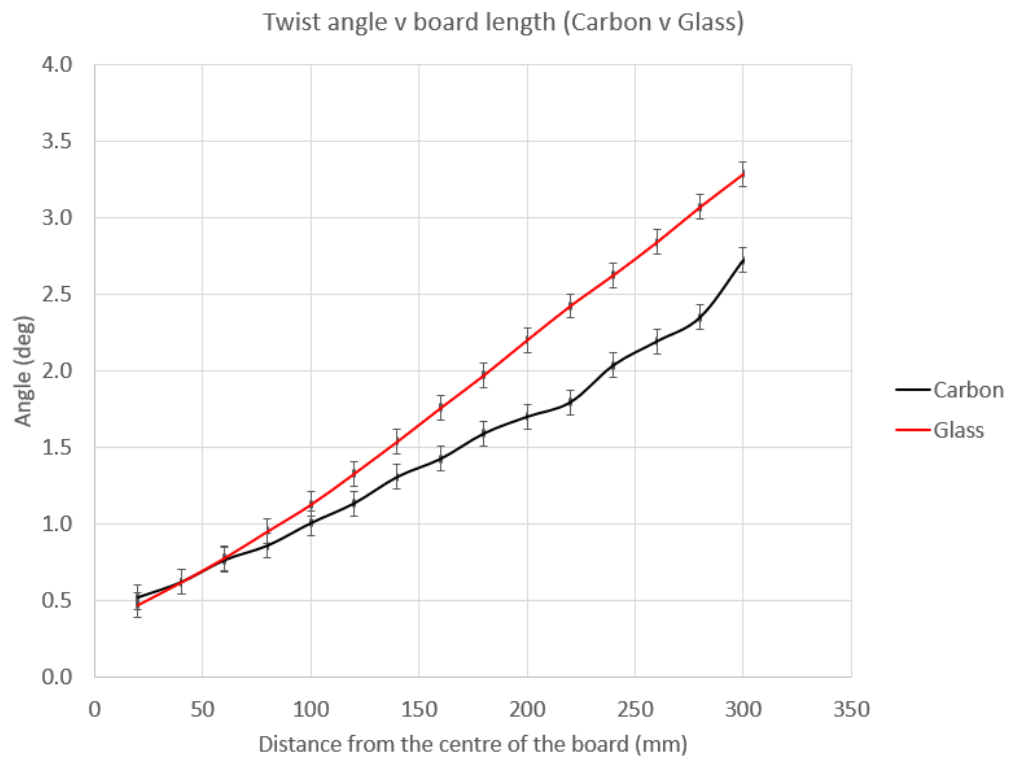


Figure 5-14: Angular Deflection in Sections 2 & 3 (Centre) due to Torsion.

Note that the stiffness of the boards varies fairly linearly over regions 2 and 3, and that the carbon board is stiffer as expected.

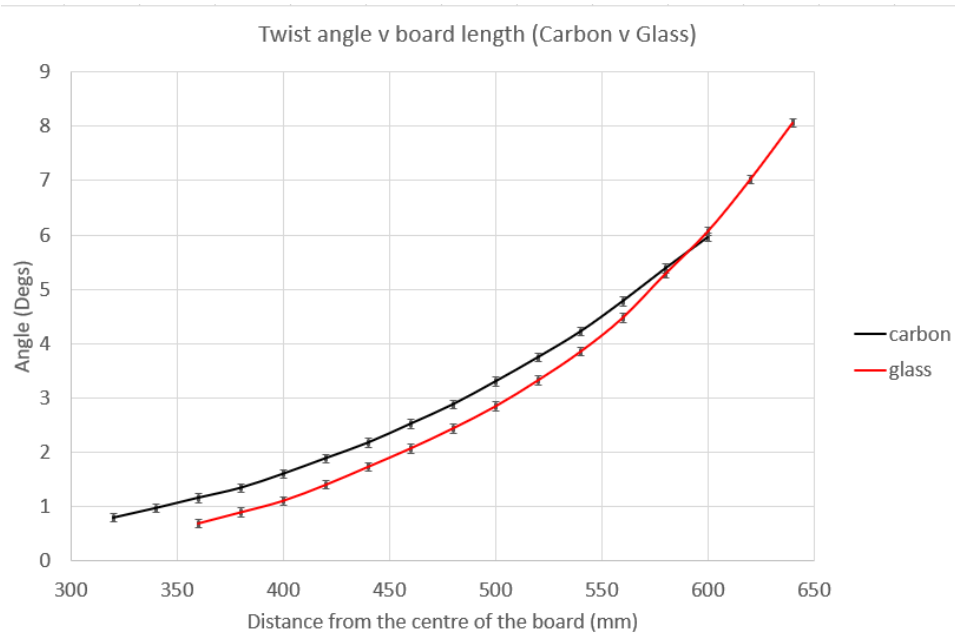


Figure 5-15: Angular Deflection Regions 1 & 4 (Tip/Tail) due to Torsion.

Notice in regions 1 and 4, the stiffness of the glass board and the carbon board are similar over a large proportion (400 – 550mm). The similar torsional stiffness in the central region is potentially explained by the fact that the glass board has an increased second moment of area over much of this region due to the wooden core as seen by the shaded region in Figure 5-16.



Figure 5-16: The Increased Depth in Regions 2 & 3 of the Glass Board.

The torsional stiffness would be reduced significantly at the edges of the board where the wooden core is reduced in section. However, this is not seen in this testing as the bar clamps the entire width of the board meaning that only the global torsional stiffness

is measured. Interestingly, this increased regional thickness is approximately where the bindings are placed, suggesting that a certain stiffness is required here for the rider to be able to transfer inputs into the board. Notice also, that the glass board has a reduced torsional stiffness at the extreme tip/tail also due to this reduction in section (past 550mm), meaning that this board would be more forgiving to the inexperienced rider. This method would need to be improved to see the increase in the flexibility of the board across the width of the board.

The torsional stiffness of the board along the length was not symmetrical about the lateral central axis as expected. Thus, an average of the torsional stiffness was calculated from either end of the board. Also, it was assumed that the board stiffness was symmetrical about the central longitudinal axis, and therefore that the deflection of the board was 0mm in the centre.

In order to calculate the torsional stiffness, the simple theory of elastic torsion could be used, that is to say;

$$\frac{T}{J} = \frac{\tau}{R} = \frac{G\theta}{L}$$

Equation 5-2: The Simple Theory of Elastic Torsion.

Thus, the torsional stiffness, GJ , is given by;

$$GJ = \frac{TL}{\theta}$$

Equation 5-3: Torsional Stiffness.

Where T is the applied torque, L is the length of the section being measured and θ is calculated as shown in Figure 5-17.

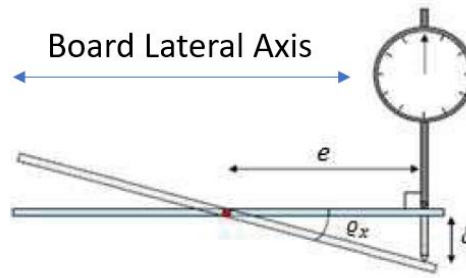


Figure 5-17: Calculating the Angle of Deflection (28).

The use of the simple theory of elastic torsion makes certain assumptions (116), as shown below, however it enables an approximate value of the torsional stiffness to be easily calculated.

1. The material is homogeneous.
2. The material is elastic.
3. The stress does not exceed the elastic limit.
4. It is developed for circular cross-sections and assumes these remain circular.
5. Cross-sections remain plane.
6. Cross-section rotate as if rigid.

However, for the same reasons, and to be consistent with the bending results, rather than plotting GJ along the length of the board, the angle of deflection was simply plotted so that the FE model could be optimised against this result. Again, as per the bending results, the carbon board will be used for correlation purposes as discussed in section 5.3.1.3.

5.3.3 Vibration

The vibrational response of the base boards needed to be assessed so that the response of the optimised board could be compared to these. It has been shown that “Computing natural frequencies and mode shapes helps to assess and understand the dynamic interaction between a component and its supporting structure” (96), however,

the board was optimised using the results from sections 5.3.1 and 5.3.2. The reason for this is as follows. Whilst the vibrational response of the board is important and can be used to compare the boards, it cannot be used to optimise the boards. Consider if the board were being optimised using the first bending mode as the constraint.

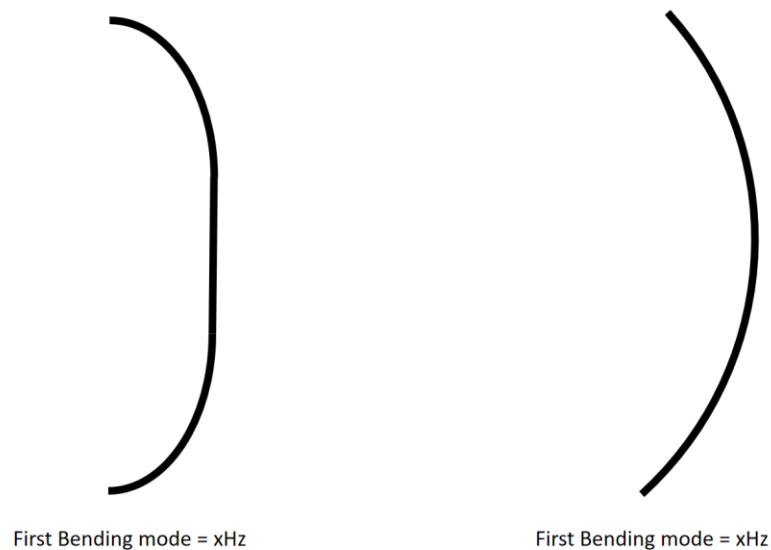


Figure 5-18: Two Boards with Bending Modes at the Same Frequency.

The baseline board shown on the left has the same frequency as the optimised board on the right, however, clearly the stiffness of the baseline board is greater in the centre of the board with more flexible tips, (as seen with the glass board in section 5.3.1). The optimised board would potentially be lighter as it has the same stiffness along the entire length of the board, however, clearly this stiffness gradient along the length of the board is different to the baseline board so the response felt by the rider would be different. As such, the stiffness along the length of the board needs to be optimised as well as the natural frequency in order to replicate the response felt by the rider. Note that it is still possible to correlate the material properties using the vibrational response

as in section 4.6 due to the fact that the stiffness of the plates made up in this section is consistent along the length of the plate.

5.3.3.1 Test Procedure



Figure 5-19: Vibrational Testing Set-up of the Boards.

As can be seen in Figure 5-19, the boards were simply hung on bungee cords using the holes for the fins in the corners of the boards, they were impacted with an actuator at the rear of the board and the response was measured using the LDV.

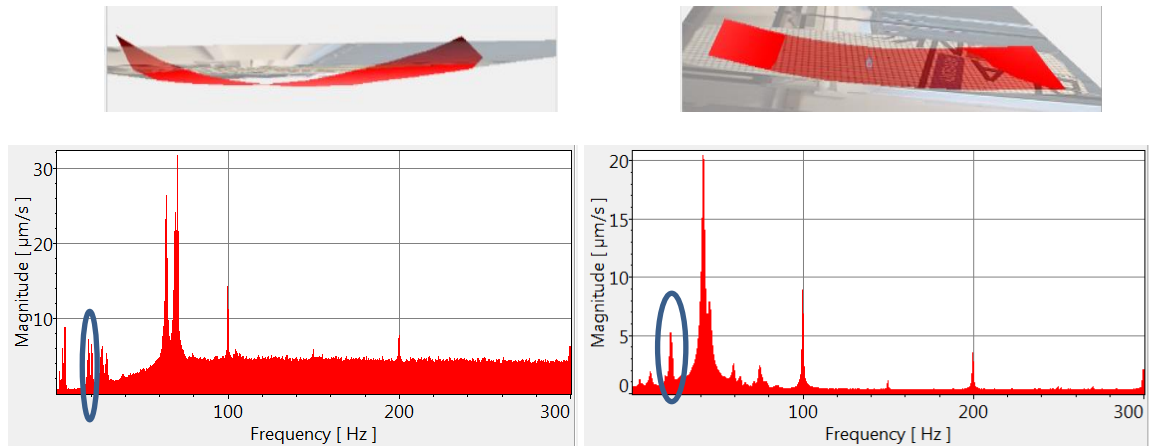
5.3.3.2 Errors

The errors in this section are assumed to be the same as those that were discussed in section 4.5.3 where vibration testing was carried out on CFRP samples when considering the material properties. These were as follows;

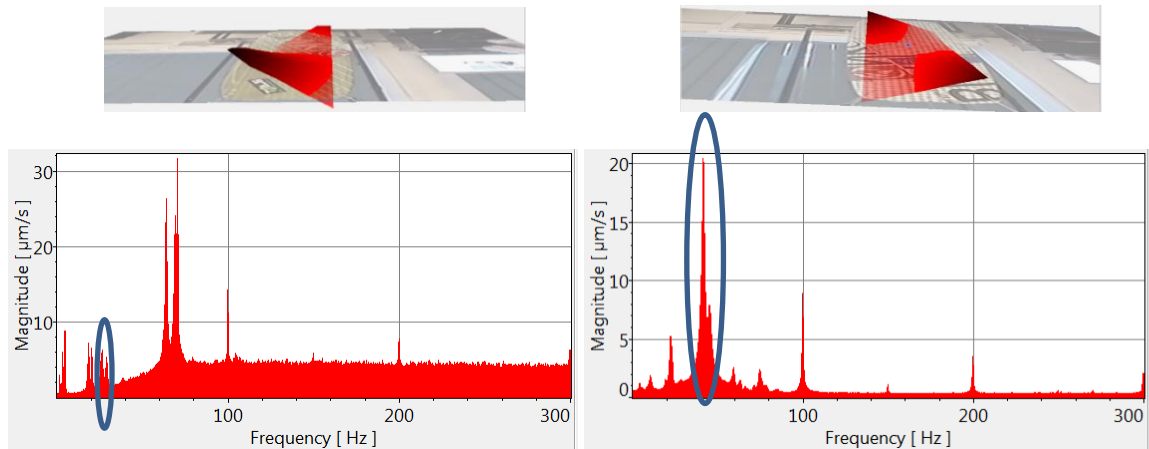
1. Error in the frequencies that were quoted by the LDV were $\pm 0.8\text{Hz}$.

5.3.3.3 Results

The results of the above testing can be seen in Figure 5-20.



Glass board first bending mode = $18.75 \pm 0.8\text{Hz}$ Carbon board first bending mode = $22.5 \pm 0.8\text{Hz}$



Glass first torsion mode = $26.9 \pm 0.8\text{Hz}$

Carbon first torsional mode = $45.3 \pm 0.8\text{Hz}$

Figure 5-20: Vibrational Test Results of the Boards.

The two boards had a very similar first bending mode. (Note that the modes seen before the first bending modes are rigid body modes due to the boundary conditions). However, the first torsional mode is higher for the carbon board suggesting that this is more important for advanced riders. This is backed up by the results seen in the torsional stiffness section which shows that the torsional stiffness of the carbon board is higher than the glass board, which would obviously relate to a higher torsional mode.

Note that the glass board has a 'double peak' for both the bending and the torsional modes. It was hypothesised that this was due to the fact that the board is essentially constructed of two structures with very different EI (Flexural stiffness) values. The edges/tip/tail of the board are relatively flexible, however the centre of the board has a significantly greater section due to the thickness of the core. This can be seen in Figure 5-21 (a reproduction of Figure 5-16, shown here for clarity) by the shaded region and then viewed from the side at the tip and along the edge of the board.

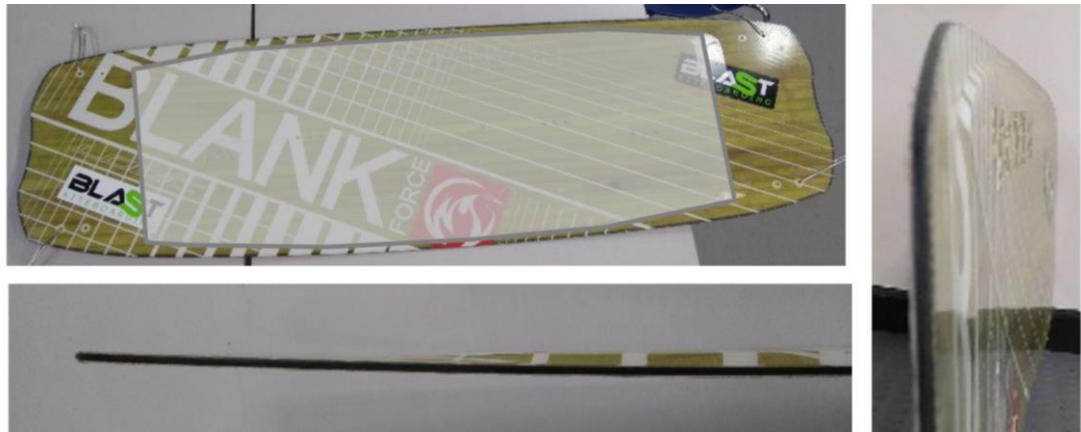


Figure 5-21: Changes in Section Along the Board.

As such these two structures will have very different responses and these will constructively/destructively interfere with each other leading to the complex system response (i.e. the double peak). Again, this confirms why optimising a board using the first bending mode should not be done as this would not be seen. In order to test this hypothesis, a similar structure was modelled in an FE environment, seen in Figure 5-22, to see if similar behaviour was observed.

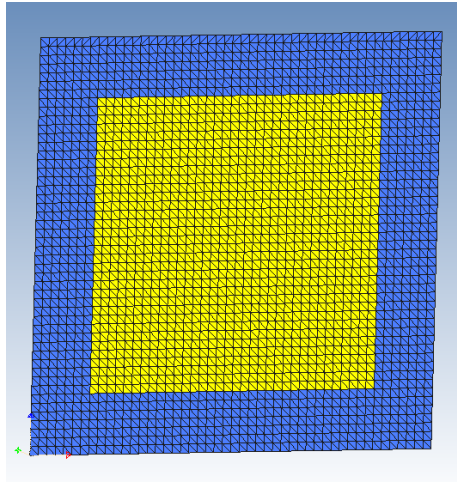


Figure 5-22: FE Model with Changes in Section.

The blue surround was 3mm thick CFRP whilst the yellow inner section was 8mm thick CFRP, thus mimicking the flexible tips seen at the edges of the glass board. The modal response of this plate was assessed in a free-free environment. Two first order bending modes were observed as seen in Figure 5-23.



First order bending at 46.0Hz



First order bending seen at 49.7Hz

Figure 5-23: Two Modes Seen in the Plate with Changes in Section.

The magnitude of these two modes is irrelevant, and would clearly be related to the geometry of the section, the boundary conditions and the material properties.

However, what is important, is that two first order bending modes were seen just as in

the glass board due to the very different section thicknesses across the board. This confirms why the boards cannot be optimised using the bending/torsional modes, as well as explaining why the ‘double peaks’ are seen in Figure 5-20.

5.4 Mock Field Testing

In order to understand how these boards will behave in ‘the field’ it was proposed that a more realistic loading mechanism needed to be applied to the boards. This would give some insight into their vibrational behaviour during use, and in addition to this, it is used in section 6.7 to ensure that the proposed boards would not fail during use using the failure criteria discussed in section 4.3. The following testing procedure was carried out on the boards to see their response.

5.4.1 Test Procedure

It was important that this procedure was repeatable with the variables associated with real life field testing removed (when data acquisition equipment is simply attached to the boards and these are ridden). As such, this procedure is based upon the work carried out by Fuss (118), who was trying to define a standardised test procedure as “Standardised laboratory conditions are required for comparing different boards”. This work looked at the response of snowboards, using an LDV when loaded as if they were being ridden and then impacted by discontinuities in the riding surface. This set-up can be seen in Figure 5-24 which has been reproduced from the work by Fuss.

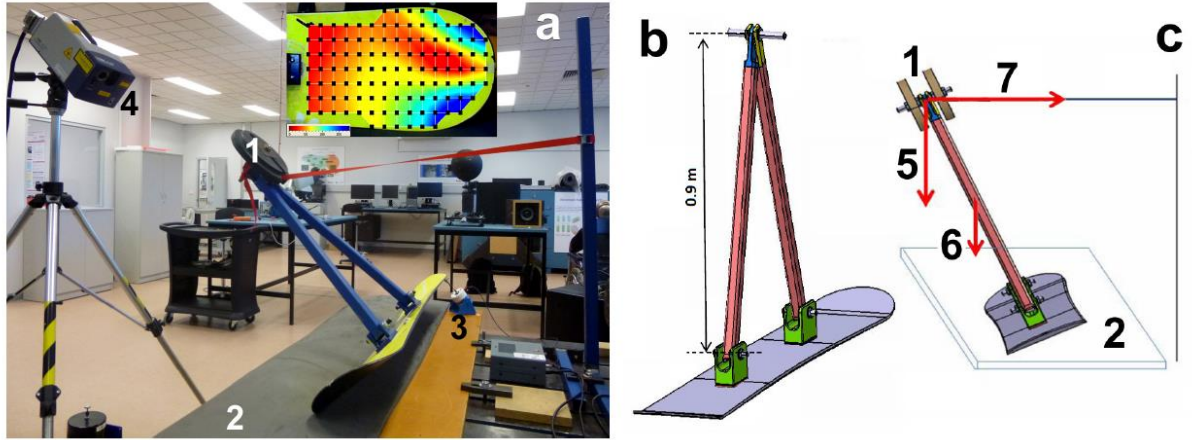


Figure 5-24: Mock Field Testing of a Snowboard (118).

Figure 5-24 shows the following;

- a) Experimental Set-up;
 - 1. Load.
 - 2. Foam mat.
 - 3. Shaker.
 - 4. Laser vibrometer.
 Top insert: Scanning grid projected on the shovel.
- b) Frame mounted on the board.
- c) Experimental Set-up;
 - 5. Weight of load.
 - 6. Weight of frame.
 - 7. Belt force.

There are clear differences that need to be considered when applying this to represent the loading on a kitesurf board. An example of a loaded kitesurf board can be seen in Figure 5-25.



Figure 5-25: Loading on a kiteboard in 'The Field' (119).

Note the following adjustments needed to be made to the methodology used in (118) ;

1. The kiteboard sits lower in the water. In the study by Fuss (118) he was looking at the response to snowboards on hard snow and therefore, a 20mm thick mat with a density of 34 kg/m^3 and 0.12 MPa stress at 0.5 strain was used. To more accurately represent how a kiteboard is loaded it was supported by a mat with a larger thickness and a lower density.
2. The edge of the board is not parallel to the surface of the water, i.e. the tail of the board sits lower in the water than the tip of the board.
3. The loading on the board is clearly not symmetrical. The rider has more weight on their rear foot. This is obviously the cause of 2. This is backed up with previous studies (120) which have shown that the “front knee joint was extended whereas the rear knee joint was flexed”.

Work by Brorsson and Lundgren (120) looked into the body position of a kitesurfer whilst riding. They showed that the angle of the board relative to the water averages 38 degrees when cutting upwind as seen Figure 5-26 (shown as 40.7 degrees in the picture), however, when preparing for take-off to do a trick this angle reduced to 20 degrees.

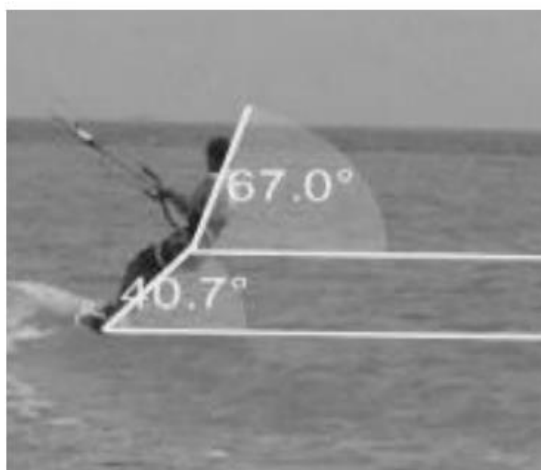


Figure 5-26: Body Position Whilst Kitesurfing (120).

Brorsson and Lundgren (120) also showed that the force generated by the kite averaged 5.1N/kg and peaked at 13.4N/kg just before a jump. Therefore, a 75kg rider on an expert board would transfer 1005N through the harness with a board angle of 20 degrees relative to the water, whilst the force due to gravity from the weight of the rider would be $75 \times 9.81 = 736\text{N}$. Other work by Sala (21) has looked at the relationship between the magnitude and the angle of the force generated by the kite as seen in Figure 5-27.

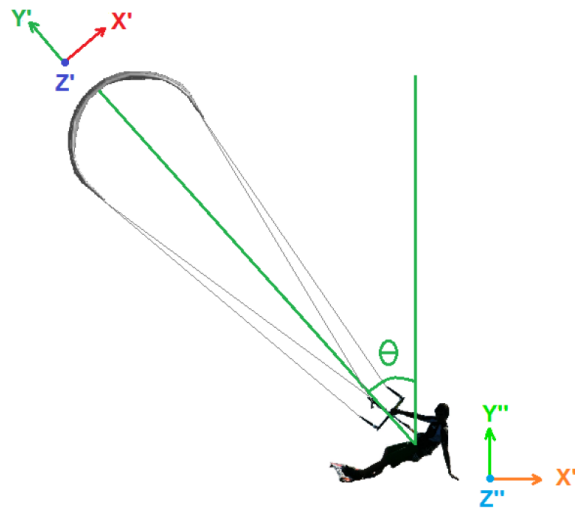


Figure 5-27: Position of the Kite Whilst Kitesurfing (21).

This study concluded that the maximum force is generated when θ is 90° . In terms of the angle that the board should be placed relative to the wind when viewed in plan view. It has been shown that whilst it is possible to cut 40° upwind, with standard equipment, generally 20° is more realistic (121), thus the board was placed 20 degrees upwind.

This means that the following loading is exerted on a kiteboard by an advanced rider (assuming that the centre of gravity is at the waist of the rider and the harness pull also is applied to the centre of gravity of the rider).

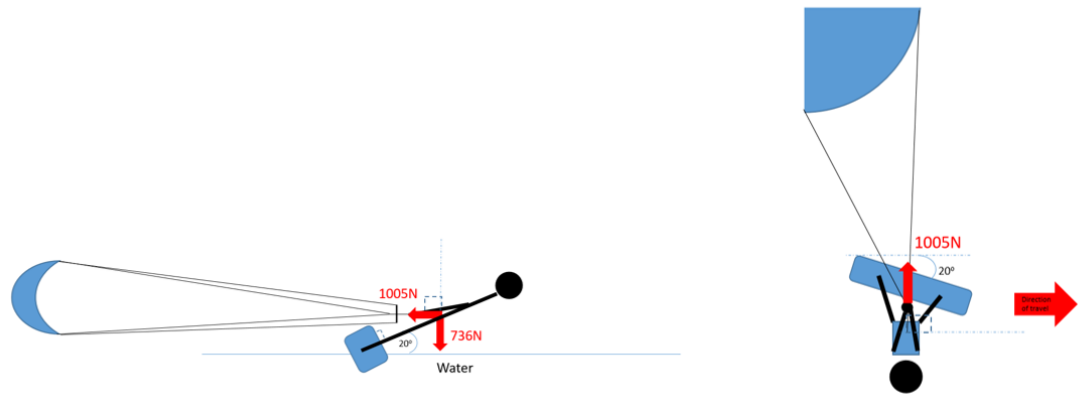


Figure 5-28: Assumed Board Position Whilst Kitesurfing: Side View (L) & Plan View (R).

The 'mock field' test can be seen in Figure 5-29.



Figure 5-29: Mock Field Testing of a kiteboard.

The set-up was as follows;

1. A jig was manufactured that connected Aluminium mounting blocks [1] to the mounting points for the footstraps on the board. These blocks were then attached to two members [2] that joined together at an apex [3] which represents the centre of gravity of the rider. These members were 0.9m long.

2. The apex was then connected to a ratchet strap [4] which was connected to the back wall of the lab. The ratchet strap was horizontal relative to the floor, thus mimicking the kite being at 90 degrees as seen by Sala (21), and the centre of the board was aligned with the position where the strap meets the rear wall.
3. The board was positioned resting on some low-density foam [5] and represented the board being in the water such that the board is submerged diagonally from corner to corner as seen in Figure 5-25. This foam was positioned so that it is initially 20 degrees to the back wall of the lab when viewed in plan view (before it is loaded by the board) as suggested by (121), thus representing the board heading upwind.
4. The ratchet strap was tightened so that the board (once loaded) is positioned so that it was 20 degrees from vertical, as suggested by Brorsson and Lundgren (120).
5. A weight of 79.3kg [6] was hung on the apex, thus creating a vertical downwards force, as generated by the rider self-weight, and the tension in the lines generated from the kite.
6. The back of the board was then impacted with an impactor [7], representing the excitation of the board due to the water, and the response of the board was measured using the LDV [8].

The methodology described allows the board response to be measured, when the board is loaded in a way similar to the loading typically seen 'in the field', i.e. in conditions that 'mock' field conditions. This will allow different boards to be loaded in a controlled repeatable fashion, which mimics how the board would be loaded when being used. This allows the response of different boards to be compared. This also allows the response of the boards in this environment to be compared to the bending stiffness, torsional stiffness and the vibrational response of these boards.

The following should be noted in terms of this procedure;

- In reality the water cannot transmit significant shear loads, although obviously there will be a small drag force. The shear loads transmitted by the foam would be significantly more.
- The fins, foot pads and handle are not on the board during this testing, it is expected that these would have a small influence on the results seen, although according to Valsecchi (14), this is not significant.

5.4.2 Errors

The errors in this section are assumed to be the same as those that were discussed in section 4.5.3 where vibration testing was carried out on CFRP samples when considering the material properties and section 5.3.3.2 where vibration testing was carried out on the board when it was hung on bungee cords. These were as follows;

1. Error in the frequencies that were quoted by the LDV were $\pm 0.8\text{Hz}$.

5.4.3 Results

5.4.3.1 Glass Board

The results for the glass board can be seen in Figure 5-30.

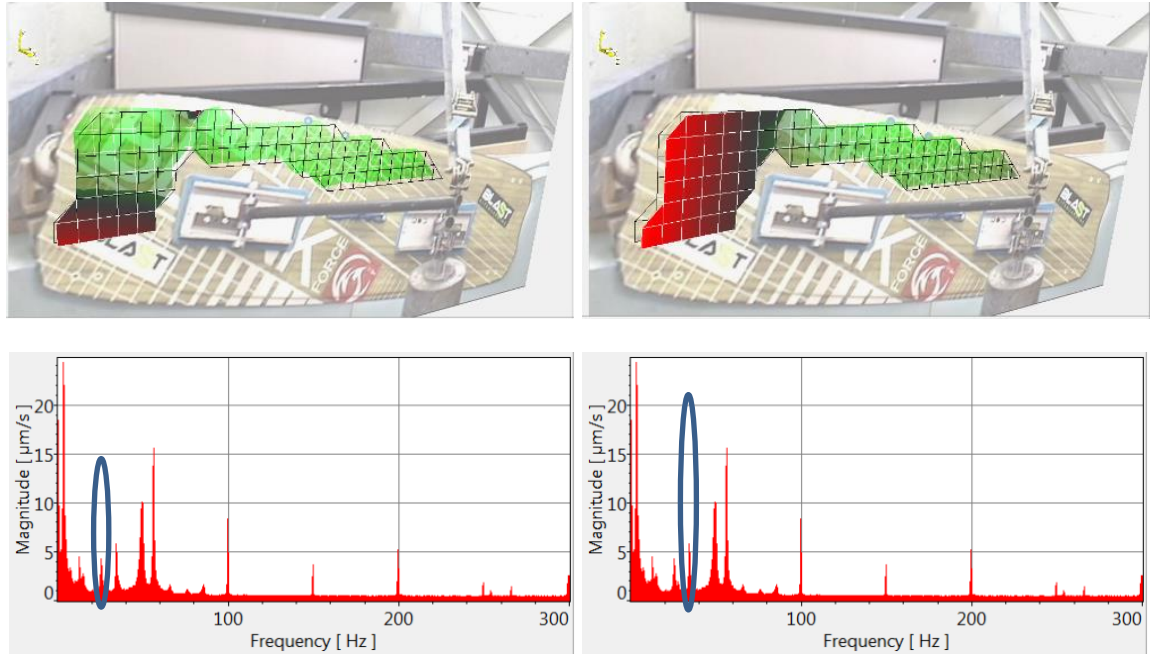


Figure 5-30: The Vibrational Response of the Glass Board Subjected to the Mock Field Test.

The first mode is a lateral mode at $50.0 \pm 0.8\text{Hz}$, whilst the second mode is a longitudinal mode at $56.6 \pm 0.8\text{Hz}$.

5.4.3.2 Carbon Board

The results for the carbon board can be seen in Figure 5-31.

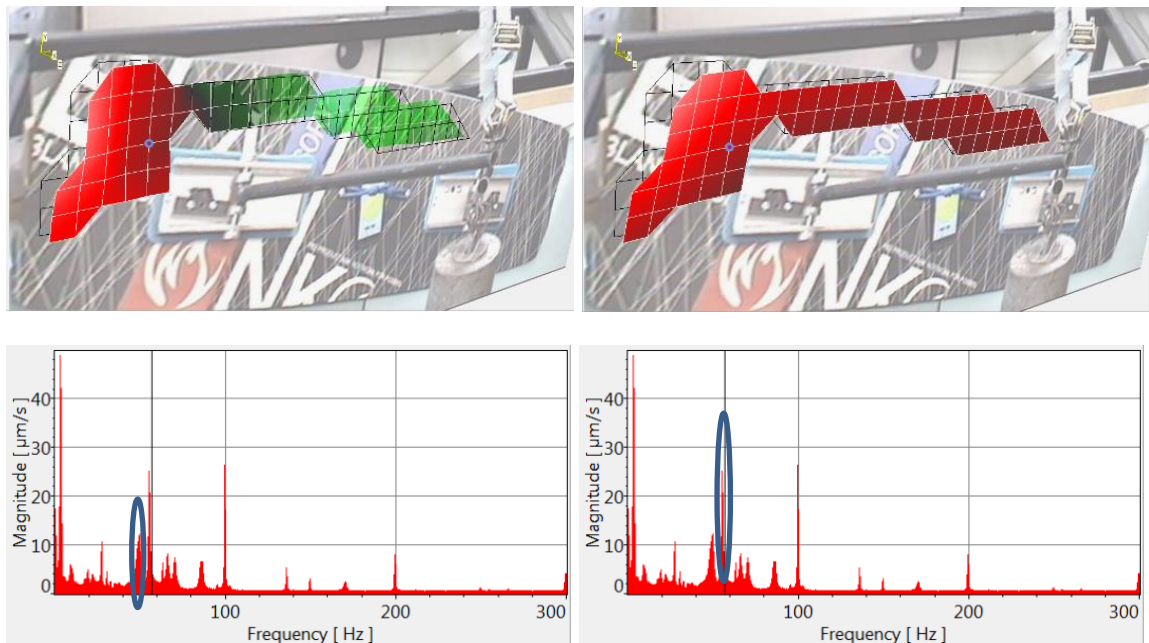


Figure 5-31: The Vibrational Response of the Carbon Board Subjected to the Mock Field Test.

The first mode is now a longitudinal mode at $50 \pm 0.8\text{Hz}$, whilst the second mode is a lateral mode at $55.6 \pm 0.8\text{Hz}$. The magnitude of the first mode has not changed, however due to the different stiffness distribution of the board, the first mode has switched from a torsional mode (as seen in the glass board), to a bending mode.

5.5 Conclusions

The system response has been assessed at both a component level, by looking at the bending, torsional and vibrational response of the boards and at a 'mock' field level.

The bending stiffness of the two boards has been shown to be linear in the centre of the board with the Carbon board being stiffer as expected. The bending stiffness of the boards across the binding locations is similar, however the bending stiffness of the glass board significantly reduces as the depth of the wooden core lessens at the tip/tail. It has also been shown that the torsional stiffness of the Carbon board is greater than the Glass board in the centre regions and the torsional stiffnesses of the two boards are then similar where the footpads are attached. The torsional stiffness of the Glass board is significantly less at the tip and the tail, again as the depth of the wooden core lessens.

The vibrational response of both boards has been assessed. The Carbon board has a higher first bending mode and first torsional mode, compared to the Glass board, although the order of these modes remains consistent. The modal response of the Glass board shows 'double peaks'. This has been shown to be due to the construction of the board. This is essentially two structures (a thick centre region and a thinner outer region). A methodology has then been proposed to mimic how the board is

loaded 'in the field'. This shows that the first mode for the Carbon board just prior to jumping would be a bending mode, whilst for the Glass board this would be a torsional mode. The mock testing is also used in section 6.7 to ensure that the proposed board would not fail during use.

6 Design of an Alternative Board using the Proposed Methodology

To demonstrate the proposed design methodology, a board was optimised that had the same structural characteristics as the carbon board that was tested in section 5. The reason for the decision to design a board comparable to the carbon board (as opposed to the glass board) is that the material being used may not be flexible enough to give the desired response of the glass board. Due to the fact that the design methodology works for the carbon board then the same process could be achieved with the glass board, however more flexible materials would potentially be needed.

6.1 The Rocker/Concave/Outline

The geometry (the rocker, concave and outline) of the proposed structure was kept identical to the original carbon board by taking measurements using the CMM. This ensured that any differences in the structural performance of the board would not be due to geometric changes. Instead, the response of the board was tuned simply by altering the lay-up of the plies of CFRP and by altering the thickness of the core.

6.2 The Wooden Core

As discussed in section 1.1, the cores of these boards are made up of a wooden structure to increase the second moment of area. It has also been shown by Wang et al (122) that the face sheets provide major contribution to the bending stiffness while the core provides the majority of the shear stiffness of sandwich structures. It can thus be concluded, that the material of the optimised board should be identical to the original carbon board. The thickness of this core will influence the second moment of area and the polar second moment of area and thus the bending and the torsional stiffness'. It was proposed that a standard Paulownia core could be obtained from a supplier such

as junksupply.com (123), this could then be machined to the thickness' suggested by the optimisation software. This core, prior to machining, can be seen in Figure 6-1.



Figure 6-1: Paulownia Wood Core (123).

The wood is supplied with a thickness of 10mm. This could then be profiled during the design of the board based on the results of the FE optimisation analysis. The material properties of the wood in the FE model were as follows, which was taken from the work by Kaymakci et al (124), The Wood Database (125) and Green et al (126).

$$\text{Density} = 150\text{kg/m}^3$$

$$\text{Young's Modulus (E}_L\text{)} = 3.88\text{GPa}$$

$$\text{Shear Modulus} = 0.2\text{GPa (3.88 x 0.054)}$$

$$\text{Poissons ratio} = 0.229$$

The Young's Modulus of this material is quoted along the grain. Values for transverse Young's Modulus are not readily available according to Niklas (127) however he has shown that a reasonable approximation is that $E_T = 0.06 E_L$. It was thus assumed that the transverse Youngs Modulus (E_T) was 0.2GPa.

The wood could then cut so that the outline of the board matched the outline of the benchmark board such that this outline would not have any influence on the change of the structural performance. It is proposed that the grain of the wood should run at 0 degrees (along the length of the board), as if the wood were to be placed at anything other than this, it would result in an asymmetric board.

This core was used in the manufacture of the scale boards in the verification study in section 7.

6.3 Building an FE Model of the Board

For this type of structure, the work by Grant (128) showed that a mesh was considered to be converged with an element size of 1 inch (25.4mm). When determining the deflection of the board in section 5.3.1 and section 5.3.2 the board was measured every 20mm, therefore the boards were meshed with 20mm elements, ensuring that the nodes corresponded with the measurement points taken on the board so that correlation could be achieved. It has also been shown by Stefani and Siebert (129) that 2D elements can accurately model the behaviour this type of structure, i.e. 3D elements are not necessary. As such, the CMM data of the bottom of the board (rather than the mid-surface) was taken to build surface models of the boards as can be seen in Figure 6-2. In terms of the element order, the optimisation results are based on the size of the elements. Therefore, for a certain number of nodes, it is better to have smaller first order elements, as opposed to larger second order elements. Due to this, first order elements were used.



Figure 6-2: Outline of the Glass (Black) & the Carbon (Blue) Boards.

The CAD models of the board loaded as in section 5.3.1 and section 5.3.2 can be seen in Figure 6-3 and Figure 6-4. From this CAD model the FE models could be built.

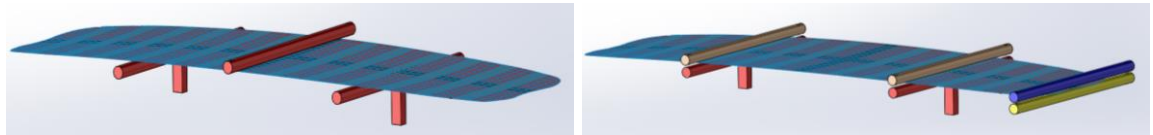


Figure 6-3: CAD Model of the Bending of the Board. Center (L) & Tip (R).

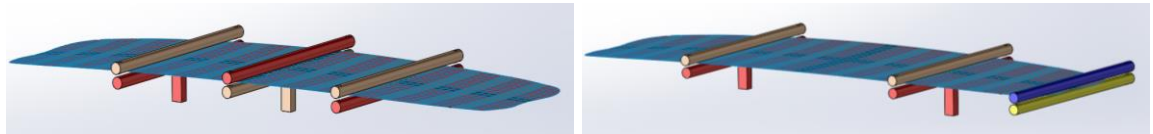


Figure 6-4: CAD Model of the Torsional Stiffness of the Board. Center (L) & Tip (R).

6.4 Optimisation of the Board

In order to design a board such that it has the same structural characteristics as the carbon board assessed in section 5, the following optimisation run was set up within Design Studio. Note that the rail on the outer edge has been ignored in the FE model as suggested by Grant (128). Although it has been found previously by Seto (130) that the rail could potentially increase the stiffness of the board further, however the work by Seto was a metal rail on a snowboard. It is assumed that a PU rail on a kitesurf board would have less influence, and hence this has been ignored, although this may want to be investigated further in the future. In order to optimise the board, variables, an objective and constraints were needed and were defined as discussed in sections 6.4.1, 6.4.2 and 6.4.3.

6.4.1 Objective

The objective was to minimise mass, as this would benefit the performance of the board as well as reducing the material costs during manufacture and shipping costs.

6.4.2 Constraints

The constraints were as follows;

1. The displacements at points along the board were limited so that when loaded in bending, the bending stiffness of the board replicates the bending stiffness obtained in section 5.3.1.3 (+/-1mm). The justification for this tolerance on the constraints can be seen in appendix D. This would ensure that the bending stiffness along the length of the board replicates those seen in the benchmark board.
2. The displacement at points along the board were limited so that when loaded in torsion, the torsional stiffness replicates the torsional stiffness obtained in section 5.3.2.3 (+/-1mm). The justification for this tolerance on the constraints can be seen in appendix D. This would ensure that the torsional stiffness along the length of the board replicates those seen in the benchmark board.
3. Initial testing showed that if the board was optimised without constraining the modal response of the board, the mass was centralised and the natural frequencies were significantly higher than the original board. To stop this happening two NVH constraints were placed upon the board during optimisation. The bending mode was limited to 23Hz and the torsional mode was limited to 46Hz to force the mass to be placed at the outer edges of the board to lower the natural frequency in line with the values seen in section 5.3.3.3.

4. At least one of the plys had to go over the entire surface of the board to ensure the continuity of the structure (achieved by adding a layer on the PCOMP that was the thickness of 2 plys and not including this in the optimisation. The board was optimised with this at 0 degrees and then at 45 degrees to see which resulted in a lighter board). In terms of manufacturability, there were no constraints in terms of the ply sequences, unlike the Aerospace industry for example that specifies that no plies of the same orientation are allowed to be stacked continuously for 3 or 4 plies as taken to account in the 'shuffling' feature within Optistruct (87).
5. The board was defined to be symmetrical about the longitudinal and lateral central axis, as well as balanced to ensure that the board did not distort during manufacture (achieved via Topometry symmetry constraints and having the same variable on either side of the core).

6.4.3 Variables

The following variables were defined;

1. The thickness of the wooden core was varied between 1 and 10mm.
2. The thickness of the 0 and the 45 degree reinforcement layers were varied between 0.001mm and 1.3mm (0 – 5 plys).
3. Topometry was applied using an area of 100mm x 100mm. Reinforcement patches and the core thickness could thus be changed in patches that were at least 100mm x 100mm in size.

6.4.4 Further Considerations

When building the FE model, the following considerations also needed to be considered;

1. The shell thickness in the contact searching was ignored as the optimisation was changing the shell thickness at each iteration.
2. It was ensured that the bottom of the board was smooth and any thickness was added to the top of the board as the bottom of the board needed to be smooth from a fluid flow perspective.

6.5 Suggested Lay-up

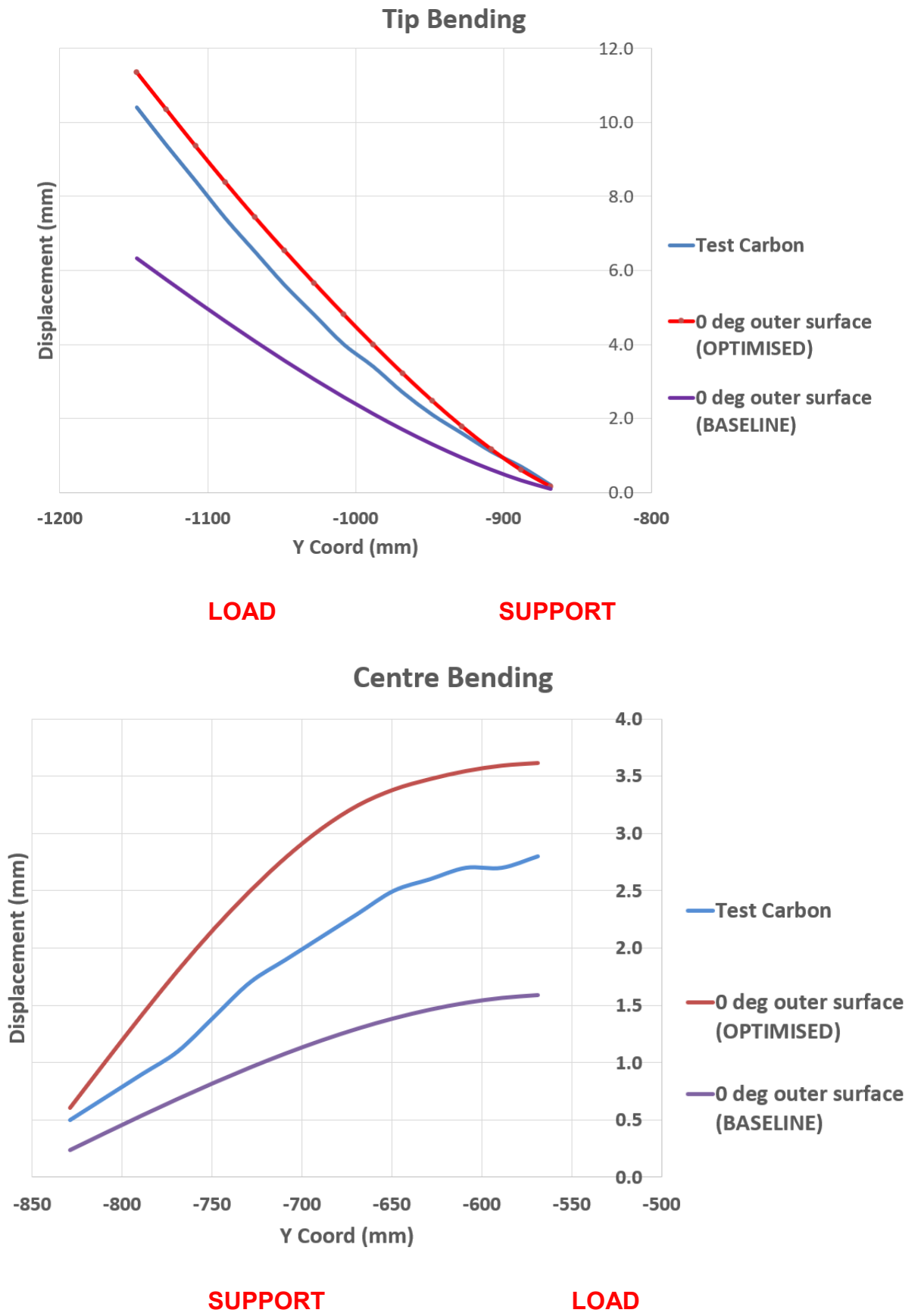
The baseline run was set-up as follows and then optimised.

1. Two layers at 0 degrees (one represented the outer ply which was non-designable and other represented the designable plys which was designable in thickness's representing 0-5 plys).
2. One layer at 45 degrees (which represented the designable plys which was designable in thickness's representing 0-5plys).
3. The wooden core at 10mm thick (which was designable from 1-10mm).

The CFRP layers were symmetrical about the wooden core. As such the baseline board consisted of 3 complete plys top and bottom with a 10mm wooden core. The optimised board could have a core between 1mm and 10mm with up to 5 plys top and bottom at 0 degrees, up to 5 plys top and bottom at 45 degrees with an additional ply which went over the complete top and bottom of the board at 0 degrees. All of the variables could be changed in patch sizes of at least 100mm x 100mm and were also symmetrical about the longitudinal and the lateral central axis.

The above was then repeated, with the outer layer which was non-designable, at 45 degrees to see if this resulted in a lighter board. However, this did not converge with zero percentage violation suggesting that all of the constraints could not be met.

The results of the bending and the torsional stiffness of the FE models can be compared to the experimental results in Figure 6-5 and Figure 6-6. These graphs show the design once it had been optimised, along with the equivalent experimental results that it has been optimised to. In addition, the baseline model for the board with the 0 degree outer layer as discussed above is shown so that it can be compared. This procedure mimics the experimental procedures used in section 5.3.1 and section 5.3.2. It can be seen that the board in the baseline model was far too stiff, whilst the optimised board matches the distributions seen in the equivalent experimental results within the $\pm 1\text{mm}$ tolerance specified.



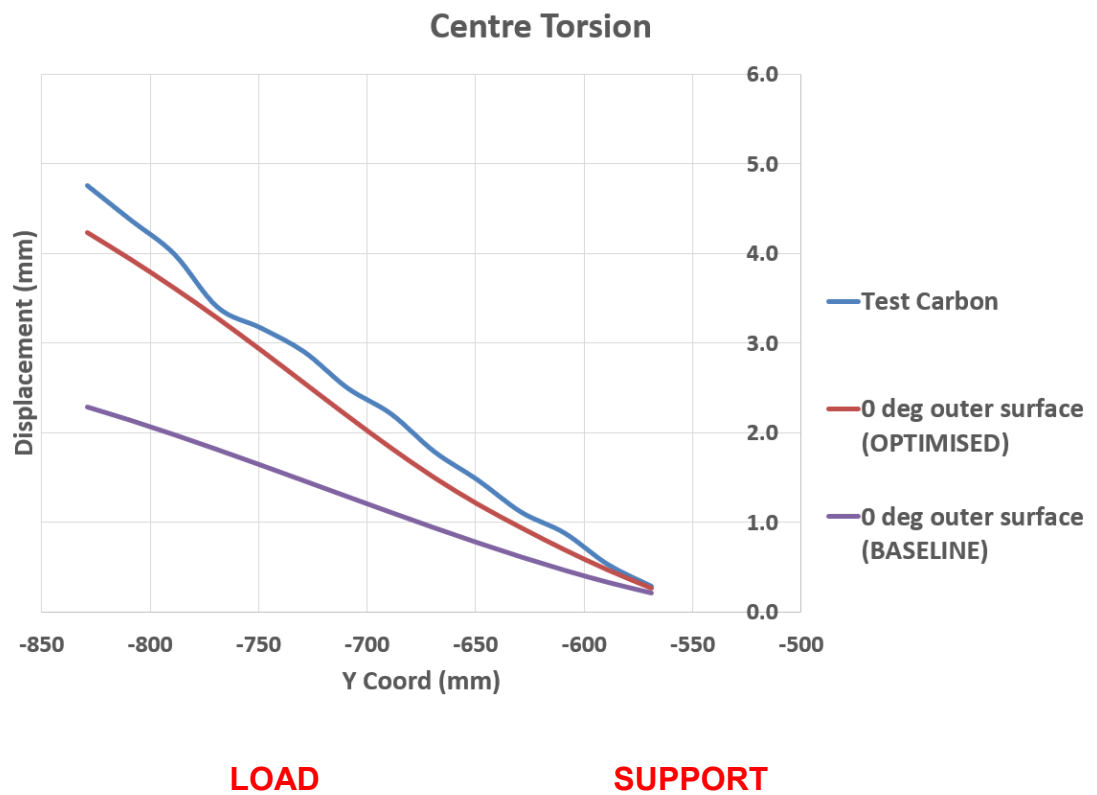
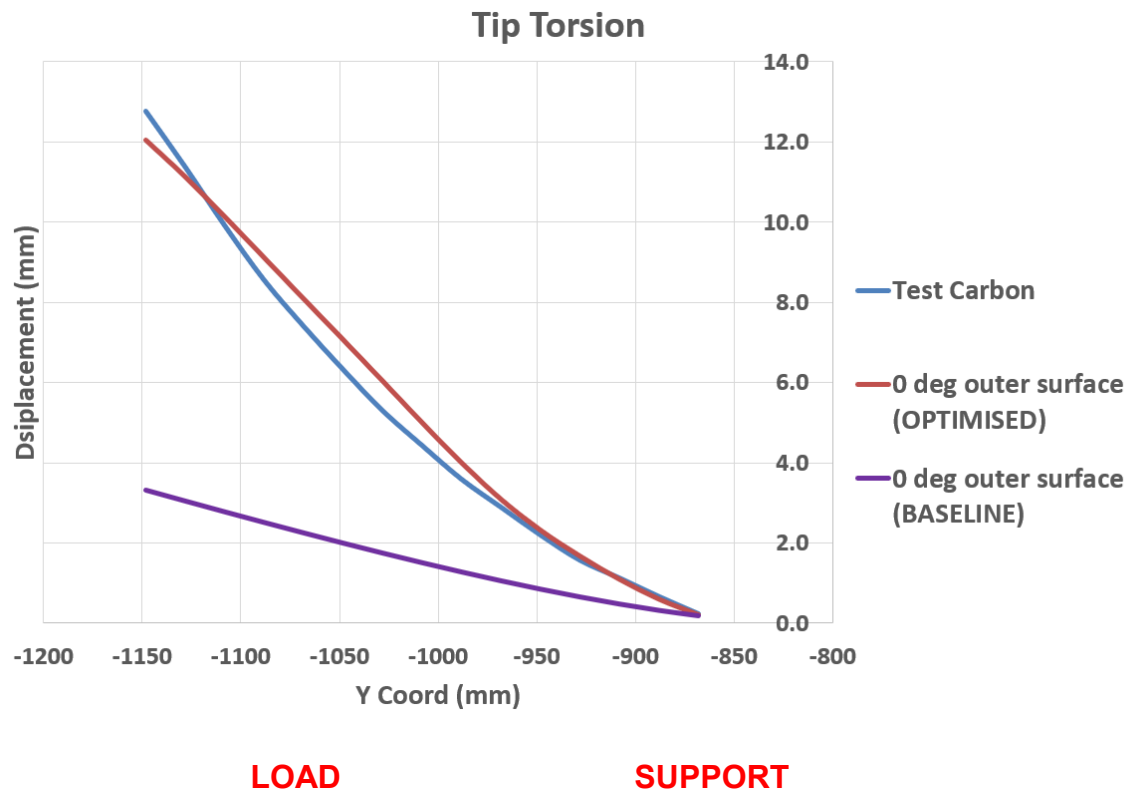


Figure 6-6: Stiffness Distribution of the Experimental, Baseline & Optimised Boards (Torsion).

The masses of the boards are as follows. The optimised board was 1.88kg compared to the original board that was 2.62kg. The first bending mode of the optimised board was 23.0Hz (compared to the original board that had a first bending mode of 22.5Hz as found in section 5.3.3.3) and the optimised board had a first torsional mode of 44.0Hz (compared to the original board that had a first torsional mode of 45.3Hz as found in section 5.3.3.3). The suggested thicknesses of each of the layers for this board are shown in Figure 6-7. Note that the scale has been set so that each increment represents one ply thickness in the CFRP layers and 1mm in the wooden core.

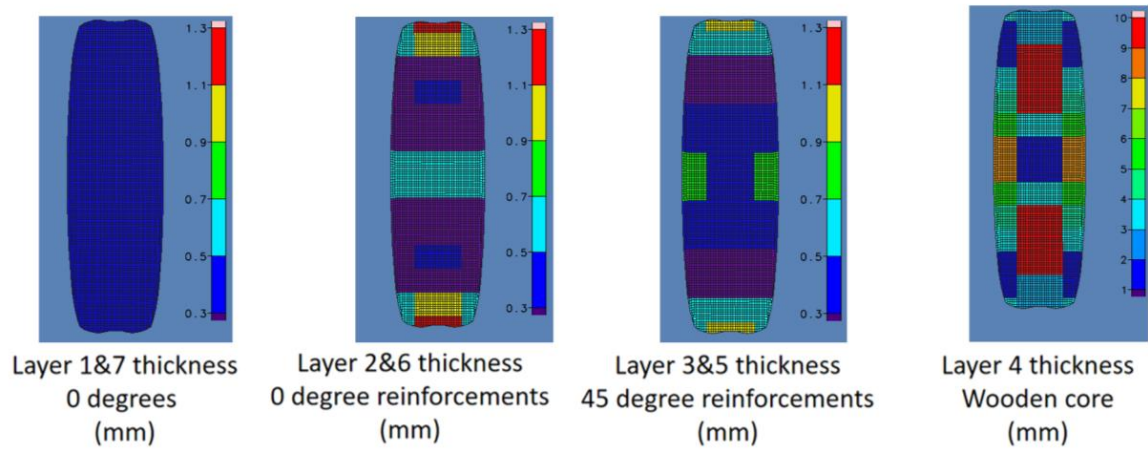


Figure 6-7: The Suggested Lay-up of the Optimised Board: Complete 0 Degree Ply.

A representation of this board with the thicknesses displayed can be seen in Figure 6-8.

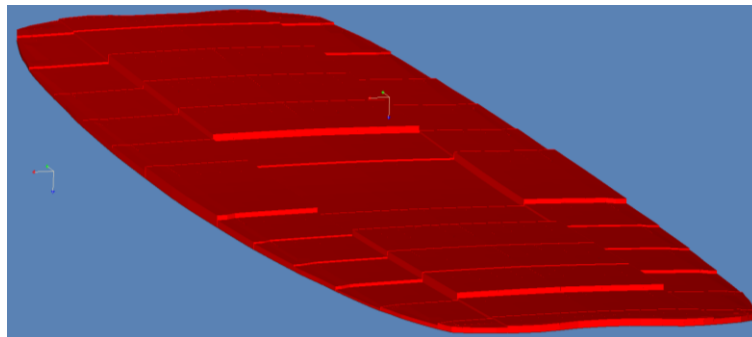


Figure 6-8: The Optimised Board.

This design was then interpreted into a manufacturable design which uses discrete ply thicknesses for each layer rather than the real numbers that were used in the optimisation run. This interpretation can be seen in Figure 6-9. This figure shows the plys as they would be entered into the mould going left to right. The manufacturable interpretation of the board was 1.89kg compared to the optimised board that was 1.88kg.

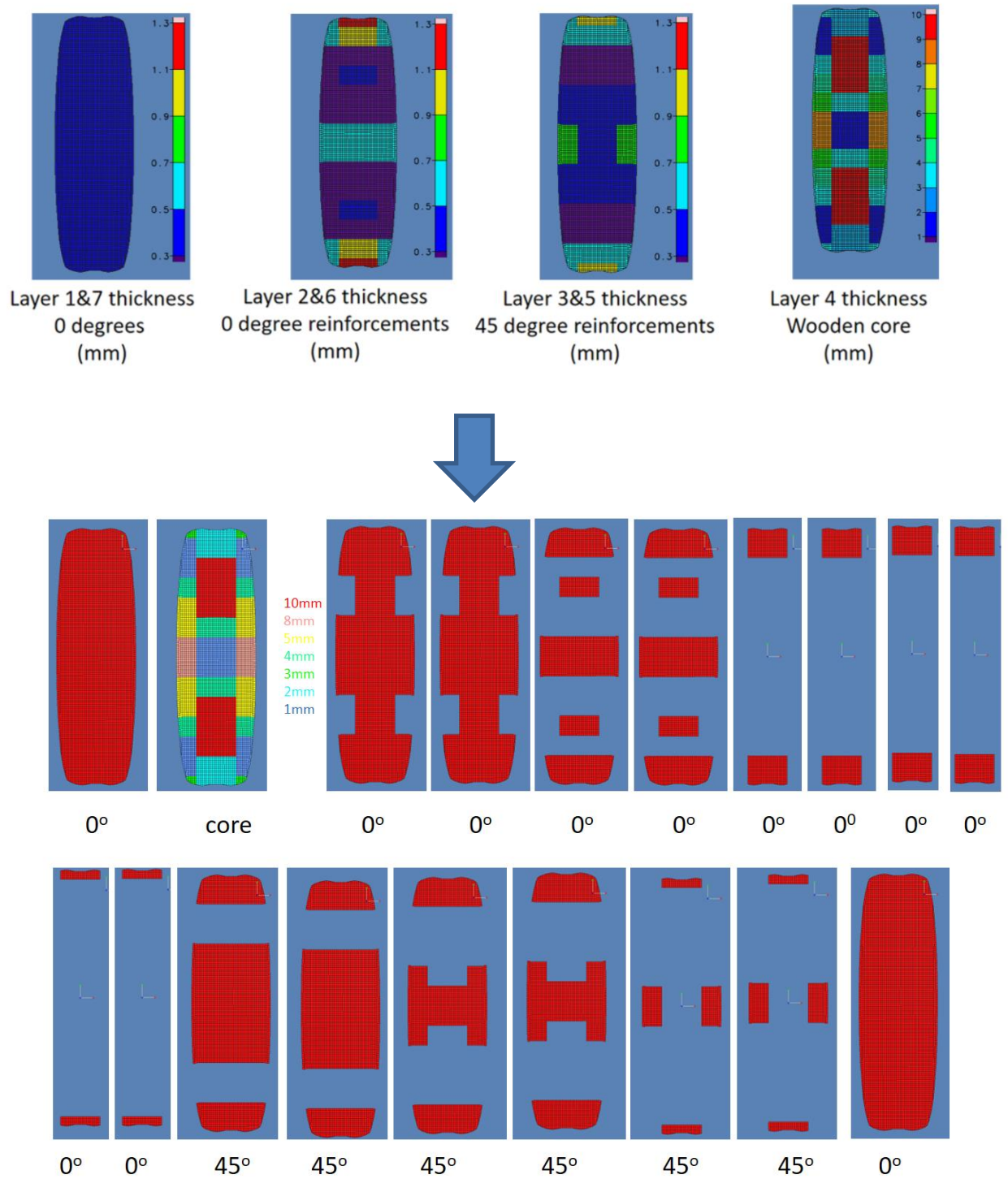
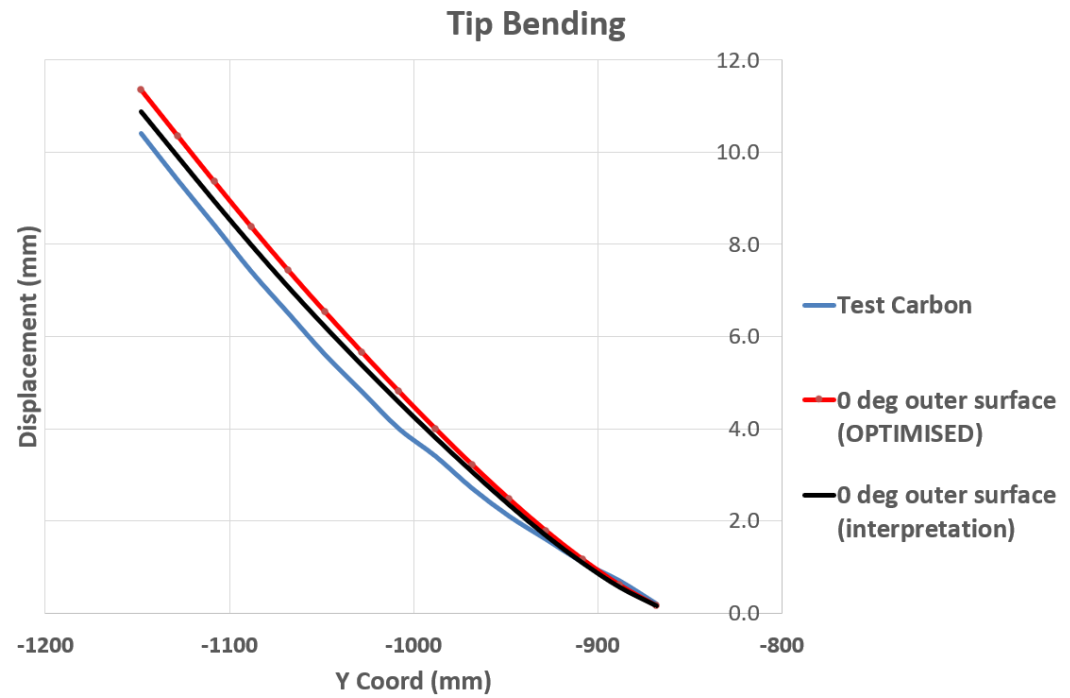


Figure 6-9: Interpretation of the Board into a Manufacturable Solution.

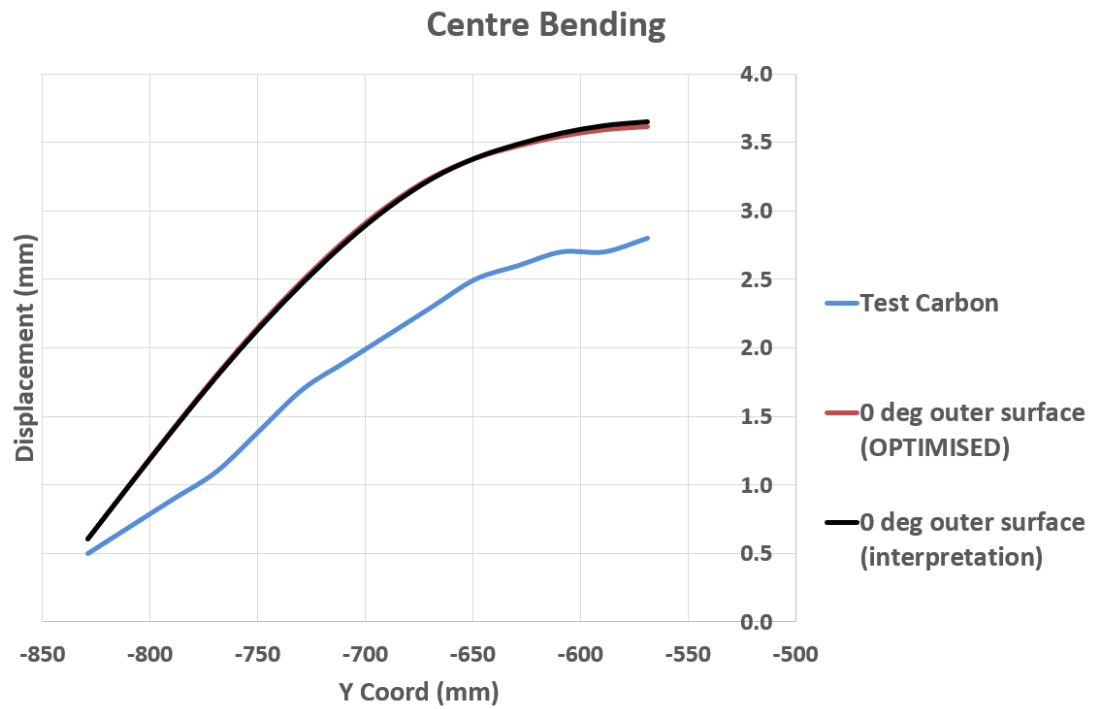
6.6 Assessment of the Interpretation of the Suggested Lay-up

The interpretation of the optimised board was then assessed in bending and torsion so that it could be compared to the optimised design to check that the structural performance was similar. The results of this can be seen in Figure 6-10 and Figure 6-11.



LOAD

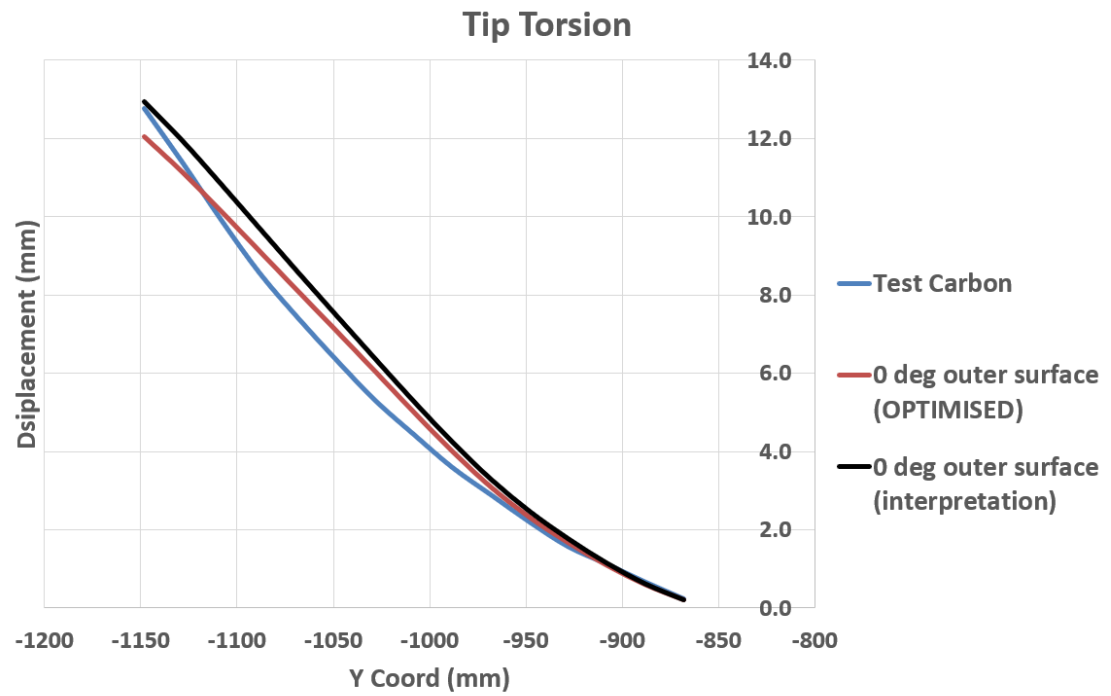
SUPPORT



SUPPORT

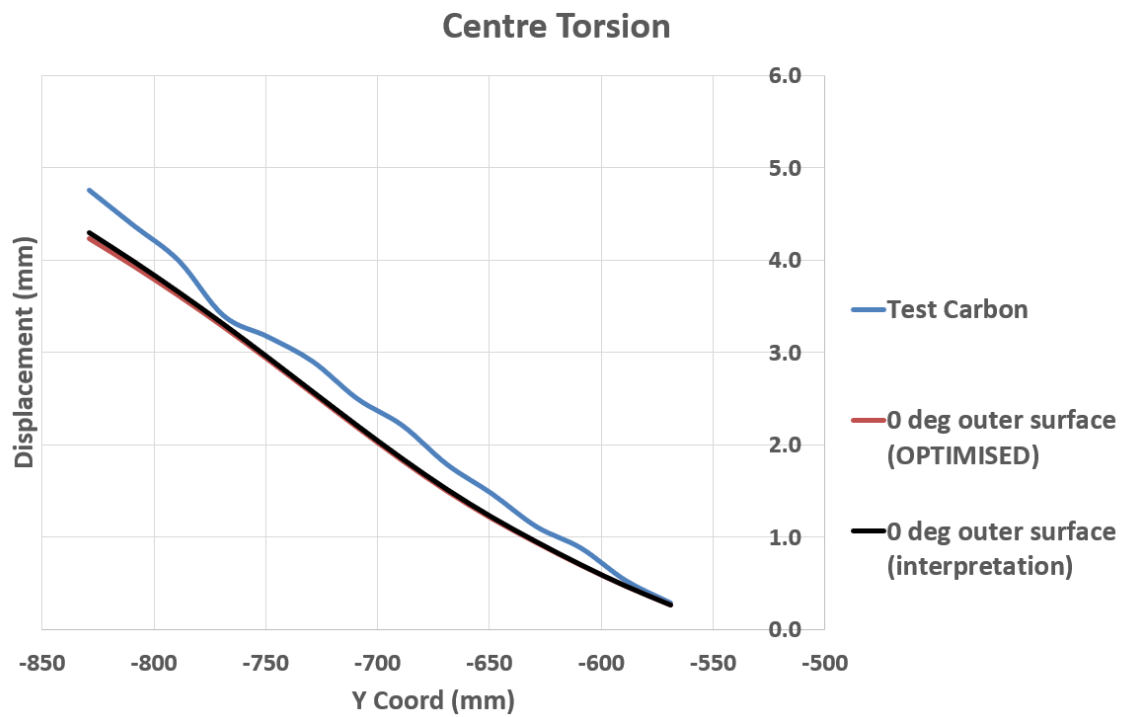
LOAD

Figure 6-10: Assessment of Bending of the Interpretation of the Suggested Lay-up.



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SUPPORT



LOAD

SUPPORT

Figure 6-11: Assessment of Bending and Torsion of the Interpretation of the Suggested Lay-up.

It can be seen that the bending and the torsional stiffness of the manufacturable interpretation of the optimised board closely matches the distributions seen in the optimised boards. An NVH assessment of the design was carried out. This can be seen in Figure 6-12.

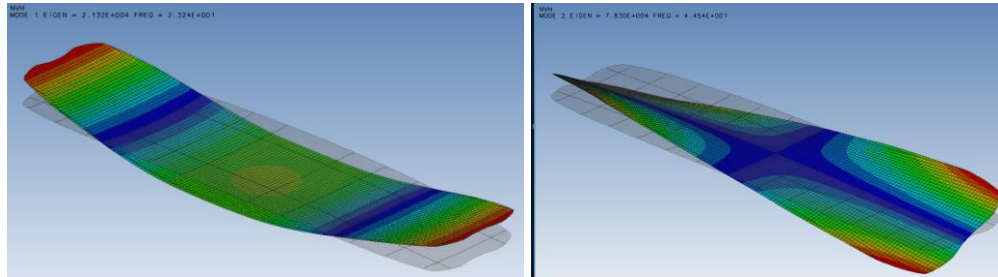


Figure 6-12: Assessment of the NVH of the Interpretation of the Suggested Lay-up.

The first mode was a bending mode at 23.2Hz. This can be compared to the original board that had a bending mode of 22.5Hz and the second mode was a torsional mode at 44.5Hz, which can be compared to the original board that had a torsional frequency at 45.3Hz.

6.7 Assessing the Design to Ensure it does not Fail

In order to establish if this board would fail during use, it was assessed using the mock field test and using the failure criteria that is used in Design Studio as outlined in section 4.3 to ensure that it was fit for purpose.

6.7.1 Assessment using the Tsai-Hill Failure Theory

Using the Tsai-Hill failure theory the maximum failure index across all of the plies can be seen in Figure 6-13, where a value of 1 would represent failure, i.e. the elements are seeing 100% of what is acceptable according to Tsai-Hill.

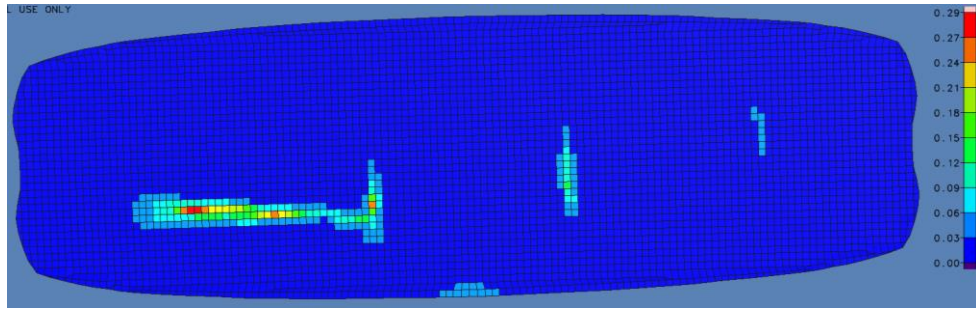


Figure 6-13: Mock Field Testing: Hill Failure Index.

As can be seen the maximum failure index is 0.29 and thus significantly below the value that it would be expected to fail at.

6.7.2 Assessment using the Hoffman Failure Theory

Using the Hoffman failure theory, the maximum failure index across all of the plies can be seen in Figure 6-14, where a value of 1 would represent failure, i.e. the elements are seeing 100% of what is acceptable according to Hoffman.

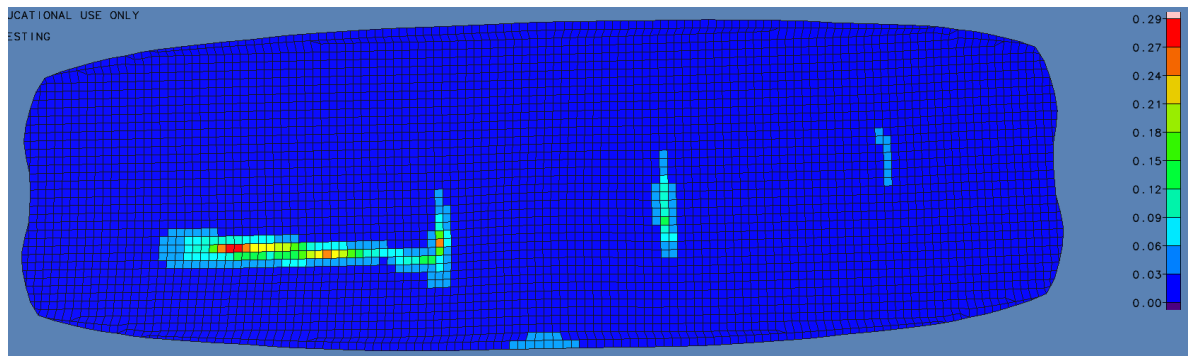


Figure 6-14: Mock Field Testing: Hoffman Failure Index.

As can be seen the maximum failure index is 0.29 and thus significantly below the value that it would be expected to fail at.

6.7.3 Assessment using the Maximum Strain Failure Theory

Using the Maximum Strain failure theory, the maximum failure index across all of the plies can be seen in Figure 6-15, where a value of 1 would represent failure, i.e. the elements are seeing 100% of what is acceptable according to this theory.

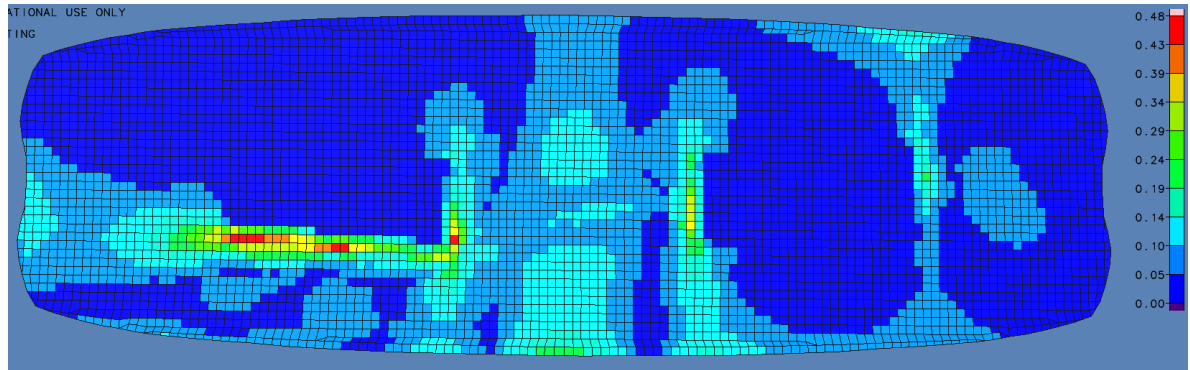


Figure 6-15: Mock Field Testing: Maximum Strain Failure Index.

As can be seen the maximum failure index is 0.48 and thus significantly below the value that it would be expected to fail at.

6.7.4 Assessment using the Tsai-Wu Failure Theory

Using the Tsai-Wu failure theory, the maximum failure index across all of the plies can be seen in Figure 6-16, where a value of 1 would represent failure, i.e. the elements are seeing 100% of what is acceptable according to Tsai-Wu.

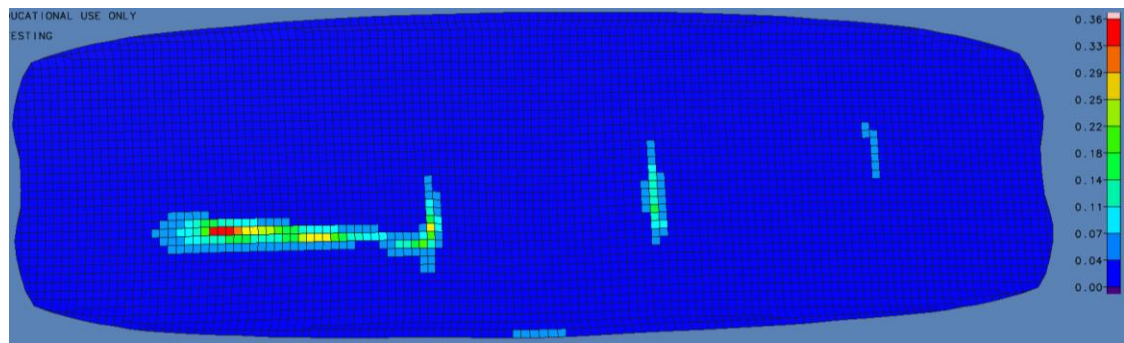


Figure 6-16: Mock Field Testing: Tsai-Wu Failure Index.

As can be seen the maximum failure index is 0.36 and thus significantly below the value that it would be expected to fail at.

As can be seen in this section, even taking the most pessimistic failure criteria (in this case the Maximum Strain criteria), this board would be able to take approximately 2 times the loading ($1/0.48$) as seen in this mock field test before it is expected to fail. It

should be noted that the geometry of the footpads would be such that the loading from this would be spread over a larger area and not have stress concentrations such as those caused by the 90 degree corners used in the mock field test. As such, the loadings from the representations of the footpads in the mock field test would be more severe than the loadings from the real footpads. It should also be noted that the geometry of the foam was morphed (flattened) where the board contacted this foam in order to make the contacts more stable. Despite these approximations in the FEA model, the failure index was found to be well within what was acceptable. Therefore, it was assumed that this board could be manufactured and tested with confidence that this would not fail. It should also be noted that this procedure could be used to give some indication as to the durability of such a component.

6.8 Conclusions

This chapter has demonstrated the feasibility of the proposed methodology.

Established optimisation techniques have been applied to determine how the board should ideally be laid-up. This has been achieved by considering the variables, objectives and constraints that are necessary when optimising anything. The variables have been assumed such that up to 5 reinforcement plys can be placed at 45 degrees and at 0 degrees and that these are symmetrical. The objective was to minimise the mass resulting in less inertia for the rider when riding the board, less material usage and less shipping costs. The constraints have been set such that the stiffness along the length of the board in terms of both bending, torsion and vibrational behaviour is similar to the baseline board seen in section 5. This ideal optimised board has then been interpreted into a manufacturable equivalent and has shown behaviour that is close to the ideal board. Finally, by considering the loading in the mock field test, the

board has been shown to be able to withstand approximately twice the loading that the board would be likely to see.

7 Verification of the Design Methodology

In order to prove that the design methodology works, the proposed board could be manufactured and compared to the responses of the original board that was seen in section 5. However due to the fact that the materials are not known that have been used in the manufacture of the original board this could still be lighter if it had used more 'advanced' materials. To conclusively prove that the proposed design methodology can successfully be applied to components such as Kitesurf boards, a scale board that is representative of the boards considered in this work was manufactured using the materials tested in this work and evaluated using a generic lay-up. An optimised version of this board was then designed using the proposed methodology. Finally, the optimised board was then manufactured using the same materials. The structural performance and weights of the original scale board and the optimised scale board could then be compared. As such the design methodology could be verified.

7.1 FE Models

7.1.1 Arbitrarily Laid-up Board

An FE model of the generically laid-up board (which was similar in construction, but a 50% scale model, of the boards considered in section 5), was created using the material properties that have been established section 4. This board was simplified such that it was flat (it had no rocker or concave) it had no inserts, and it had no rail, as this would significantly simplify the manufacturing process. In order to prove the methodology, the lay-up would be isolated and optimised. This FE model can be seen in Figure 7-1. This model uses the rationale for the element choices as discussed in section 6.3. Note that this model uses 2D elements and PCOMPS. The thickness of

these PCOMPS are displayed in Figure 7-1 so that the construction of the board can be better understood.

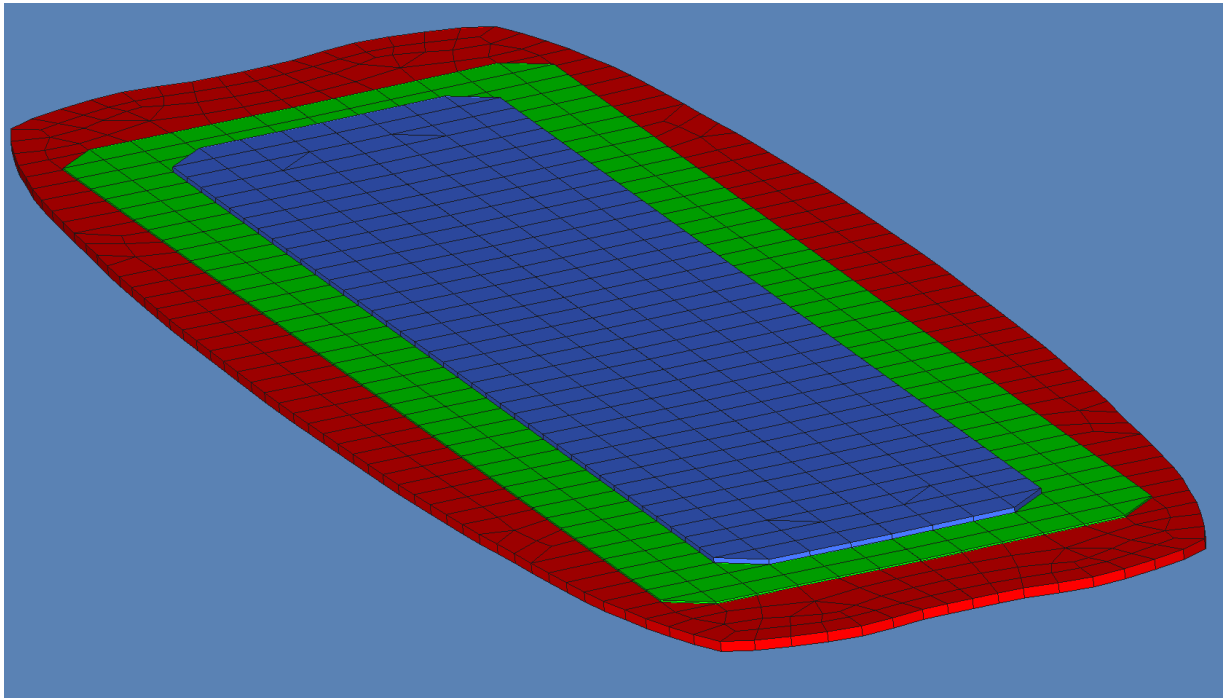


Figure 7-1: Original Scale Board with Generic Lay-up.

The generic layup of this board is shown in Figure 7-2.

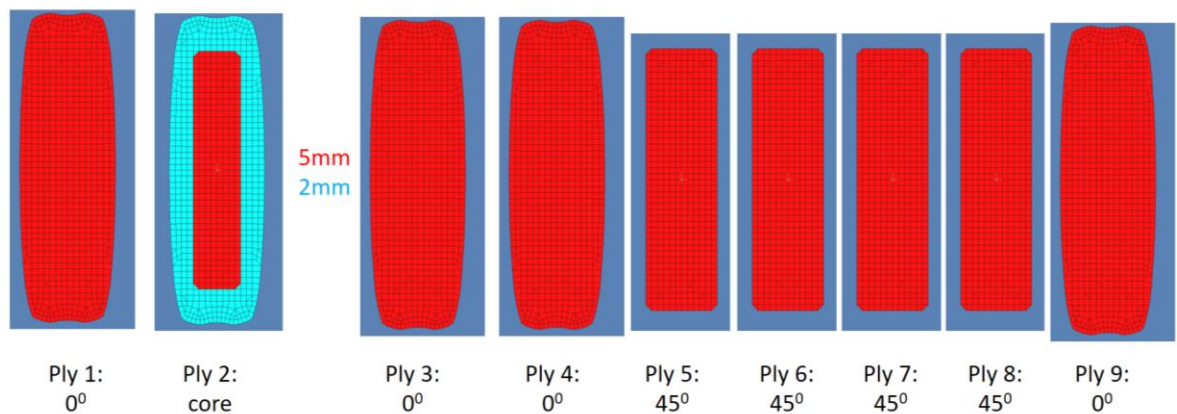


Figure 7-2: Arbitrary Lay-up of the Original Board.

The board was arbitrarily laid-up using 'best practice' that has been informed by the research carried out in section 2. The construction consists of a wooden core that is 2mm thick at the edges which increases to 5mm thick in the centre. There are then 4

plys at zero degrees that run over the entire board and then 4 reinforcement plys that are at 45 degrees placed at the board's centre.

7.1.1.1 Bending

The FE model of the board was loaded in bending to see the response that would be used to optimise the board using the proposed methodology. The board was clamped in 6dofs at the edge of the 45 degree reinforcements (shown in green) and then a force of 20N was applied at either end of the 250mm wide RBE2 that represents the loading bar that is clamped to the board at the edge of the 5mm raised section of the wooden core (shown in blue).

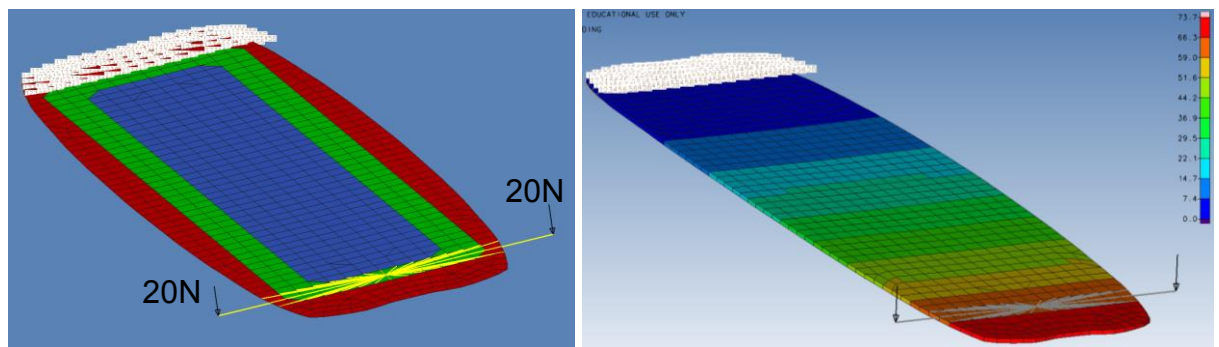


Figure 7-3: Bending of the Original Scale Board.

This resulted in a maximum deflection of 73.7mm.

7.1.1.2 Torsion

The FE model of the board was then loaded in torsion to see the response that would be used to optimise the board using the proposed methodology. The board was clamped in 6dofs at the edge of the 45 degree reinforcements (shown in green) and a force of 50N applied in opposite directions at either end of the 250mm wide RBE2 that represents the loading bar that is clamped to the board at the edge of the 5mm raised section of the wooden core (shown in blue).

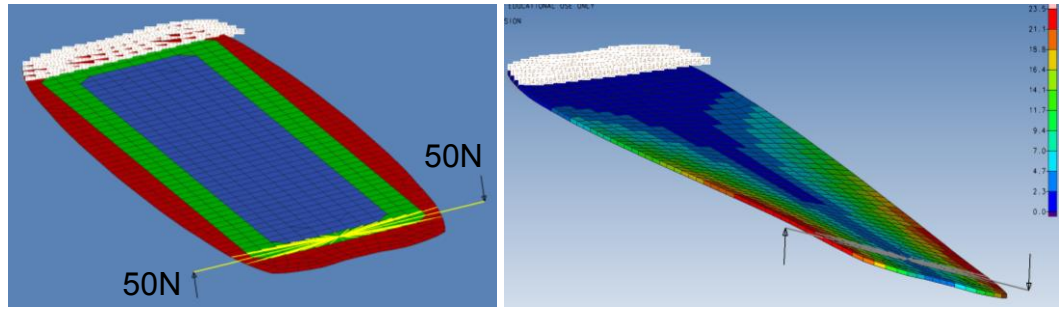


Figure 7-4: Torsion of the Original Scale Board.

This resulted in a maximum deflection of 23.5mm.

7.1.1.3 Vibration

The vibrational performance of the board was assessed whilst clamping the board using the same constraints as for the bending and the torsional loadcases. The results of this can be seen in Figure 7-5.

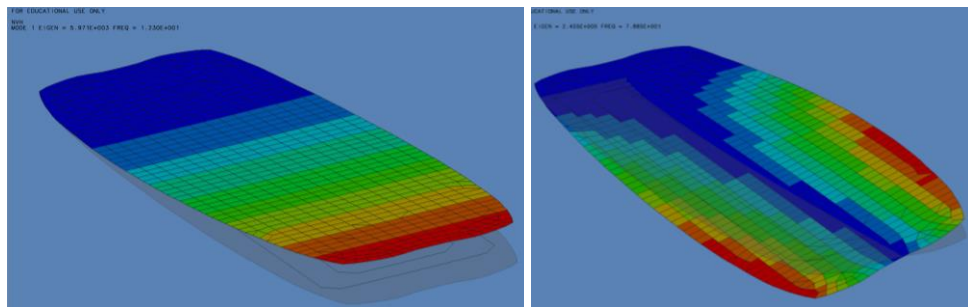


Figure 7-5: Vibration of the Original Scale Board.

This resulted in a first bending mode of 11.5Hz and a first torsional mode of 51.8Hz.

7.1.2 Alternative Board using the Proposed Design Methodology

An alternative structure was then optimised in the same way that the alternative board was optimised as discussed in section 6.4. The stiffness distributions obtained in section 7.1.1.1 and section 7.1.1.2, as well the NVH results obtained in section 7.1.1.3 were used as design constraints. There was a zero degree ply that run over the entire board on the top and the bottom of the board. The variables were the 0 and 45 degree reinforcements which could be placed in patches 50mm x 50mm and could be between

1e-6mm and 1.3mm thick (0 and 5 plies). The core could be between 1mm and 5mm in 50mm x 50mm patches. This had the same symmetry constraints as the board designed in section 6. The board was then interpreted into a manufacturable equivalent which was manufactured and tested so that the method could be verified. The comparison of the bending and the torsional performance of the FE models of the original board and the manufacturable interpretation can be seen in Figure 7-6. These graphs show the response of the original (generically laid-up board) and the interpretation of the optimised as discussed previously in section 6.6.

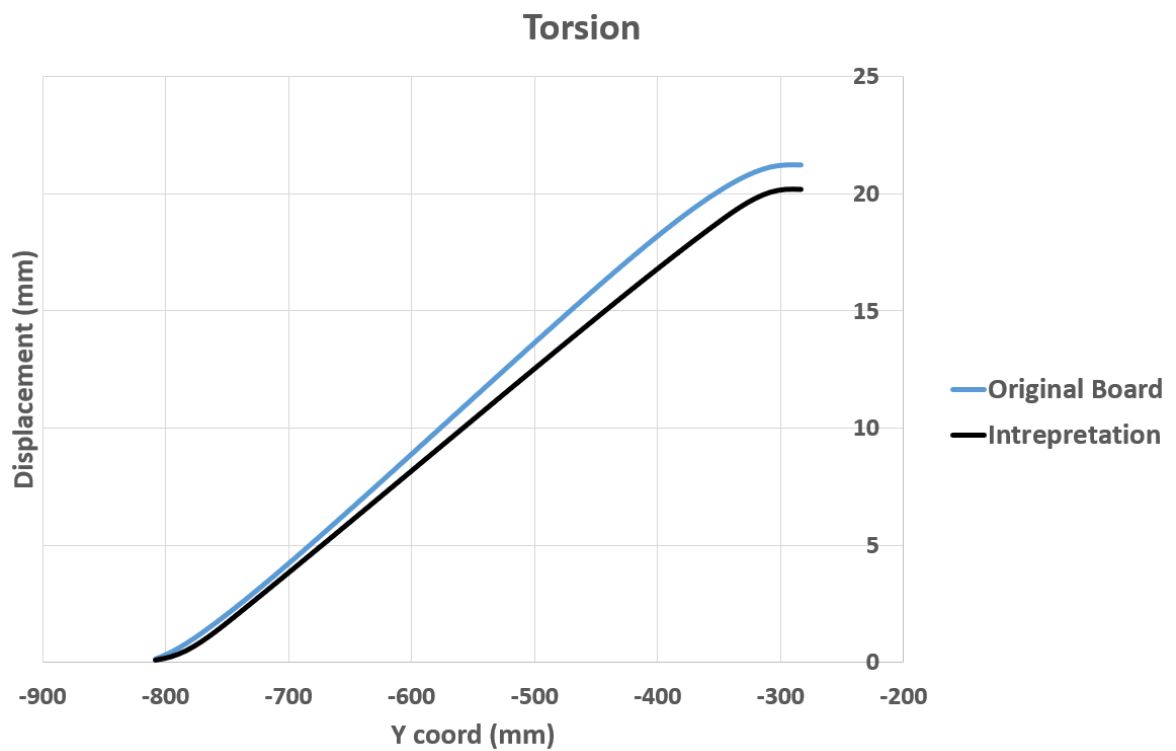
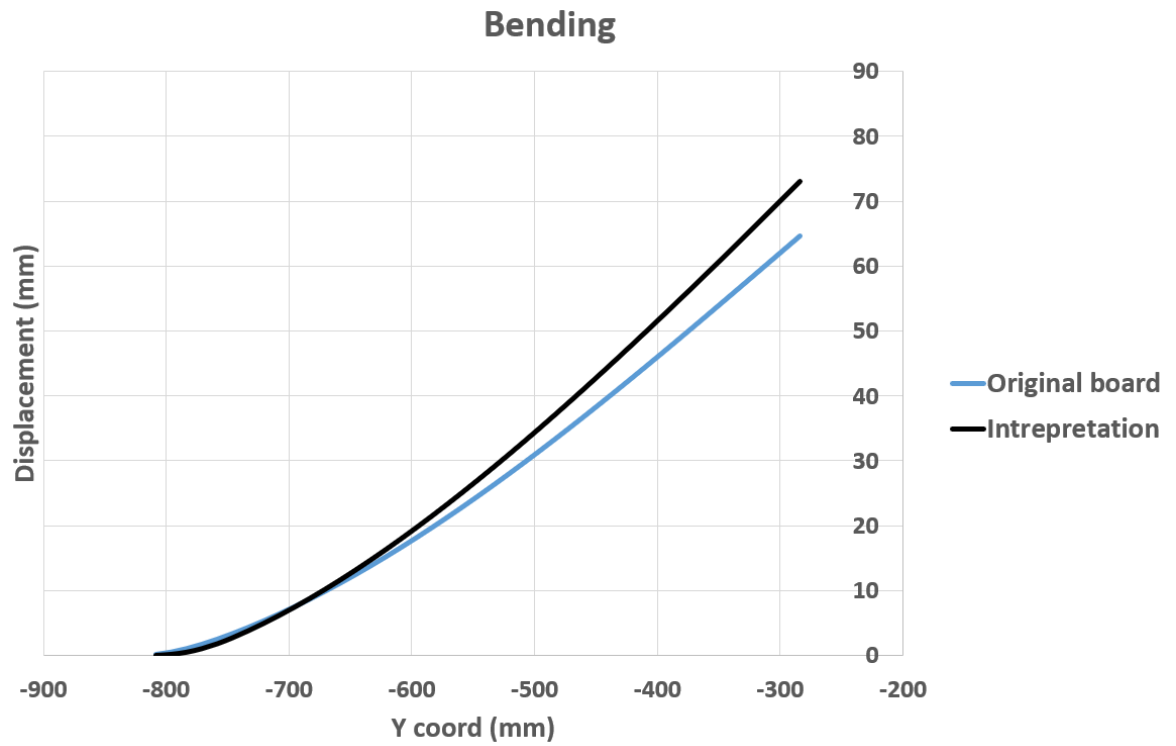


Figure 7-6: Stiffness Distributions of the Original & Interpretation of the Optimised Scale Boards.

As seen in Figure 7-6, the stiffness distribution of the interpretation of the optimised board closely matches the original board both in terms of bending and torsion. The original board had a first bending mode of 11.5Hz and a first torsional mode of 51.8Hz. This compared to the interpretation of the optimised board that had a first bending mode of 11.3Hz and a first torsional mode of 53.0Hz.

7.1.3 Comparison Between the Arbitrarily Laid-up Board and the Optimised Structures

The original structure can be seen in Figure 7-7 on the left and the interpretation of the optimised structure is shown on the right. The original structure weighed 394g, the optimised structure weighed 360g (-9%) and the manufacturable interpretation of this weighed 364g (-8%).

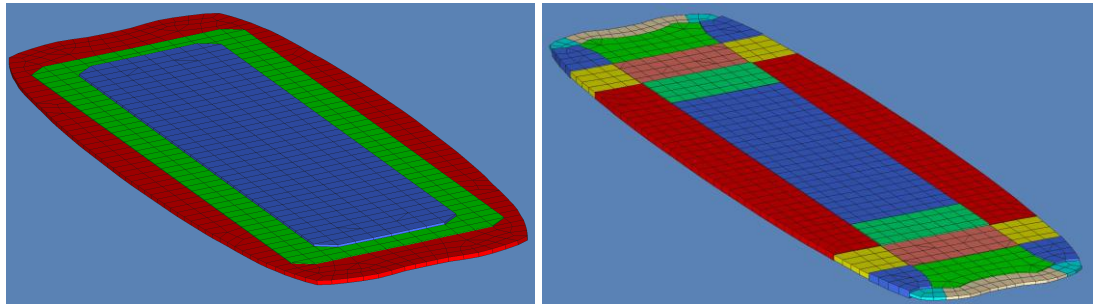


Figure 7-7: Comparison of the Original and the Optimised Scale Boards.

The suggested lay-up of the interpretation board is shown in Figure 7-8.

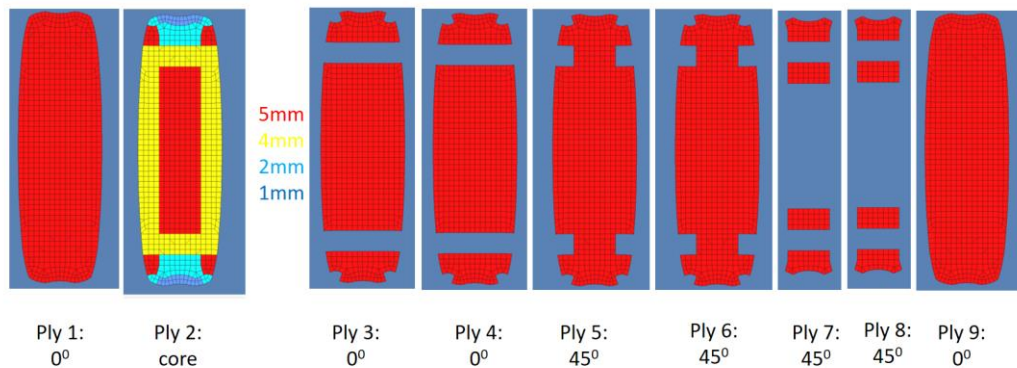


Figure 7-8: The Suggested Lay-up of the Interpretation Board.

As can be seen in Figure 7-8 there were essentially still 4 plies over the entire surface at 0 degrees, however the reinforcement plies that are placed at 45 degrees have changed significantly. In the original board there were 4 plies over the central section of the board. These have been replaced with 2 plies that go over the majority of the board, 8 additional reinforcement patches at the tips and the core has increased in thickness in strategic places to increase the second moment of area in these regions. In addition, since the stiffness is the same at a reduced mass, the mass distribution has changed accordingly (it has placed mass at the outer edges) to ensure that the vibrational behaviour remains the same.

7.2 Physical Components

7.2.1 Manufacture

A CAD representation of the original scale board and the optimised equivalent can be seen in Figure 7-9. Note that in the optimised version of the scale board the wooden core reduced to 1mm at the extreme tips. However, during manufacture, this resulted in the wooden core splitting. In order to stop this from happening, this reduction in thickness was removed. The FE model of the board was updated accordingly to see the implications of this. The stiffness distribution of the board in bending and torsion does not extend to the extreme tips of the board and thus these were not influenced. However, due to the increased weight at the tips of the board the frequency of the first bending mode was reduced by 0.1Hz to 11.2Hz.

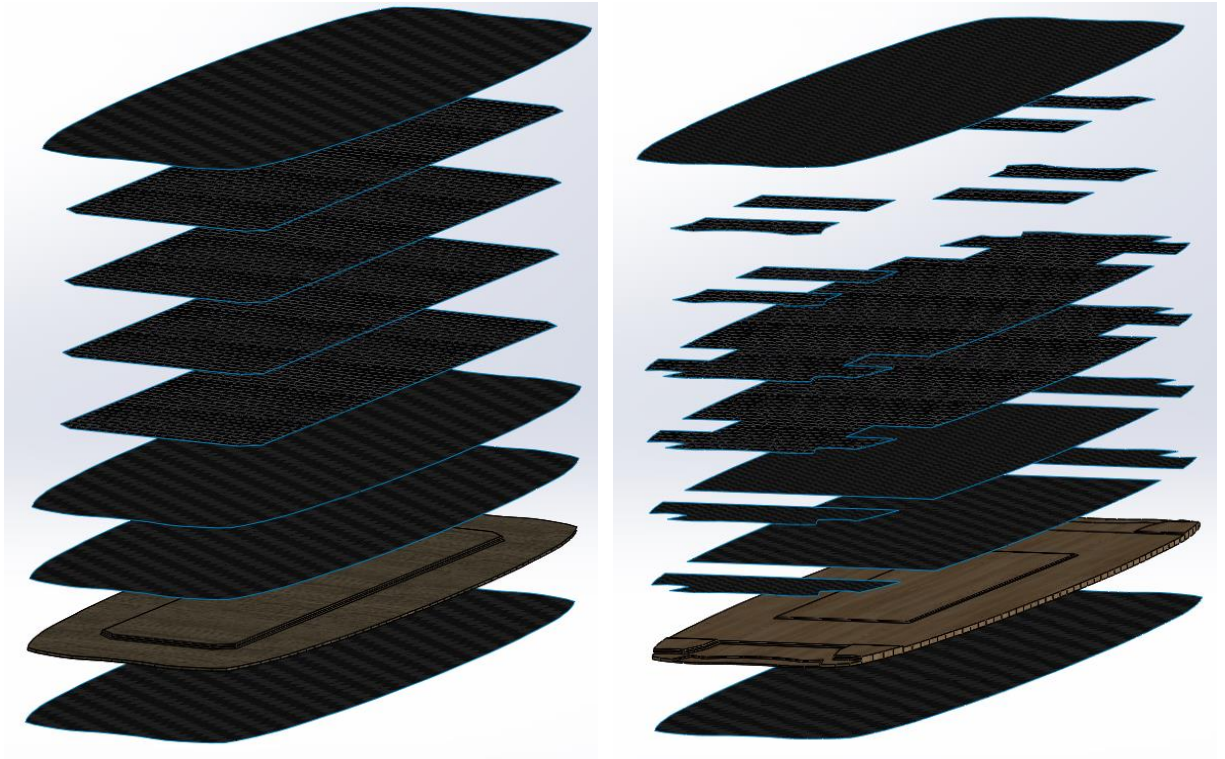


Figure 7-9: A CAD Representation of the Original Scale Board (L) & the Optimised Equivalent (R).

The manufacturing process can be seen in Figure 7-10 and can be described as follows;

1. The wooden core was machined using a CNC machine.
2. Templates of the plies were machined using the CNC machine.
3. Pre-preg was cut by hand using the machined templates.
4. The plies were assembled as shown in Figure 7-9.
5. The boards were cured in the autoclave as per the data sheet provided by the material supplier.
6. Once the boards were cured, they were sanded by hand to remove any sharp edges and then lacquered so that they could be tested.

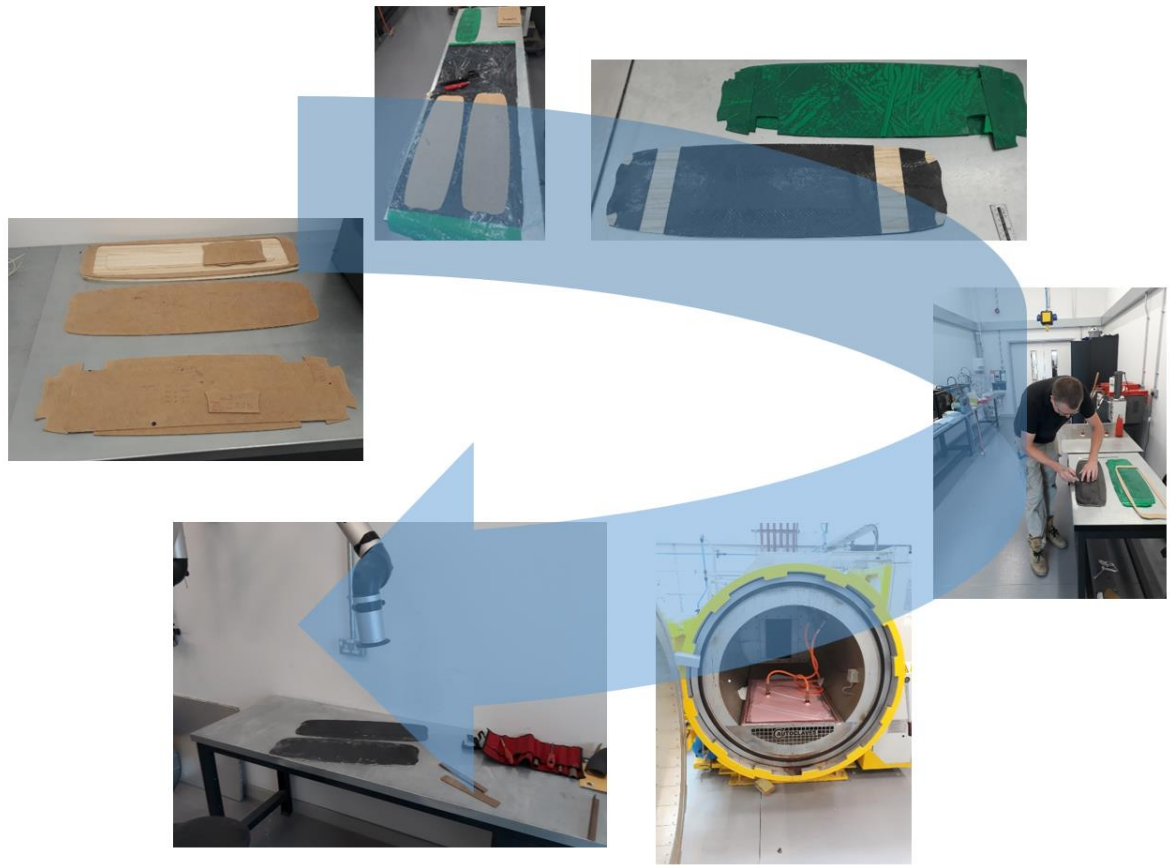


Figure 7-10: Manufacturing the Scale Boards.

The finished scale boards can be seen in Figure 7-11. The physical original board weighed 443g, compared to the FE model of the board that weighed 394g. The physical optimised board weighed 427g (4% less than the physical original), compared to the FE model of the board that weighed 364g. (8% less than the FE model of the original). Note that there are slight differences between the FE models and the physical boards due to the hand finishing and the lacquer that was applied to the boards.

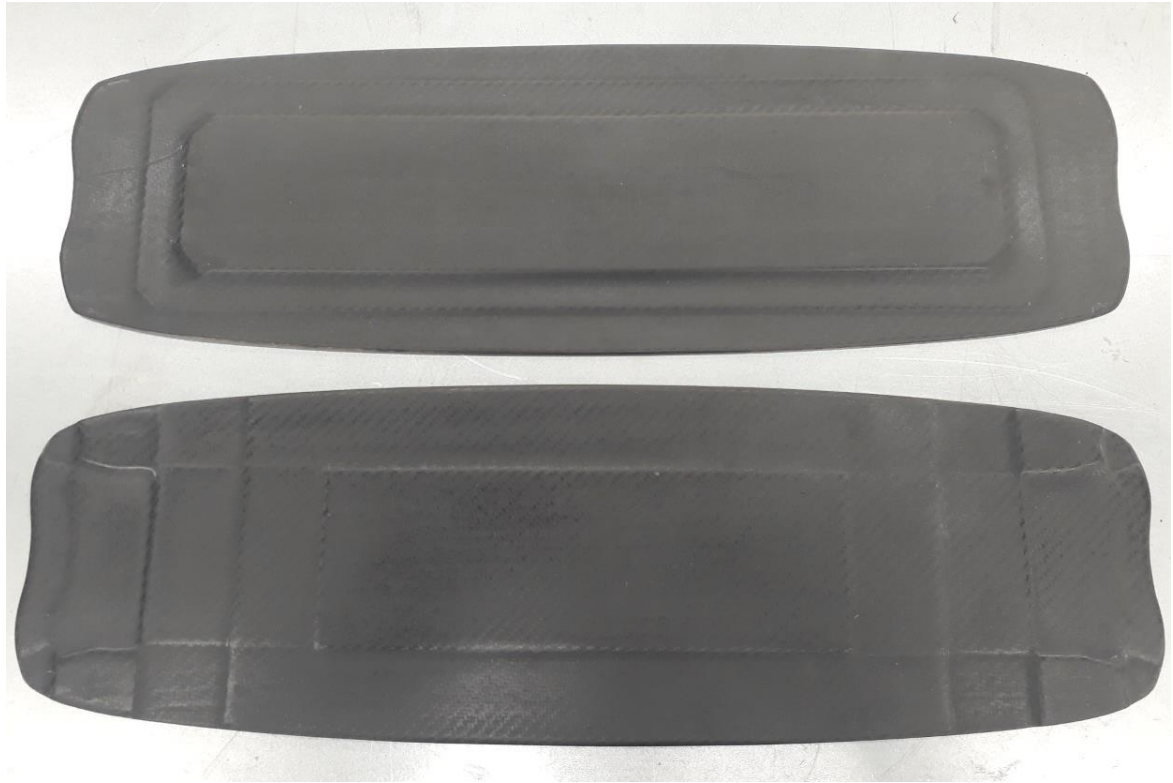


Figure 7-11: The Finished Scale Boards: Original (Top) & Optimised (Bottom).

7.2.2 Bending Testing

According to the proposed design process, the bending stiffness distribution of the original board and the optimised board should be identical. The physical part was tested as shown in section 7.1.1.1 so that the optimised and the original scale boards could be compared. One end of the board was clamped to a workbench and then a loading bar was attached to the other end. Weights were then added to the loading bar. DTI gauges were placed along the length of the board so that the stiffness distribution could be established.

7.2.2.1 Errors

The errors in this section are assumed to be the same as those that were discussed in section 5.3.1.2 where bending testing was carried out on the original boards that are

being benchmarked. With the exception that instead of using the CMM machine to measure the vertical position, DTIs were used. The errors were therefore as follows;

1. The DTIs were accurate to $\pm 0.005\text{mm}$, however an error of $\pm 0.1\text{mm}$ was taken for the height calculation due to the textured nature of the board topsheets as suggested by Clifton (28). Thus when the final position is subtracted from the initial position, this error is doubled to $\pm 0.2\text{mm}$.
2. In terms of position of the measurements, these were taken to be $\pm 0.5\text{mm}$.

7.2.2.2 Results

This experimental set-up can be seen in Figure 7-12, along with the displacement results taken from the DTI gauges of the original and the optimised scale boards.

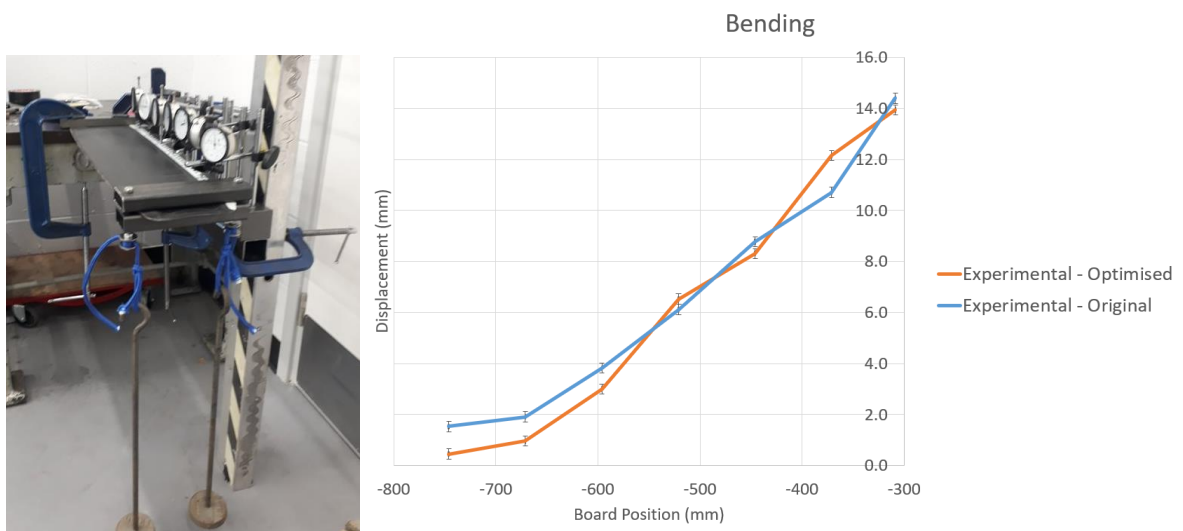


Figure 7-12: Bending of the Original & the Optimised Scale Boards.

As can be seen in Figure 7-12, the bending stiffness distribution along the length of the original and the optimised boards suggests that these boards would feel the same to the rider.

7.2.3 Torsion Testing

The physical part was tested as shown in section 7.1.1.2 so that the optimised and the original scale boards could be compared. One end of the board was clamped to a workbench and then a loading bar was attached to the other end. Weights were then added to the loading bar. On one side of the loading bar, the weights were hung directly on the bar, thus pulling the board down. On the other side, a string was attached to the loading bar, fed over a pulley and attached to the weights. This meant that this side of the loading bar was pulled up, placing the board in torsion. DTI gauges were placed along the length of the board so that the stiffness distribution could be established.

7.2.3.1 Errors

The errors in this section are assumed to be the same as those that were discussed in section 5.3.2.2 where torsion testing was carried out on the original boards that are being benchmarked. With the exception that instead of using the CMM machine to measure the vertical position, DTIs were used. The errors were therefore as follows;

1. The DTIs were accurate to $\pm 0.005\text{mm}$, however an error of $\pm 0.1\text{mm}$ was taken for the height calculation due to the textured nature of the board topsheets as suggested by Clifton (28). Thus, when the final position is subtracted from the initial position, this error is doubled to $\pm 0.2\text{mm}$.
2. The distance from the centre of the board to the edges where the measurements were taken was 75mm . This means that the error was:

$$\tan^{-1}\frac{0.1}{75} = \pm 0.08^\circ.$$

3. In terms of position of the measurements, these were taken to be $\pm 0.5\text{mm}$.

7.2.3.2 Results

This experimental set-up can be seen in Figure 7-13, along with the displacement results taken from the DTI gauges of the original and the optimised scale boards.

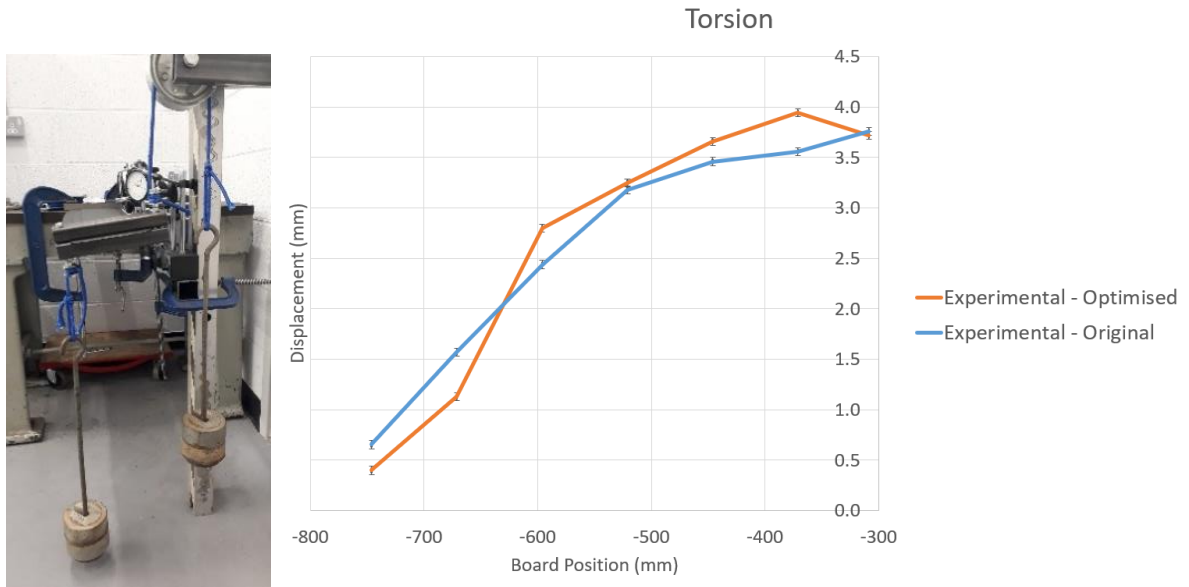


Figure 7-13: Torsion of the Original & the Scale Boards.

As can be seen in Figure 7-13 the torsional stiffness distribution along the length of the original and the optimised boards suggests that these boards would feel the same to the rider.

7.2.4 Vibration Testing

The physical part was tested as discussed in section 4.5 (when the vibrational performance of the flat CFRP plates was assessed) so that the vibrational response of the optimised and the original scale boards could be compared.

7.2.4.1 Errors

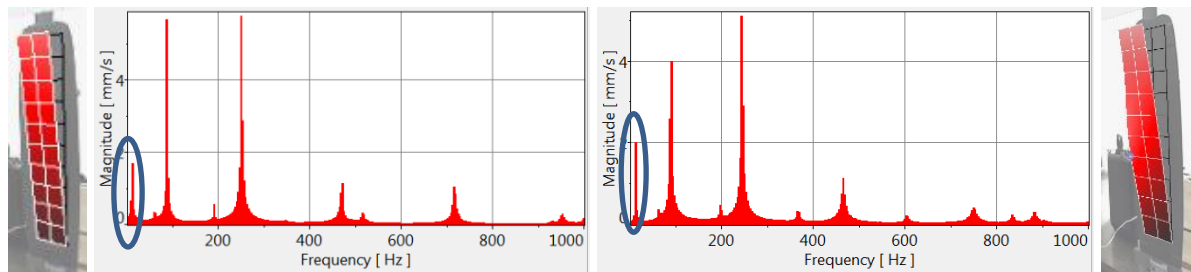
The errors in this section are assumed to be the same as those that were discussed in section 4.5.3 where vibration testing was carried out on the CFRP samples when considering the material properties, section 5.3.3.2 where vibration testing was carried

out on the benchmark board that are held on bungees and finally section 5.4.2 during the mock field testing. These were as follows;

1. Error in the frequencies that were quoted by the LDV were $\pm 0.8\text{Hz}$.

7.2.4.2 Results

The results from this testing can be seen in Figure 7-14.

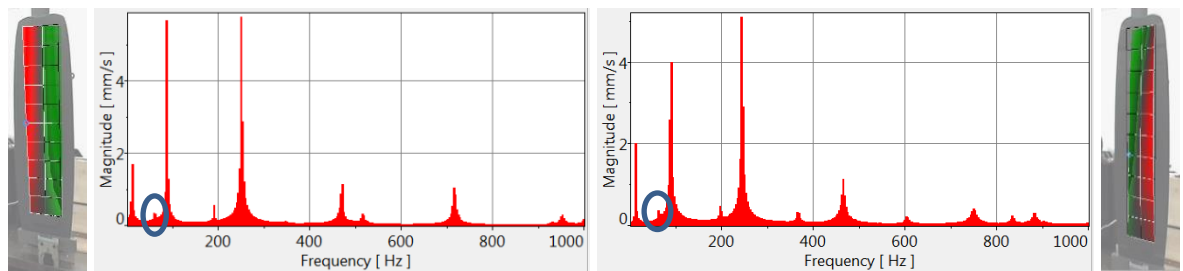


Original Generic Lay-up:

Optimised Lay-up:

1st Bending mode = $12.5 \pm 0.8\text{Hz}$

1st Bending mode = $12.5 \pm 0.8\text{Hz}$



Original Generic Lay-up:

Optimised Lay-up:

1st Torsional mode = $59.4 \pm 0.8\text{Hz}$

1st Torsional mode = $62.5 \pm 0.8\text{Hz}$

Figure 7-14: Comparison of the Vibrational Behavior of the Original & the Optimised Boards.

As can be seen in Figure 7-14 the vibrational response of the original and the optimised boards suggests that these boards would feel the same to the rider.

7.3 Correlation Between the FE Models and the Physical Components

This section now correlates the FE models to the physical components to see if the original models predict their structural behaviour. The simplest, most robust way of

doing this is by looking at the vibrational response. The bending and torsional static stiffness tests introduce many variables that could influence the correlation such as;

1. The stiffness of the boundary conditions.
2. The friction/weight of the DTI gauges.
3. The rolling resistance of the pulley used in the torsional test.
4. The clamping force and coefficient of friction between the loading bar and the board.
5. The position of the loads when they are attached to the loading bar.
6. The position of the DTI gauges.

Due to the above reasons, the vibrational response of the board was tested and correlated to the FE model as discussed in section 4.5. The results of this testing can be seen in Figure 7-15.

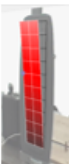
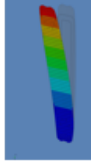
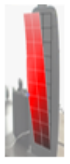
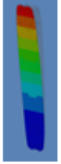
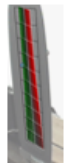
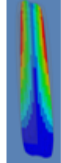
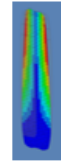
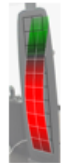
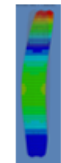
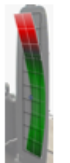
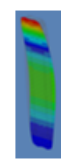
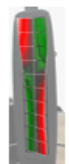
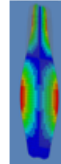
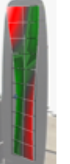
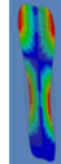
		Lay-up					
		Generic Lay-up			Optimised Lay-up		
		Test	FE model	Δ (%)	Test	FE model	Δ (%)
Mode 1							
	Freq (Hz)	12.5 $\pm$ 0.8	11.5	-8.0	12.5 $\pm$ 0.8	11.3	-9.6
Mode 2							
	Freq (Hz)	59.4 $\pm$ 0.8	51.8	-12.8	62.5 $\pm$ 0.8	53.0	-15.2
Mode 3							
	Freq (Hz)	87.5 $\pm$ 0.8	76.5	-12.6	90.6 $\pm$ 0.8	76.1	-16.0
Mode 4							
	Freq (Hz)	190.6 $\pm$ 0.8	168.9	-11.4	196.9 $\pm$ 0.8	178.5	-9.3

Figure 7-15: Comparison of the Original & the Optimised Scale Boards: NVH Performance.

As can be seen in Figure 7-15, good correlation was seen between the physical boards and the FE models for the first four modes that were tested.

7.4 Conclusions Drawn from the Verification Study

An FE model of the optimised board was created in section 7.1.2 which had the same structural characteristics as the original board. In section 7.2 it was shown that the resulting physical boards had the same structural characteristics as each other.

This verification study has achieved the following;

1. The design methodology was used in section 7.1.2 to establish the construction needed of an optimised component to replicate the structural response of an original structural orthotropic composite component seen in section 7.1.1.
2. The optimised component that was designed in section 7.1.2 was manufactured in section 7.2. This was shown to have the same structural characteristics that are important to the original component that have been discussed in section 2.
3. The numerical models that were created in section 7.1 have been shown to correlate to the physical components that were manufactured in section 7.2.

In addition, the physical component that was manufactured, was manufactured 'right first time', without the need for timely and costly iterations and used less material than the original board so was lighter and would result in lower fuel consumption etc during transportation, as discussed in section 3.

8 Conclusions of the Thesis and Proposed Future Work

8.1 Conclusions of the Thesis

The aim of this project was to demonstrate how, by using modern numerical techniques and in particular, FEA and optimisation methods, it is possible to produce improved products whilst at the same time reducing the development time. This has been achieved through the use of a Design Methodology for the Implementation of Structural Orthotropic Composite Materials in a High-Performance System. It has been demonstrated how the structural performance of an existing structure can be replicated. This methodology however does have limitations. In order to successfully implement numerical methods to drive the design process, one of the most fundamental inputs to these models are the material properties. It has been demonstrated in chapter 4 that this is by no means a trivial task, particularly when dealing with orthotropic materials. There are clearly differences between the values that are published by material manufacturers and values needed for good correlation during physical testing. These differences could be a barrier to OEMs implementing the proposed methodology. After all, if good correlation cannot be achieved with simple structures of a known lay-up, any suggested lay-up as a result of optimisation should be received with low confidence. Having said that, it has been demonstrated in section 4.6 how good material correlation can easily be achieved by simply scaling any given values. It has then been demonstrated how these scaled values can then be used in subsequent numerical models.

In terms of what is desirable, a greater understanding of this particular application has been demonstrated. Clear differences between the desired structural characteristics of boards aimed at different riders has been seen. This has been seen in the bending stiffness, the torsional stiffness and the vibrational response of the boards. Once, what is desirable is known, then this can be replicated using numerical methods.

One of the fundamental issues with reverse engineering a product and then optimising a new component to have the same mechanical characteristics of the original, is that if the original component was over engineered, then the new component will also be over engineered. Rather than measuring what characteristics the original component has, it is better to measure what is needed. Having said that, the components in this study have evolved over many iterations such that it can be assumed that the stiffness distribution seen and the NVH characteristics are those that are desirable to the user.

The fact that the components in this work have evolved over many iterations goes some way to explain why the boards have resulted in modest weight savings, suggesting that the original components are fairly efficient. However, the suggested components that have been designed using the proposed methodology have been proven to result in reasonable correlation both in terms of the numerical models and the representations of the physical boards that have been manufactured. The methodology has proven that a 'right first time' design is possible in terms of the board rather than relying on many costly iterations. However, the costs associated with this development time have not been quantified in this particular application.

A methodology has been proposed to assess this particular application that mimics the use of these components. This has been shown to be a powerful tool to compare boards. Boards that are compared using this technique show differences that can be related to the structural performance of the component. In addition, this technique can be used to assess the board to ensure that it does not fail during use.

The purpose of this study was to show that SMEs should be integrating the tools discussed in this work in their design methodologies. This study has used an Autoclave and an LDV which are prohibitively expensive to most SMEs, however the same approach would be possible using a wet lay-up and Piezo transducers.

Whilst this work has gone some way to demonstrate that OEMs should be implementing the design methodologies that have been used by industries such as Automotive and Aerospace, the statement made by Bevington seems very relevant when he stated that “It is a well established fact of scientific investigation that the first time an experiment is performed the results often bear all too little resemblance to the “truth” being sought” (98).

8.2 Proposed Future Work

In terms of additional work that could and should be carried out as a result of this research, the following could be addressed;

1. Whilst this work has used scale boards that are representative of the structure developed in section 6, it would have been good to manufacture the full scale board, including the rails and the inserts, although this would obviously introduce many more variables as discussed in section 7.

2. Whilst the need for 'mock field' testing has been discussed throughout this work, additional field studies would add to this. The response of the original board and the optimised alternative could then be measured. Traditionally this has been done qualitatively, however further quantitative work such as that carried out by Valsecchi (14) could be carried out using a baseline structure and the structure developed using the proposed methodology. This seems simpler to achieve with recent advancements in data acquisition equipment such as that developed like Woo (131) or Xenser (132).
3. Numerical models could be built to assess the components as they are being used in their environment such as the study that Brennan (26) carried out for Snowboards.
4. The proposed methodology has been demonstrated in a particular application using a particular piece of software. This study should be repeated with different applications and different software.
5. If this exercise were to be repeated, it would be interesting to see this carried out on a highly curved structure so that the influence of draping could be understood.
6. Whilst composite optimisation has been used throughout this work, it would be interesting to see similar studies carried out using different optimisation techniques such as Shape optimisation as detailed in section 3.3.1.
7. The approach to scaling the material properties (in order to correlate vibrational behaviour and obtain material properties) on a simple flat plate detailed in Section 4.6 could be repeated with piezo transducers instead of using an LDV to see if similar results can be obtained cheaply. However, the work by Fuss et al (33) which discussed influence of the mass of the sensors and the cables must

be considered. Design of Experiments (DoE) techniques should be integrated into this procedure.

8. This study could be repeated using wet lay-up instead of an Autoclave.
9. During this study, it has been assumed that there have been no defects introduced into the structures during manufacture. These should be checked for delamination etc.
10. The change in the bending stiffness and the torsional stiffness across the width of the board should be investigated. This was assumed to be constant in section 5.3.1 and section 5.3.2

Appendices

A Copies of Previously Published Work by the Author

A.1 The 8th World Congress on Structural and Multidisciplinary Optimisation, (1)

The Application of FEA in the Optimisation of Die Cast Components & the Consequent Reduction in Development Costs & Time

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1. Abstract

This paper details applications of FEA techniques in the optimisation of large, thin-walled, high-pressure, die cast magnesium alloy components for the automotive industry. Initially the processes of topology, topometry and shape optimisation are discussed before detailing good practice in the use of these techniques in the automotive industry by a first and second tier supplier to automotive OEMs. Finally, the way in which CAE led design fits into the product lifecycle, successfully reducing development times and costs, is discussed and demonstrated.

2. Keywords: Structural Optimisation, Automotive, FEA, GENESIS

3. Introduction

Despite optimisation techniques being well understood, readily available and relatively cheap to implement, they are still underutilised in industry. In 2004 Vanderplaats stated that “In engineering, while development [of optimisation techniques] has been underway for over forty years, applications have lagged far behind. The time has come for that to change” and that “optimization technology has matured to the point where it can and should be used routinely for engineering design”[1]. In a 2004 paper, he reminded us that in 1978 he “predicted that optimization would be commonplace by 1985”, but then went on to say, “Clearly, that did not happen and predictions are no longer made”[2].

With increasing focus on the environment and the cost of engineering design, it is imperative that designs are as efficient as possible. This efficiency can be improved by using CAE to lead design, rather than simply using it as a verification tool. In particular, this can be achieved by using optimisation techniques which “can be used with minimal added cost or effort to significantly reduce energy consumption”[3]. For example, “a ten percent mass reduction will increase fuel economy by about seven percent”[1], and given the current economic climate there are also significant benefits to the economy since a “one percent economy improvement [in fuel consumption] will save nearly three billion dollars per year in the U.S. at the pump”[1].

However, “despite the widespread availability of this technology, it is seldom taught as a design tool by universities and remarkably underutilised by industry”[1]. This is now starting to change, for example, in February 2009 Swansea Metropolitan University became the first university in the U.K. to include the practical application of optimisation tools in GENESIS into one of its courses.

There are however, examples where optimisation is used successfully. This paper summarises some of the FEA techniques that are available and then details the applications of these techniques in the optimisation of large, thin-walled, high-pressure, die cast magnesium alloy components for the automotive industry by a first and second tier supplier to automotive manufacturers. Finally, the way in which CAE led design fits into the product lifecycle, successfully reducing development times and costs, is discussed and demonstrated using this example.

4. FEA Optimisation Processes

In order that the examples given in this paper can be clearly understood, the optimisation techniques that are used in the examples will now be discussed.

4.1. Topology

Topology is used to define where the most efficient structural loadpaths are within a structure, and is a “specialization of material optimization, forcing zero/one answers”[4]. A package volume is needed within which the part must fit and then loadcases are applied to the package volume to determine these loadpaths. This

is achieved by varying the modulus of elasticity and density. As described by Leiva, “In Topology optimization, the design variables correspond to the element volume fractions. The volume fraction designs simultaneously the material properties E (modulus of elasticity) and ρ (density) with the purpose of getting a 0-1 answer to identify the key elements to keep and the rest to discard”[5].

Topology can successfully be used to design the a-surface of a part (i.e. what the main shape of the part needs to be), as well as being used to determine how an existing part needs to be reinforced to be as efficient as possible, for example in a casting, by the use of ribbing.

The results of a topology optimisation are mathematically correct, but are not final solutions, and these need to be interpreted into a manufacturable engineering design. In many ways this is a limitation of topology. As described in 2005 by Schramm “the interpretation of the results is still a mostly manual process”[6].

Topology is generally used as the first step in optimising a design, “topology optimization seldom produces a final part, even though manufacturing constraints are used. This is because topology optimization normally does not include stress and other constraints. However, it does identify load paths and provides a very good starting point for shape and sizing optimization”[1].

4.2. Shape Optimisation

Once the major structural members have been defined, the cross sectional size of those members can be determined by using shape optimisation. Loads are applied to the structure and the physical dimensions of these members are altered to meet the set criteria by using “scale factors of perturbation vectors”[5] as the design variables.

4.3. Topometry

Once the position and size of the major structural members has been established, topometry can then be used to determine how much material is needed throughout the structure to ensure that specific loadcases are met.

Size optimisation is well established where a physical property (for example the thickness of a group of shell elements) is varied. “Sizing optimization has traditionally been the entry point to the use of optimization technology in design. Especially in designs that use a lot of sheet metal this is a useful tool. It is well established and implemented”[6].

However, topometry is significantly different to traditional size optimisation, “Topometry optimization is a generalization of sizing optimization. Unlike size optimization, where all elements associated to a property data entry are designed with the same values, in topometry optimization each element is designed independently”[5]. This can have significant advantages. Size optimisation can work well, for example, as stated above in a fabricated structure. However, in order to run a size optimisation the analyst must group the elements together that will be optimised thus influencing and limiting the results. Topometry “allows performing sizing optimization on each element individually”[5] thus resulting in a more efficient structure. Castings have the huge benefit that a varying wallstock can be applied throughout the structure (as long as certain manufacturing constraints are met), ensuring that extra material need only be added where it is required. Thus topometry results can be integrated well into a cast design.

5. Examples of use by an Automotive Supplier

Structural optimisation in the automotive industry is not a new idea. In 1923, Henry Ford said “Saving even a few pounds of a vehicle’s weight means it could go faster and consume less fuel. Reducing weight involves reducing materials, which in turn, means reducing cost as well”[7]. However, as already stated, optimisation techniques are underutilised. In reality it is difficult to determine the real extent to which optimisation techniques are used, as highlighted by Vanderplaats, due to the fact that “most real commercial problems are proprietary and cannot be published”[1]. Although necessity has meant that use must increase as “New designs are expected to improve performance, meet new stringent weight targets and at the same time they need to be more economical to manufacture”[4]. Examples have been documented elsewhere, such as, a steering knuckle mass which was reduced by “five percent or more with no loss in strength”[2]. Optimisation has been used to “increase the first frequency of a car body by ten percent for the cost of under one percent added mass”[2], and Delphi Automotive describe an automotive dash cross beam in which they “reduce the mass by 33% and

material cost by 39%”[3], however, this paper will highlight the tools, processes used, and the benefits of these for an automotive supplier.

Meridian Lightweight Technologies Inc. are the world leaders in the design, engineering and manufacturing of high quality magnesium die cast components. CAE is used extensively throughout the development of their products and this will now be highlighted as an example of good practice and specific examples will be demonstrated using the optimisation tools that have already been described. The manner in which Meridian Technologies successfully use CAE led designs both in conceptual designs, and in development of these concepts, through to production once the business has been secured is discussed.

Meridian’s components range from the structural part of a car dash board, through to car seats and grille reinforcements used at the front of the vehicle. These components are large, high pressure die castings (typically around 3mm wall thickness, the width of a car and 5kg in weight), which lend themselves to topology, topometry and shape optimisation. A customer may have defined the a-surface of a component and then require the development of that structure in magnesium so that it meets the customer’s structural targets. In this case topology is used to define the ribbing required before topometry is used to define the required wallstock of the structure. Alternatively, a customer may come to Meridian with a package space and loading requirements for a structure, in which case, topology is used initially to determine the a-surface shape of component. Once this has been established, topology can be used to determine the ribbing needed, shape optimisation is used to determine the final dimensions of the part and finally topometry determines the required wall thickness of the castings.

These components are optimised to ensure no plastic deformation occurs in linear loadcases and there is no failure in non-linear loadcases. In addition, Meridian’s components are required to contribute to the vehicles in ways such as global stiffness and crashworthiness as well as achieving certain local stiffness requirements such as providing stable platforms for the steering wheel assembly and the airbags, whilst ensuring the designs are as light and cheap to produce as possible.

Meridian’s products are structurally quite complex and thus utilise many different types of optimisation tools where appropriate. Leiva supports this idea when discussing topology and topometry “a complex structure can benefit from both [types of optimisation], as some parts of it could only be designed by one type of optimization, whereas other parts could only be designed with the other type”[4].

5.1. Conceptual Design Example 1 – A Rear Cross Car Beam

A structure was required for a coupe convertible which had the following functions; Provide hinge points for the folding roof, provide attachment points for the roll over hoops and to fit into the design volume provided by the customer. The customer provided “designable” regions, i.e. the package space that was available and “non-designable” regions, i.e. the attachment points for the b-pillars, the floor, the hinge points, and the roll over hoops. These regions can be seen in Figure 1.



Figure 1: Non-designable and designable regions

These two regions were then combined to form the complete package. This was then used as the base for the initial topology run. A topology run was set up with two loadcases that represented the rollover of the vehicle and the roof opening. There were two requirements for these loadcases, one, that the structure did not fail in the rollover loadcase, and two, that there was no fatigue failure in the roof opening loadcase. To ensure that no failure occurred in the rollover loadcase the amount of plastic strain seen by the magnesium was limited below a certain percentage, and to ensure no fatigue failure in the roof opening loadcase the stress seen in the magnesium was also limited.

Optimisation techniques are currently available only in linear software (although coupling of optimisation with non-linear software is now available in software such as GENESIS). Hence to define the loadpaths in this design the non-linear rollover loadcase was approximated to a linear loadcase in the topology run.

A topology run was set up with the linear interpretation of the rollover loadcase, and a linear roof opening loadcase with the designable region as per Figure 1 with the objective to minimise strain energy in the structure and the constraint of only using 30% of the material by mass. The results of this initial topology and the interpretation can be seen in Figure 2.

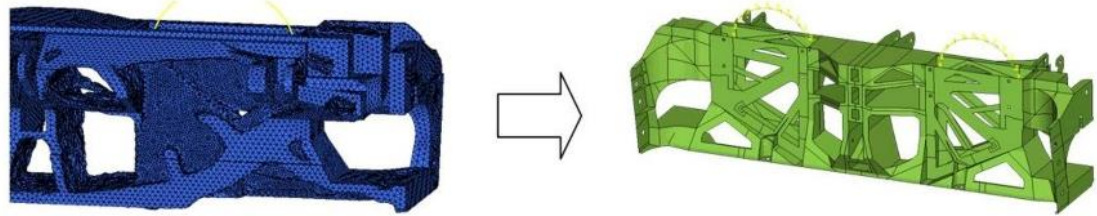


Figure 2 – Topology results for the rear cross car beam

Once the loadpaths from the topology run were interpreted into a castable design a shape optimisation study was then carried out to define the size of the sections in this structure, the result of which is seen in Figure 3.

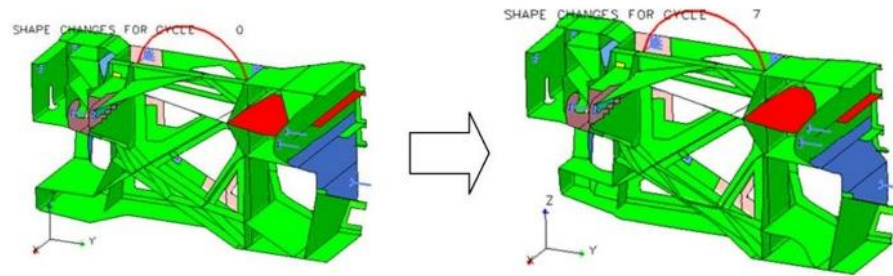


Figure 3 – Shape optimization results

Finally, once the position and the size of the sections were established a topometry optimisation was then carried out to determine the required thickness of the wallstock whilst meeting the stress constraints. The minimum wallstock that could be cast was 2.3mm. Thus this was set as the minimum and the structure was then optimised to minimum mass. The results and interpretation of this can be seen in Figure 4.

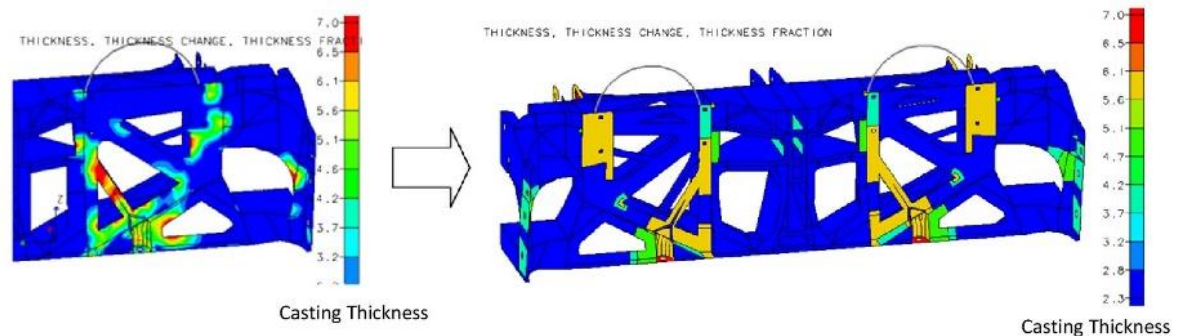


Figure 4 – Topometry results

To ensure that that the design met the two loadcases a linear verification model was then set up to ensure that the stress limit was not exceeded in the roof opening loadcase and a non-linear model was set up to ensure that the plastic strain was not exceeded in the rollover. The results of these two runs can be seen in Figure 5.

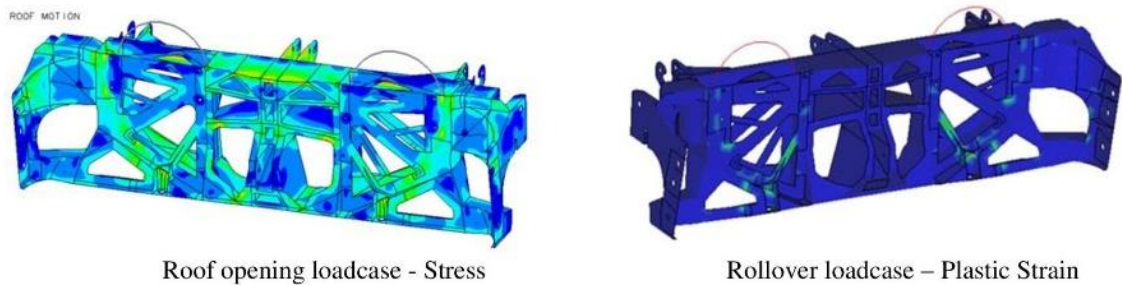


Figure 5 – Verification runs

As can be seen in the roof opening loadcase the majority of the structure is stressed suggesting a very efficient structure, whilst in the roll over loadcase the diagonal members that were upgauged by the topometry run are clearly strained.

The final CAE model is seen in Figure 6. This can be delivered to the customer to prove the structural performance of the component and additionally a surface CAD model of the part can be produced within approximately 10 minutes from the CAE model which can also be delivered to the customer for an initial package check.

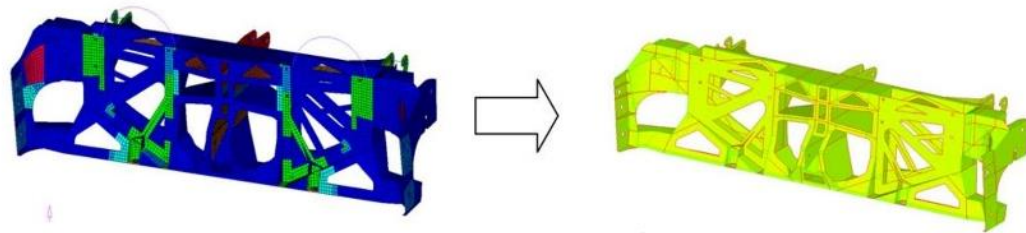


Figure 6 – The final CAE model and surface CAD model

5.2. Conceptual Design Example 2 – A Front End Carrier

In this example the customer had defined the a-surface of the component but required Meridian to develop the structural performance of the component. The a-surface model of this component as supplied is shown in Figure 7. This structure bolts onto the vehicle longitudinals.



This structure has the following functions;

- Support the headlamps, horn and radiator
- Provide support to the bonnet
- Provide vehicle torsional stiffness
- Withstand low speed impacts

Figure 7 – Model supplied to Meridian

In addition an NVH target was set for the structure. A topology run was set up with multiple loadcases and the back of the structure was filled with designable material to define where the major loadpaths were and thus where the ribbing should be. The requirement was once again set up to minimise strain energy whilst only using 30% of the available material, and a manufacturing constraint was set up so that the ribs could be cast in the longitudinal direction. The results of this topology run and the interpretation of these results into a rib pattern can be seen in Figure 8.

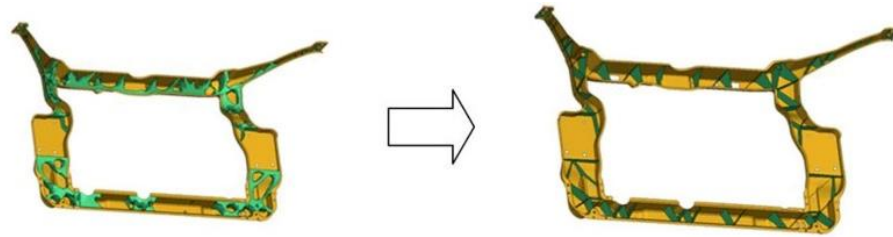


Figure 8 – Results of the topology run and their interpretation into a rib pattern

Once the ribbing had been defined, a topometry run was then set up to determine the thickness of the wallstock throughout the structure, as demonstrated in the previous example.

5.3. From Concept to Production

The previous two examples shown were both examples of conceptual designs used by Meridian to try to secure future business. Once the business has been secured the same optimisation techniques are then used to ensure that the final design is as efficient as possible. A good example of this was the R56 beam as produced by Meridian for the second generation BMW mini. This is shown in Figure 9.



Figure 9 – The R56 cross car beam

This design was an example where the a-surface was provided by the customer to Meridian who then developed the structural performance of the beam for several loadcases including; providing a stiff platform for the steering wheel, providing support to the passenger airbag, providing a structural loadpath in side impact and not breaking in frontal impact. The original topology runs carried out by Meridian that led to the initial CAE concept can be seen in Figure 10, this can then be compared to the final CAD model. The major loadpaths in the original Topology run can clearly be seen. The early use of topology ensures that ribbing is added in the most efficient way and only where necessary. This can be compared to a previous generation Meridian beam (which did not utilise optimisation techniques) as seen in Figure 11 where ribbing was simply added in an 'X' structure throughout the beam to make the beam as stiff as possible in all directions, regardless of where the ribbing was actually needed, resulting in a beam not being as efficient as it could be.

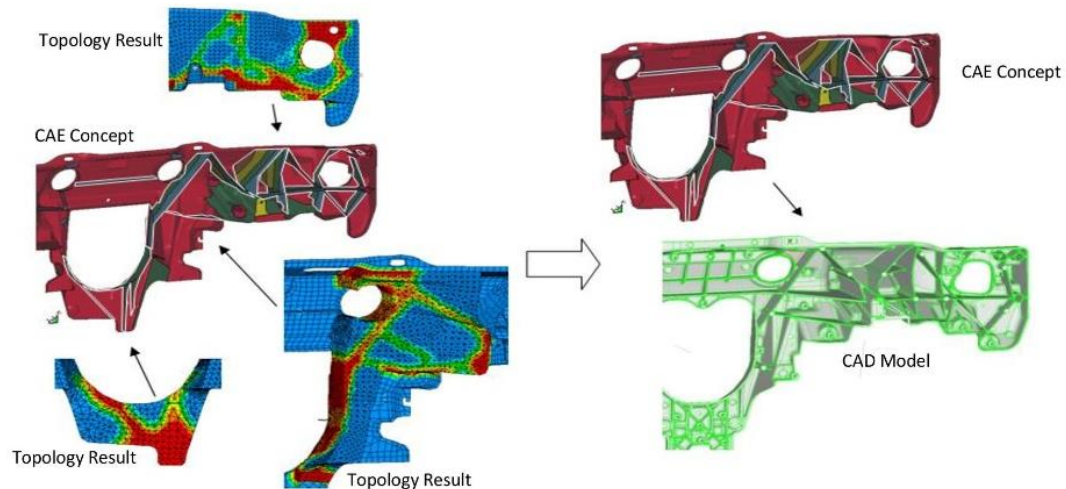


Figure 10 – Initial topology runs done at concept through to final production



Figure 11 – Previous generation Meridian beam

Throughout development, it is inevitable that as the loadcases and package matures that the design will need to change. When this happens the same optimisation tools can and should be used. One such iteration is now discussed as an example of this.

The R56 beam in an interim stage of development was 4.81kg and met all the required loadcases. An optimisation sweep was then carried out on this beam using the following steps; local topology runs were performed to redefine specific areas of ribbing as shown in Figure 12, a topometry run was performed to highlight which areas needed to be reinforced or could be reduced in section, and finally, a topometry run was performed to highlight any areas where mass-saving holes could be introduced, as seen in Figure 13.



Figure 12 – An example of a local topology run

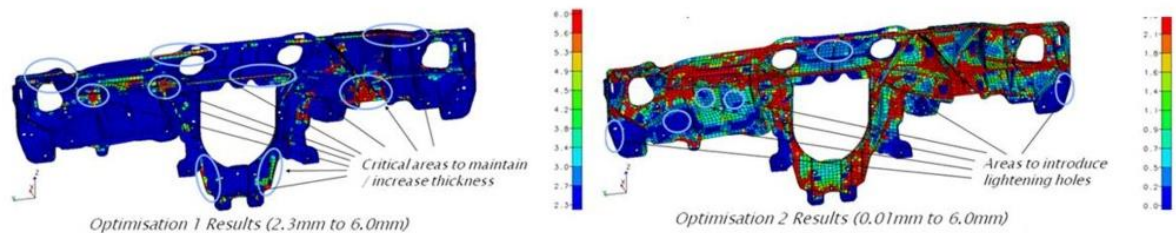


Figure 13 – Topometry runs

The resulting beam still met all of the required loadcases but the mass was reduced to 4.49kg. Thus mass was reduced by nearly 7% whilst crash performance and NVH performance remained unchanged.

6. CAE Led Design in the Product Lifecycle

The use of CAE early on in the development of Meridian's products has significant effects on the product lifecycle in terms of reducing development time and costs, of both concepts and final designs, especially when compared to a more traditional CAD led approach. This is particularly important when developing concepts, since it is becoming more and more common for OEMs to expect suppliers to offer conceptual designs when quoting for business, carried out at the suppliers' risk. Also, once the business has been secured, by using these techniques, development times are reduced, helping OEMs to get products to market quicker and the final designs will be as light and cheap as possible. All of this, results in cost savings for both the supplier, the OEM and ultimately the final consumer.

These benefits are a direct result of the early use of CAE optimisation. As stated by Vanderplaats "Optimization will improve the quality or reduce design for any system, subsystem or component to which it is applied" [2]. All of this is achieved at very little expense as the "benefit will exceed the cost" [2] and give companies who use optimisation tools, a competitive edge over companies that do not employ such techniques as "the companies that strongly embrace this technology will have a clear competitive edge for the betterment of all" [2].

To fully understand this we will now compare a traditional CAD led design to the CAE led designs that are carried out by Meridian in Figure 14.

CAD Led Design v CAE Led design in the Design Process

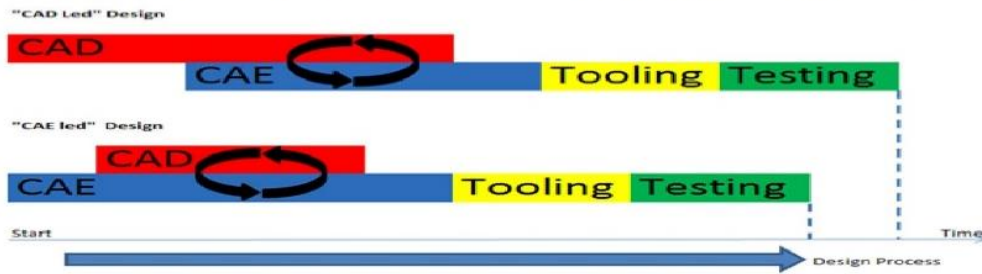


Figure 14 – Comparison of a CAD led design to a CAE led design

In a traditional CAD led design, a concept is first produced in a 3D CAD package. Once an initial design has been developed, CAE is used as a verification tool, as Vanderplaats states, “we normally use computer analysis to judge the quality of our designs”[3], not to actually design the parts. Results of this analysis are fed back to the designer, the CAD is updated and a new CAE model is built and this process is repeated until a final design is reached and verified. Once the structural performance of the design has been proved, prototype tooling is produced, and finally prototypes are then cast and tested. Although the use of CAE in judging designs has significantly reduced the product lifecycle time by minimising the amount of prototypes needed, as supported by Schramm when he wrote that “Physical experiment has been widely reduced or eliminated to be replaced by computational methods”[6] there are however, significant opportunities to further reduce the time taken to develop a design by utilising optimisation techniques.

Leiva summarises CAD led designs when he says “The design of many of these objects has been done over the years based on judgement, experience of history. Structural analysis to study the behaviour of parts has only been used in the last few decades, optimization has been used even less”[8], and it is by introducing optimisation techniques that the change from CAD led designs to CAE led designs can be made. When describing current design processes Schramm stated that “Simulation can now keep up with design changes and therefore drive design decisions”[6], but by introducing optimisation techniques, the CAE model should not only keep up with design changes, but should actually lead the design.

In a CAE led design, such as the type used by Meridian, the design process can be summarised as follows. The initial design is developed in CAE using optimisation techniques thus ensuring the major loadpaths are used. From this a CAE model can be built and hence the design can be verified - all of this can be achieved before any CAD needs to be created. Once an initial concept has been created and proved, CAD can then be produced. Inevitably there will still be iterations once the initial concept has been developed as the design matures. There is still, therefore, a feedback loop. This results in a final design that is verified before prototype tools are cut and prototype parts are produced and tested. As can be seen in Figure 14 this results in a shortening of the design process time.

The benefits of CAE led concepts also give significant benefits to Meridian. To demonstrate this we will consider the concept design process, in order to show the impact on timing and cost. To do this, we will make the following assumptions;

- Timing Assumptions
 - The time taken to produce the initial CAD in a 3D CAD package takes 6 man weeks which is typical given the complex nature of Meridian’s parts which have draft, fillets and ejector bosses etc.
 - To produce an initial concept in the CAE environment including a topology study, topometry study and structural assessment, takes 2 man weeks
 - To carry out a structural assessment of a design takes 1 man week
 - A CAD led approach requires 2 iterations to achieve acceptable structural performance
- Cost Assumptions
 - We will assume that for these concepts all CAD and CAE is outsourced
 - The hourly rate of a CAE analyst is £45/hour (based on 2007 Prices)
 - The hourly rate of a CAD analyst is £35/hour (based on 2007 prices)
 - A working week is 40 hours

Using these assumptions we can now compare the cost and timings of a CAD led design and a CAE led design as seen in Figure 15.

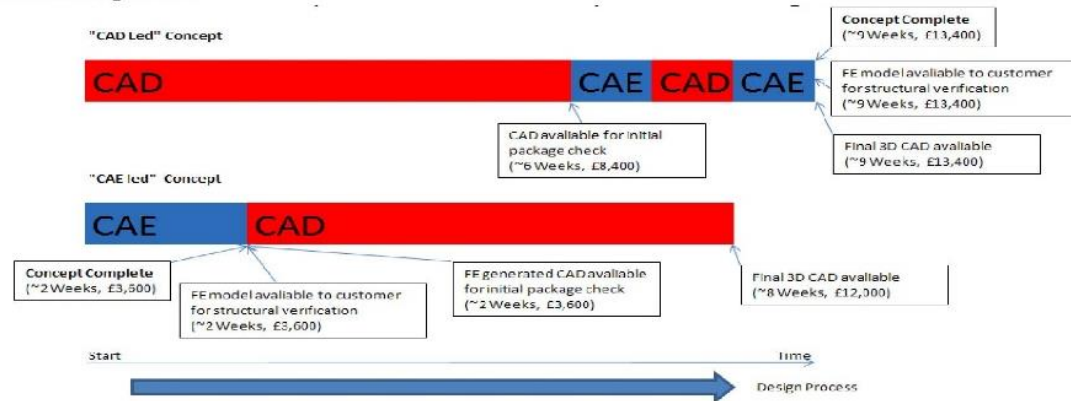


Figure 15 – CAD led concept v CAE led concept in design process

In the CAD led concept, initially a 3D CAD model is produced which takes 6 man weeks and costs £8,400. Once this has been done, initial CAD can be provided to the customer for an early package check. This design is then assessed in CAE and then goes through an additional design iteration before the concept is complete resulting in the following (Note that most of the deliverables are at the end of the design process);

- CAD is available for an initial package check after 6 man weeks and £8,400
- The Final FE model is available to prove the structural performance after 9 man weeks and £13,400
- The Final 3D CAD is available and the concept is complete after 9 man weeks and £13,400
- The geometry and weight of the component is known, and can thus be costed, after 9 man weeks at a cost of £13,400

In comparison, the CAE led design creates a concept purely in the CAE environment within two man weeks at a cost of £3,600. From this a simple surface CAD model can be produced and delivered to the customer for an initial package check. In addition, the FE model can be delivered to the customer to prove the structural performance of the structure. Once the customer has seen this concept and if, and only if, they are still seriously interested in the concept, then the expense and time of producing a 3D CAD part is incurred, resulting in the following deliverables (Note the deliverables are now much earlier in the design process);

- The initial concept is complete after 2 man weeks at a cost of £3,600
- The FE model is available to the customer to prove the structural performance of the concept after 2 man weeks at a cost of £3,600
- A simple surface model is available to the customer for an initial package check after 2 man weeks at a cost of £3,600
- The geometry and weight of the component is known, and can thus be costed, after 2 man weeks at a cost of £3,600
- The Final 3D CAD can be delivered to the customer after 8 man weeks at a cost of £12,000.

As can be seen in Figure 15, by using CAE led design, as opposed to CAD led design, Meridian has achieved the following; Conceptual designs are delivered with reduced cost and timing by approx 75% when 3D CAD is not required, and 10% when 3D CAD is required. The CAE led approach is likely to result in lighter and cheaper conceptual designs, and is therefore more likely to secure new business. Finally, production designs are likely to be lighter, cheaper and more efficient.

In addition to the quantifiable benefits highlighted above, there are also the unquantifiable benefits of improved customer perception of Meridian Technologies advanced engineering capabilities, improved customer confidence in Business Development quotes when quoting for business, and improved customer perception of magnesium and its capabilities.

In a year when Meridian are aggressively pursuing new business they may undertake up to 30 conceptual studies and of these, approximately 10% would be of enough interest to potential customers to warrant full 3D CAD to be generated. Thus these CAE led concepts would cost as follows;

CAE studies	30 x £3,600	= £108,000
3D CAD	0.1 x 30 x £8,400	= <u>£25,200</u>
TOTAL		£133,200

If these studies were all traditional CAD led designs then the cost would be as follows;

CAD led studies 30 x £13,400 = **£402,000**

Thus a direct saving of over 66% or £268,800 can directly be attributed to using optimisation techniques. This is a significant saving, particularly as these concepts are nearly entirely unfunded by the customer.

7. Conclusion

This paper has used Meridian as an example to show how optimisation can be routinely used in the design process. Furthermore, based on Meridian's successful implementation of the tools of topology, shape optimisation and topometry into their design process, the paper argues that optimisation is not simply an option to consider as part of the design process; rather, the evidence presented suggests that it is a necessity, given the benefits it produces in terms of cost, productivity and, ultimately, environmental benefits such as reduced materials usage and fuel consumption. These tools are used right from the start of the design process for conceptual designs and then throughout the development of any products right through to production. It has been shown that these tools reduce the time taken to produce fully matured designs.

When considering conceptual designs, costs are reduced by up to 75%, which is particularly relevant since OEMs are increasingly expecting suppliers to provide conceptual designs at their own cost. These conceptual designs result in faster generation of more efficient and lighter products, which are more likely to result in securing new business.

The optimisation tools used by Meridian are routinely available, yet remain remarkably underutilised. As Vanderplaats argues, "the key to their use is that we must change the way we do design. The changes are not big, only necessary"[3]. This paper has attempted to demonstrate the validity of Vanderplaats' argument and has illustrated both the manner in which optimisation may be used to drive the design process, and the benefits that accrue as a result.

8. References

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An Example of the use of optimisation in the development of high pressure die cast components & the resulting quantified cost & time benefits

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1. Abstract

This paper details an application of FEA techniques in the optimisation of large, thin-walled, high-pressure, die cast magnesium alloy components for the automotive industry. Initially the processes of topology, topometry & shape optimisation are discussed before detailing good practice in the use of these techniques in the automotive industry by a first & second tier supplier to automotive OEMs. Finally, the way in which CAE led design fits into the product lifecycle, successfully reducing development times & costs, is discussed & demonstrated.

2. Keywords: Structural Optimisation, Automotive, FEA, GENESIS, Die Casting

3. Introduction

Despite optimisation techniques being well understood, readily available & relatively cheap to implement, they are still underutilised in industry. In 2004 Vanderplaats stated that “In engineering, while development [of optimisation techniques] has been underway for over forty years, applications have lagged far behind. The time has come for that to change” & that “optimization technology has matured to the point where it can & should be used routinely for engineering design”[1].

However, “despite the widespread availability of this technology, it is seldom taught as a design tool by universities & remarkably underutilised by industry”[1]. This is now starting to change, for example, in February 2009 Swansea Metropolitan University became the first university in the U.K. to include the practical application of optimisation tools in GENESIS into one of its courses.

This paper summarises some of the FEA techniques that are available & then details an application of these techniques in the optimisation of large, thin-walled, high-pressure, die cast magnesium alloy components for the automotive industry by a first & second tier supplier to automotive manufacturers. Finally, the way in which CAE led design fits into the product lifecycle, successfully reducing development times & costs, is discussed & demonstrated using this example.

4. FEA Optimisation Processes

4.1. Topology

Topology is used to define where the most efficient structural loadpaths are within a structure, & is a “specialization of material optimization, forcing zero/one answers”[2]. A package volume is needed within which the part must fit & then loadcases are applied

to the package volume to determine these loadpaths. This is achieved by varying the modulus of elasticity & density, and removing appropriate elements.

4.2. Shape Optimisation

Once the major structural members have been defined, the cross sectional size of those members can be determined by using shape optimisation. Loads are applied to the structure & the physical dimensions of these members are altered to meet the set criteria by using “scale factors of perturbation vectors”[3] as the design variables.

4.3. Topometry

Once the position & size of the major structural members has been established, topometry can then be used to determine how much material is needed throughout the structure to ensure that specific loadcases are met.

Topometry is significantly different to traditional size optimisation, “Topometry optimization is a generalization of sizing optimization. Unlike size optimization, where all elements associated to a property data entry are designed with the same values, in topometry optimization each element is designed independently”[3]. Castings have the huge benefit that a varying wallstock can be applied throughout the structure, ensuring that extra material need only be added where it is required.

5. Example of use by an Automotive Supplier

Structural optimisation in the automotive industry is not a new idea. In 1923, Henry Ford said “Saving even a few pounds of a vehicle’s weight means it could go faster & consume less fuel. Reducing weight involves reducing materials, which in turn, means reducing cost as well”[4]. This paper demonstrates through the use of an automotive supplier the real world application of optimisation techniques & then quantifies the benefits to this supplier.

Meridian Lightweight Technologies Inc. design, engineer & manufacture high quality magnesium die cast components. CAE is used extensively throughout the development of their products & this will now be highlighted as an example of good practice & a specific example will be demonstrated using the optimisation tools described.

Meridian’s components range from the structural part of a car dash board, through to car seats & grille reinforcements used at the front of the vehicle. These components are large, high pressure die castings, which lend themselves to topology, topometry & shape optimisation. Meridian’s products are structurally quite complex & thus utilise many different types of optimisation tools where appropriate.

A customer may have defined the a-surface of a component & then require the development of that structure in magnesium so that it meets the customer’s structural targets. In this case topology is used to define the ribbing required before topometry is used to define the required wallstock of the structure. Alternatively, a customer may come to Meridian with a package space & loading requirements for a structure, in which case, topology is used initially to determine the a-surface shape of component. Once this has been established, topology can be used to determine the ribbing needed, shape optimisation is used to determine the final dimensions of the part & finally topometry

determines the required wall thickness of the castings. The following example is a demonstration of this second approach.

A structure was required for a coupe convertible which had the following functions; Provide hinge points for the folding roof, provide attachment points for the roll over hoops & to fit into the design volume provided by the customer. The customer provided “designable” regions, i.e. the package space that was available & “non-designable” regions, i.e. the attachment points for the b-pillars, the floor, the hinge points, & the roll over hoops. These regions can be seen in Figure 1.



Figure 1: Non-designable & designable regions

These two regions were then combined to form the complete package. This was then used as the base for the initial topology run. A topology run was set up with two loadcases that represented the rollover of the vehicle & the roof opening. There were two requirements for these loadcases, one, that the structure did not fail in the rollover loadcase, & two, that there was no fatigue failure in the roof opening loadcase. To ensure that no failure occurred in the rollover loadcase the amount of plastic strain seen by the magnesium was limited below a certain percentage, & to ensure no fatigue failure in the roof opening loadcase the stress seen in the magnesium was also limited.

A topology run was set up with the linear interpretation of the rollover loadcase, & a linear roof opening loadcase with the designable region as per Figure 1 with the objective to minimise strain energy in the structure & the constraint of only using 30% of the material by mass. The results of this initial topology & the interpretation can be seen in Figure 2. (animation can be seen [here](#))

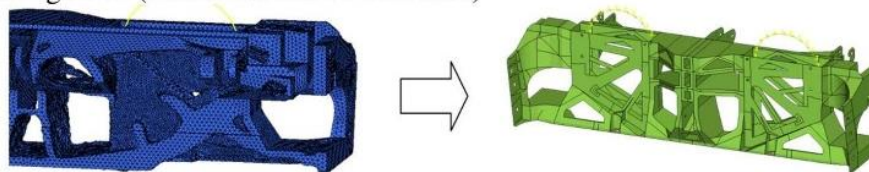


Figure 2 – Topology results for the rear cross car beam

Once the loadpaths from the topology run were interpreted into a castable design a shape optimisation study was then carried out to define the size of the sections in this structure, the result of which is seen in Figure 3. (animation can be seen [here](#))

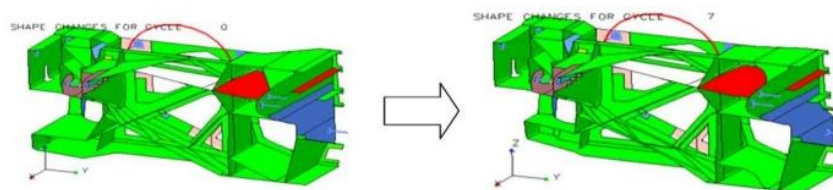


Figure 3 – Shape optimization results

Finally, once the position & the size of the sections were established a topometry optimisation was then carried out to determine the required thickness of the wallstock whilst meeting the stress constraints. The minimum wallstock that could be cast was 2.3mm. Thus this was set as the minimum & the structure was then optimised to minimum mass. The results & interpretation of this can be seen in Figure 4. (animation can be seen [here](#))

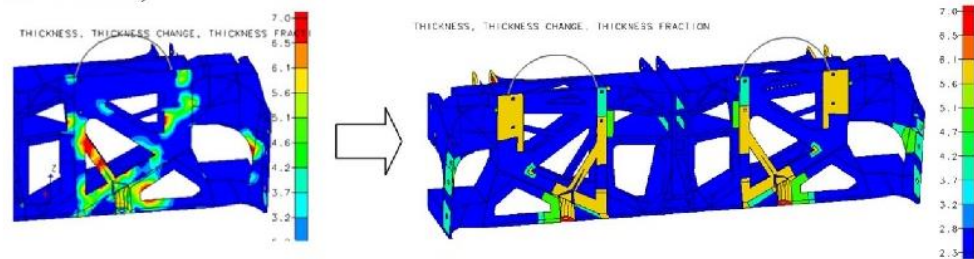
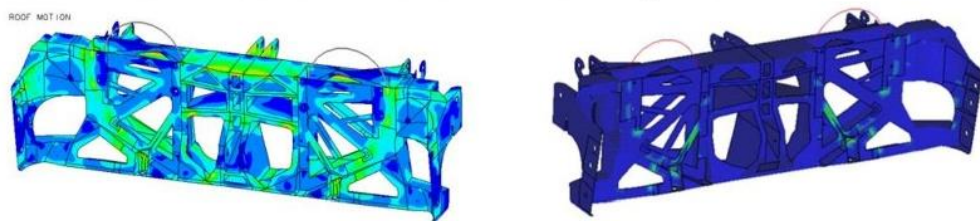


Figure 4 – Topometry results

To ensure that that the design met the two loadcases a linear verification model was then set up to ensure that the stress limit was not exceeded in the roof opening loadcase & a non-linear model was set up to ensure that the plastic strain was not exceeded in the rollover. The results of these two runs can be seen in Figure 5.



Roof opening loadcase - Stress

Rollover loadcase – Plastic Strain

Figure 5 – Verification runs

As can be seen in the roof opening loadcase the majority of the structure is stressed suggesting a very efficient structure, whilst in the roll over loadcase the diagonal members that were upgauged by the topometry run are clearly strained.

The final CAE model is seen in Figure 6. This can be delivered to the customer to prove the structural performance of the component & additionally a surface CAD model of the part can be produced within approximately 10 minutes from the CAE model which can also be delivered to the customer for an initial package check.

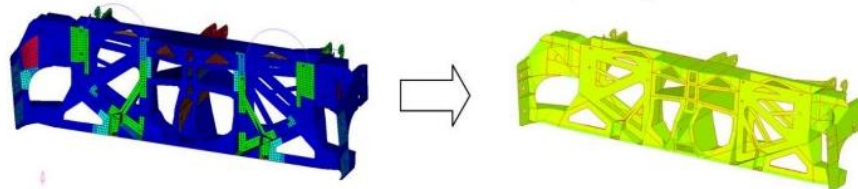


Figure 6 – The final CAE model & surface CAD model

6. CAE Led Design in the Product Lifecycle

The use of CAE early on in the development of Meridian's products has significant effects on the product lifecycle in terms of reducing development time & costs, of both concepts & final designs, especially when compared to a more traditional CAD led

approach. This is particularly important when developing concepts, since it is becoming more & more common for OEMs to expect suppliers to offer conceptual designs when quoting for business, carried out at the suppliers' risk. Also, once the business has been secured, by using these techniques, development times are reduced, helping OEMs to get products to market quicker & the final designs will be as light & cheap as possible. All of this, results in cost savings for both the supplier, the OEM & ultimately the final consumer.

These benefits are a direct result of the early use of CAE optimisation. To fully understand this we will now compare a traditional CAD led design to the CAE led designs that are carried out by Meridian in Figure 7.

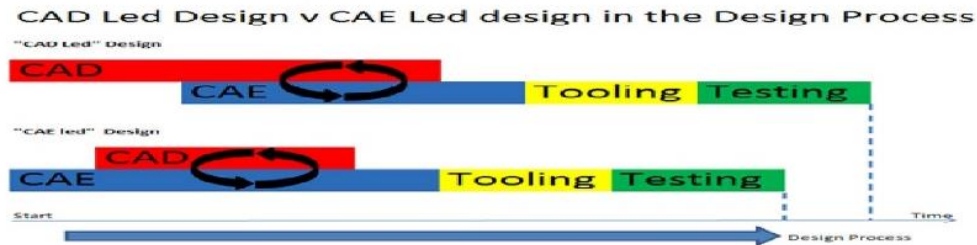


Figure 7 – Comparison of a CAD led design to a CAE led design

In a traditional CAD led design, a concept is first produced in a 3D CAD package. Once an initial design has been developed, CAE is used as a verification tool, as Vanderplaats states, “we normally use computer analysis to judge the quality of our designs”[5], not to actually design the parts. Results of this analysis are fed back to the designer, the CAD is updated & a new CAE model is built & this process is repeated until a final design is reached & verified. Once the structural performance of the design has been proved, prototype tooling is produced, & finally prototypes are then cast & tested. Although the use of CAE in judging designs has significantly reduced the product lifecycle time by minimising the amount of prototypes needed, as supported by Schramm when he wrote that “Physical experiment has been widely reduced or eliminated to be replaced by computational methods”[6] there are however, significant opportunities to further reduce the time taken to develop a design by utilising optimisation techniques.

Leiva summarises CAD led designs when he says “The design of many of these objects has been done over the years based on judgement, experience of history. Structural analysis to study the behaviour of parts has only been used in the last few decades, optimization has been used even less”[7], & it is by introducing optimisation techniques that the change from CAD led designs to CAE led designs can be made. When describing current design processes Schramm stated that “Simulation can now keep up with design changes & therefore drive design decisions”[6], but by introducing optimisation techniques, the CAE model should not only keep up with design changes, but should actually lead the design.

In a CAE led design, such as the type used by Meridian, the design process can be summarised as follows. The initial design is developed in CAE using optimisation techniques thus ensuring the major loadpaths are used. From this a CAE model can be

built & hence the design can be verified - all of this can be achieved before any CAD needs to be created. Once an initial concept has been created & proved, CAD can then be produced. Inevitably there will still be iterations once the initial concept has been developed as the design matures. There is still, therefore, a feedback loop. This results in a final design that is verified before prototype tools are cut & prototype parts are produced & tested. As can be seen in Figure 7 this results in a shortening of the design process time.

The benefits of CAE led concepts also give significant benefits to Meridian. To demonstrate this we will consider the concept design process, in order to show the impact on timing & cost. To do this, we will make the following assumptions;

- Timing Assumptions
 - The time taken to produce the initial CAD in a 3D CAD package takes 6 man weeks which is typical given the complex nature of Meridian's parts which have draft, fillets & ejector bosses etc.
 - To produce an initial concept in the CAE environment including a topology study, topometry study & structural assessment, takes 2 man weeks
 - To carry out a structural assessment of a design takes 1 man week
 - A CAD led approach requires 2 iterations for acceptable structural performance
- Cost Assumptions
 - We will assume that for these concepts all CAD & CAE is outsourced
 - The hourly rate of a CAE analyst is £45/hour (based on 2007 Prices)
 - The hourly rate of a CAD analyst is £35/hour (based on 2007 prices)
 - A working week is 40 hours

Using these assumptions we can now compare the cost & timings of a CAD led design & a CAE led design as seen in Figure 8.

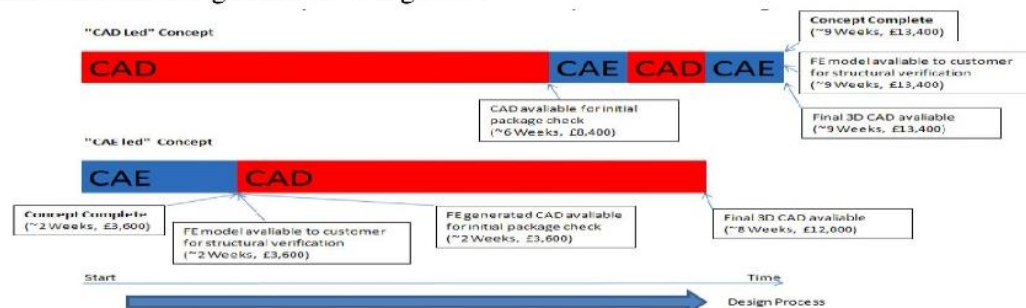


Figure 8 – CAD led concept v CAE led concept in design process

In the CAD led concept, initially a 3D CAD model is produced which takes 6 man weeks & costs £8,400. Once this has been done, initial CAD can be provided to the customer for an early package check. This design is then assessed in CAE & then goes through an additional design iteration before the concept is complete resulting in the following (Note that most of the deliverables are at the end of the design process);

- CAD is available for an initial package check after 6 man weeks & £8,400
- Proof of the structural performance after 9 man weeks & £13,400
- Final 3D CAD is available & the concept is complete after 9 man weeks & £13,400

- The geometry & weight of the component is known, & can thus be costed, after 9 man weeks at a cost of £13,400

In comparison, the CAE led design creates a concept purely in the CAE environment within two man weeks at a cost of £3,600. From this a simple surface CAD model can be produced & delivered to the customer for an initial package check. In addition, the FE model can be delivered to the customer to prove the structural performance of the structure. Once the customer has seen this concept & if, & only if, they are still seriously interested in the concept, then the expense & time of producing a 3D CAD part is incurred, resulting in the following deliverables (Note the deliverables are now much earlier in the design process);

- The initial concept is complete after 2 man weeks at a cost of £3,600
- The FE model is available to the customer to prove the structural performance of the concept after 2 man weeks at a cost of £3,600
- A simple surface model is available to the customer for an initial package check after 2 man weeks at a cost of £3,600
- The geometry & weight of the component is known, & can thus be costed, after 2 man weeks at a cost of £3,600
- The Final 3D CAD can be delivered after 8 man weeks at a cost of £12,000.

As can be seen in Figure 8, by using CAE led design, as opposed to CAD led design, Meridian has achieved the following; Conceptual designs are delivered with reduced cost & timing by approx 75% when 3D CAD is not required, & 10% when 3D CAD is required. The CAE led approach is likely to result in lighter & cheaper conceptual designs, & is therefore more likely to secure new business. Finally, production designs are likely to be lighter, cheaper & more efficient.

In addition to the quantifiable benefits highlighted above, there are also the unquantifiable benefits of improved customer perception of Meridian Technologies advanced engineering capabilities, improved customer confidence in Business Development quotes when quoting for business, & improved customer perception of magnesium & its capabilities.

In a year when Meridian are aggressively pursuing new business they may undertake up to 30 conceptual studies & of these, approximately 10% would be of enough interest to potential customers to warrant full 3D CAD to be generated. Thus these CAE led concepts would cost as follows;

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TOTAL		£133,200

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Thus a direct saving of over 66% or £268,800 can directly be attributed to using optimisation techniques. This is a significant saving, particularly as these concepts are nearly entirely unfunded by the customer.

7. Conclusion

This paper has used Meridian as an example to show how optimisation can be routinely used in the design process. Furthermore, based on Meridian's successful implementation of the tools of topology, shape optimisation & topometry into their design process, the paper argues that optimisation is not simply an option to consider as part of the design process; rather, the evidence presented suggests that it is a necessity, given the benefits it produces in terms of cost, productivity &, ultimately, environmental benefits such as reduced materials usage & fuel consumption. These tools are used right from the start of the design process for conceptual designs & then throughout the development of any products right through to production. It has been shown that these tools reduce the time taken to produce fully matured designs.

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B Rationale for Quoting Standard Deviation

In section 4.5, the standard deviation has been quoted for all of the tested vibrational performance of the plates that have been tested. The reason for this is as follows. The statistical variation of these properties can be expected to vary as a normal distribution. The standard deviation shows how ‘spread out’ this normal distribution is. If the standard deviation is high, this would suggest that you would expect large differences between similar tests. If the standard deviation is low, this would suggest that you would expect small differences between similar tests. This can be seen in Figure B-1 which has been reproduced from the work by Galarnyk (133).

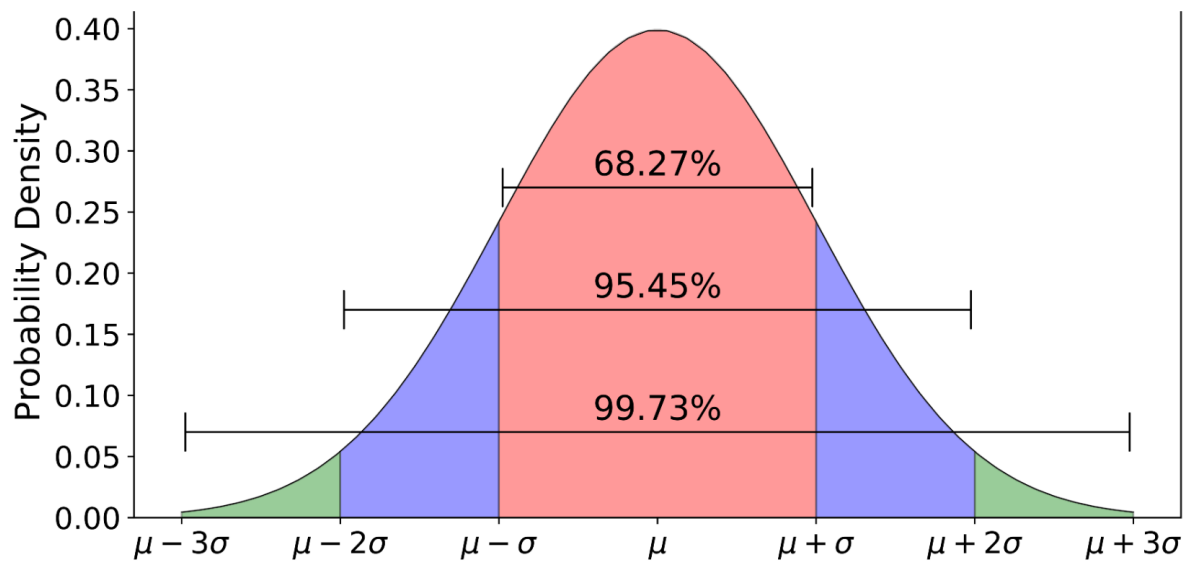


Figure B-1: A Normal Distribution with Standard Deviation Shown (133).

In Figure B-1, μ represents the mean value and σ represents the standard deviation. It can be seen that 68.3% of data is within $\pm\sigma$, 95.5% of data is within $\pm2\sigma$ and 99.7% of data is within $\pm3\sigma$. Standard deviation is therefore quoted alongside the mean values for any of the tested properties to demonstrate how ‘spread out’ this data is.

C Rationale for the Order of Curves During Curve Fitting

In section 4.6 a 5th order polynomial curve was fitted when scaling the material properties used. The percentage differences between the FE model and the test results, along with the percentage differences between the curve that has been fitted to the FE models and the test results is seen in Table C-1 and Table C-2. The sum of the errors is then plotted by polynomial order in Figure C-1 and Figure C-2.

[0]8 mode 1 difference (%)									
E1 (Gpa)	FE model	2nd order	Delta	3rd order	Delta	4th order	Delta	5th order	Delta
8	-39.1	-38.0	1.14	-39.0	0.14	-39.1	0.02	-39.1	0.01
18	-8.9	-10.7	1.79	-9.3	0.39	-8.9	0.03	-8.9	0.03
28	13.4	13.0	0.42	13.8	0.38	13.5	0.14	13.4	0.03
38	32.1	33.1	0.96	32.3	0.16	32.0	0.08	32.1	0.03
48	48.4	49.5	1.14	48.1	0.26	48.5	0.10	48.4	0.04
58	63.3	62.4	0.88	63.4	0.12	63.3	0.00	63.3	0.01
Sum			6.32	1.45		0.38		0.15	

Table C-1: Errors Seen by Polynomial Order when Curve Fitting Longitudinal Stiffness.

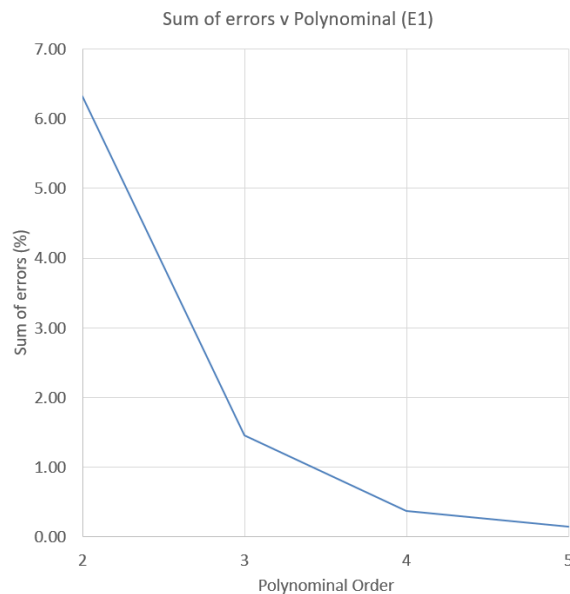


Figure C-1: The Sum of the Errors by Polynomial Order when Curve Fitting Longitudinal Stiffness.

[90]8 mode 1 difference (%)									
E2 (Gpa)	FE model	2nd order	Delta	3rd order	Delta	4th order	Delta	5th order	Delta
8	-44.8	-43.8	0.97	-44.7	0.12	-44.8	0.00	-44.8	0.00
18	-17.3	-18.9	1.57	-17.7	0.37	-17.3	0.02	-17.3	0.01
28	3.2	2.8	0.42	3.5	0.26	3.2	0.03	3.2	0.00
38	20.2	21.1	0.91	20.4	0.23	20.2	0.01	20.2	0.02
48	35.3	36.1	0.83	34.9	0.36	35.3	0.01	35.3	0.03
58	48.6	47.8	0.76	48.7	0.10	48.6	0.02	48.6	0.02
Sum			5.47		1.44		0.09		0.07

Table C-2: Errors Seen by Polynomial Order when Curve Fitting Lateral Stiffness.

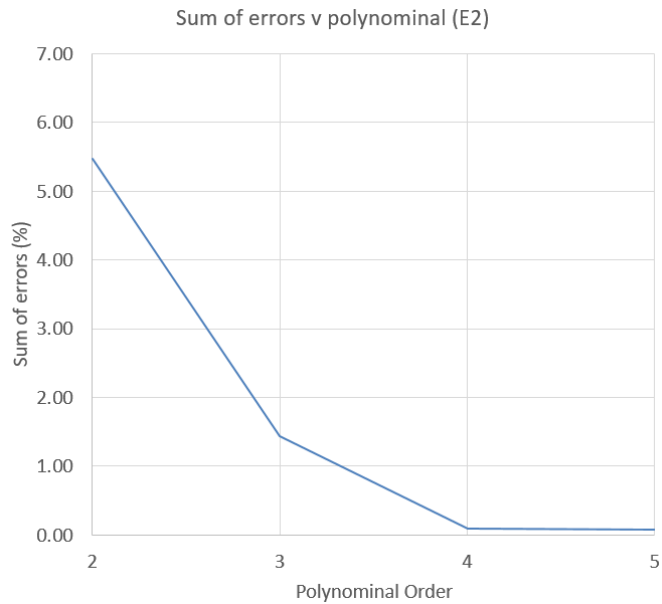


Figure C-2: The Sum of the Errors by Polynomial Order when Curve Fitting Lateral Stiffness.

As seen in Figure C-1 and Figure C-2 the errors have are converging by the 5th order polynomial, hence the decision to use this order or polynomial during the correlation exercise.

D Rationale for the Tolerance on the Constraints

During the optimisation of the board, a tolerance of $\pm 1\text{mm}$ was placed on the displacement constraints. A study was undertaken to see how significant this tolerance was on the design of the board. The same optimisation was therefore carried out with tolerances set at $\pm 2.0\text{mm}$, $\pm 1.5\text{mm}$, $\pm 1.0\text{mm}$ and $\pm 0.5\text{mm}$. This study resulted in the objectives and the constraint violations seen in Figure D-1.

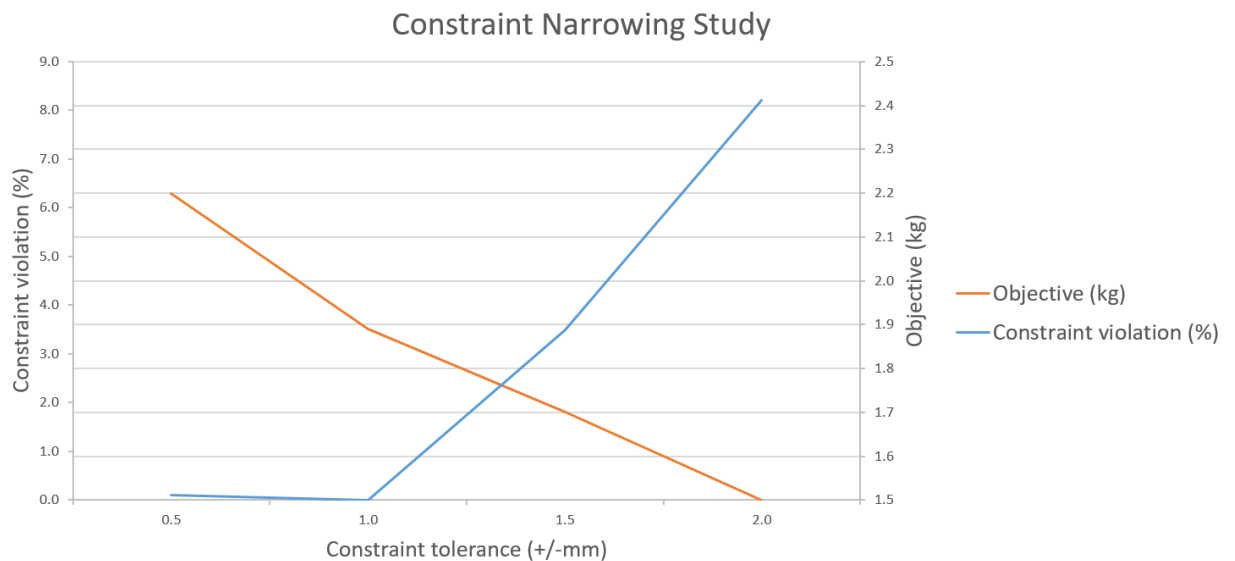


Figure D-1: The Objective and the Constraint Violation During the Constraint Narrowing Study.

The following observations could be made;

1. If the tolerance was reduced below $\pm 1\text{mm}$, the violation was not met.
2. If the tolerance was increased above $\pm 1\text{mm}$, then a lighter board resulted, although interestingly this violated the constraint.
3. A limit of 99 cycles was set for all the optimisation runs. When the constraint tolerance was set to $\pm 1.5\text{mm}$ the optimisation run did not converge and the maximum number of cycles was reached. When the constraint tolerance was set to $\pm 2\text{mm}$ Design Studio converged at 43 cycles despite the constraint still being violated.

The suggested lay-ups were then compared to see if this had a significant impact on the design of the board. These can be seen in Figure D-2 to Figure D-5. Whilst the thicknesses of the layers change slightly, the position of the suggested plys remains largely constant. Once these designs are interpreted into a manufacturable solution the resulting boards would have almost identical ply-books. As such it was established that the tolerance on the constraints did not significantly influence the design and a constraint tolerance of $\pm 1\text{mm}$ was deemed acceptable as this resulted in the lightest board.

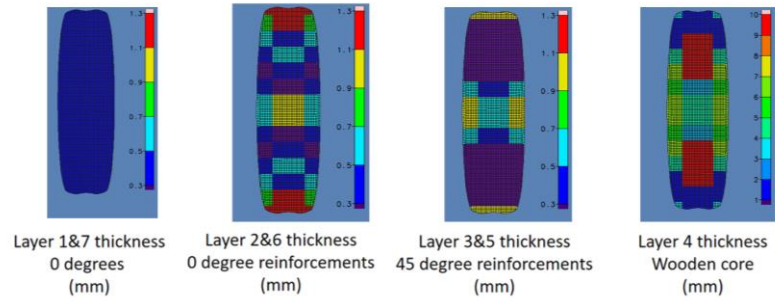


Figure D-2: Suggested Lay-up with a Tolerance of ± 0.5 mm on the Displacement Constraints.

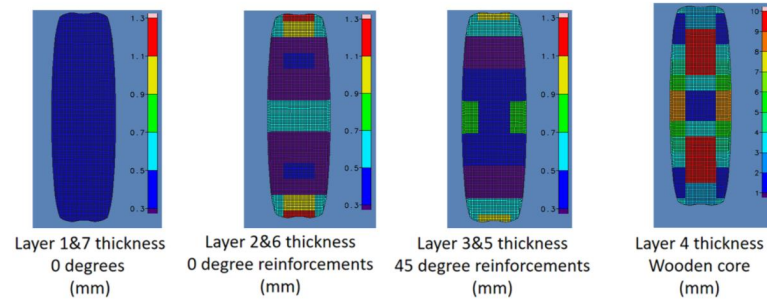


Figure D-3: Suggested Lay-up with a Tolerance of ± 1 mm on the Displacement Constraints.

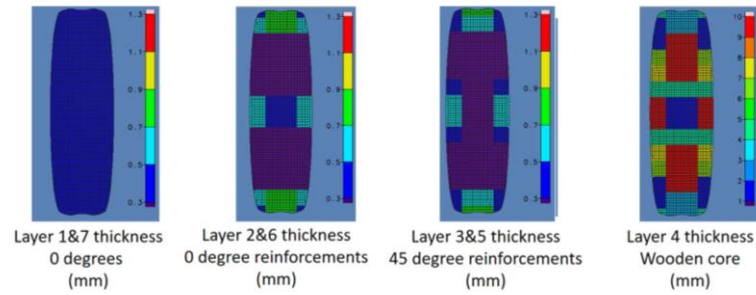


Figure D-4: Suggested Lay-up with a Tolerance of ± 1.5 mm on the Displacement Constraints.

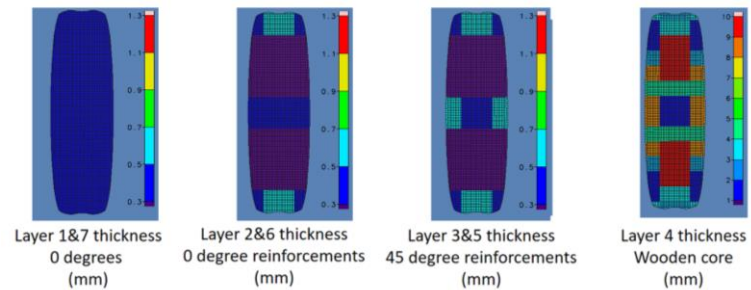


Figure D-5: Suggested Lay-up with a Tolerance of ± 2 mm on the Displacement Constraints.

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