TOPOLOGY OPTIMISATION OF 5000 LB OVER-CENTER BUCKLE

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A dissertation submitted in partial fulfillment of the requirements for the degree of

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DECLARATION

I declare that this is my own work and this thesis does not incorporate without acknowledgement any material previously submitted for a degree or diploma in any other university or institute of higher learning and to the best of my knowledge and belief it does not contain any material previously published or written by another person except where the acknowledgement is made in the text.

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The above candidate has carried out research for the Master thesis under my supervision. I also acknowledge the contributions made by Lecturer KH Janaka Mangala for the completion of the work.

Name of the supervisor: Senior Lecturer. R. K. P. S. Ranaweera

Signature of the supervisor: Date:

Abstract

Topology optimisation has for a considerable time been applied successfully in the automotive industry, but still has not commonly become a mainstream technology in the aerospace industry. The aircraft manufactures have already been achieving benefits with optimisation for some areas where as the bottom layer suppliers in the aerospace industry are still following conventional design techniques. Most of metal fittings which are widely used in the aerospace industry with safety nets and straps are identified as bulky and heavy as they are based on conventional designing techniques. 5000 lb over-center buckle (OCB) is one of the most frequently used tightening devices having the aforementioned characteristics.

The purpose of this study is to formulate a mechanism for a strength-based weight reduction on standard 5000 lb OCB which is used in the aerospace industry and consequently, to propose a light-weight design. First objective was to identify the relevant design considerations of existing 5000 lb OCB. Design specifications and standards related to 5000 lb OCB and 5000 lb safety strap were collected and reviewed for collecting necessary strength, functionality and other requirements of 5000 lb OCB. Second objective was to develop a finite element methodology for static structural analysis of 5000 lb OCB. 5000 lb OCB samples were carefully examined to identify the functionality and other necessary requirements of the OCB. OCBs were then subjected to a detail measurement check and the dimensions were used to build a computer aided design (CAD) model for the study. Engineering drawings were also created from the model for future reference. Then OCB samples with polyester webbing parts were subjected to various kind of strength tests using tensile testing machine. Purposes of these tests were to identify the failure loads and failure modes of the OCB itself and the OCB with safety strap in the operational conditions. These experimental results showed that the 5000 lb OCB used in the aerospace industry is an over-design. Last objective was to optimise the 5000 lb OCB using an effective optimisation scheme. Having reviewed on optimisation procedures and current trends in the aerospace industry, Altair HyperMesh software was selected as the numerical simulation tool to setup the finite element model and 'Topology Optimisation' was selected as optimisation method for the study. The finite element model was validated using simulation results and experimental results and the validated methodology was used to setup optimisation problem with aim of reducing weight. In formulating the topology optimisation problem, the minimum averaged compliance of the buckle was taken as the objective, and element density was used as the design variable.

Topology optimisation results were analysed and the elements in the critical regions were derived as geometries to compare those with original OCB model. Considering other functionality requirements with the topology optimisation results, a light-weight design was proposed with step-by-step modifications. Subsequently, FE simulations were repeated for the proposed light-weight design. Comparing the results of the light-weight design with the original model results, the proposed light-weight design can be noted as a better alternative. Nearly 7% (41g) weight reduction could be achieved for 5000 lb OCB using the proposed optimisation procedure.

Keywords:

Finite Element Analysis, Topology optimisation, 5000 lb over-center buckle, Weight reduction

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This thesis is dedicated to my loving parents, Mr. & Mrs. Batuwatta Gamage.

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List of Abbreviations

Abbreviation	Description
OCB	Over-center Buckle
CAD	Computer Aided Design
CAE	Computer Aided Engineering
FE	Finite Element
FEA	Finite Element Analysis
FEM	Finite Element Model
RL	Rated Load
SWL	Safe Working Load
IB	Inner Body
OB	Outer Body
SAE	Society of Automotive Engineers

1. CHAPTER 01: GENERAL INTRODUCTION

1.1. Over-center buckle (OCB) and Application

Over-Center Buckle (OCB) is a metal fitting which is mostly used with tie-down safety straps as a tightening and locking mechanism. The mechanism of the OCB enables the user to put additional tension on a tie-down safety strap through an effective over-center toggle action with minimum wear to the webbing. With an OCB, once the user has pulled as much webbing through the latch as can, the latch is toggled shut. This adds several inches more worth of tension depending on the size of the latch.

Typically, several designs are available for over-center buckle by means of load applied as well as size of the webbing is used with. Sizes are available for the rated load capacities of 300 lb, 1980 lb, 2500 lb, 3000 lb, 4400 lb, and 5000 lb and the webbing sizes used with of 1”, 1-3/4”, and 2” as required. [03]

Figure 1.1: Different types of over-center buckles

Source: [03]

Over-center buckle is made from hot rolled AISI 1044 carbon steel. Heat treatments depend on the Rating Load (RL) and the Safety Working Load (SWL) of the design. OCB is designed to reuse with maximum load of SWL without critical deformations. If the OCB is loaded to RL or above, then the OCB needs to be replaced as the deformations that occur are more significant. SWL can be expected in the elastic range and RL can be expected in the plastic range as shown in Figure 1.2. Surface finish comes with zinc plated, powder coated or chrome finish. Surface finish is to be selected according to usage requirements.

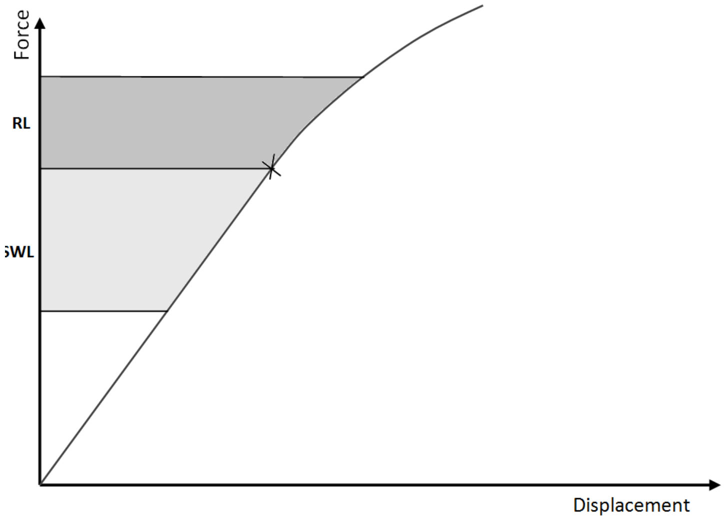


Figure 1.2: Rated Load (RL) and Safe Working Load (SWL) for OCB

OCBs are widely used with tie-down safety straps in different industries. Safety strap has two or more non-adjustable fittings at both ends which are used to fix the strap. OCB is located somewhere in the middle and used to adjust the length as required. Finally, extra tension could be given using the latching mechanism.

Figure 1.3: Tie-down straps with different OCBs (Original in color)
Source: [04]

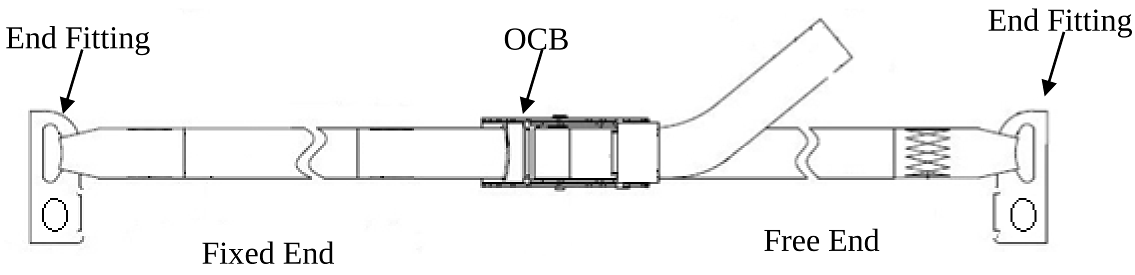


Figure 1.4: Tie-down safety strap with OCB

The webbing part is not a single part for the reason of length adjustment requirement for the safety strap. One webbing part is permanently stitched to the middle buckle which can be identified as fixed end. The other webbing part is kept with free hand hold with use of tightening mechanism of the buckle which can be identified as free end. The hand hold can be used to tighten the strap through the latch, the latch is toggled shut. More importantly, the webbing feeding method is the critical step to have required tightness in the operation. Incorrect webbing feeding will result in loose contact.

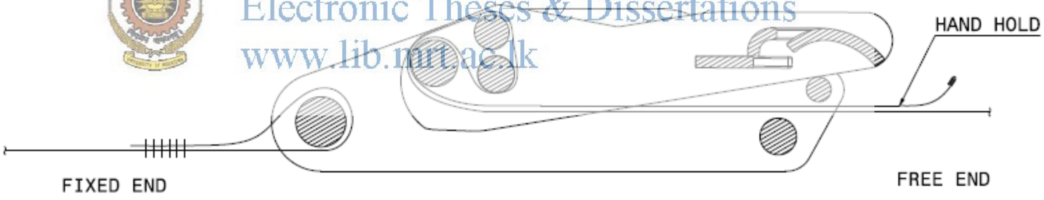


Figure 1.5: Webbing feeding method for OCB

Safety straps with OCBs are a great solution for quick tightening applications. OCB doesn't provide ratchet tight but significant tightness is provided with a much easier mechanism. Hence, tie-down safety straps can be seen in any industry to hold something with sufficient gripping force. Commonly, OCB straps can be seen in transportation sector. One or more tie-down straps are normally used to fix cargos, vehicles and various kinds of other goods in transportation field. However, the OCB design selected for this study is used in safety equipment in aerospace sector.

Figure 1.6: Applications of tie-down straps (Original in color)
Source: [05]

1.2. Selected 5000 lb OCB Details

The selected 5000 lb OCB is commonly used in aerospace sector and is manufactured using hot rolled sheet metal parts with thickness of 3 mm and hot rolled pins. The model can be mainly divided into two bodies as Inner Body (IB) and Outer Body (OB). The inner body consists of two sheet metal parts, two pins, a supportive sheet part, a spring and a plunger. The outer body also consists of two sheet metal parts, three major connecting pins and a hinge pin. Inner body as a single unit, can be rotated over the axis of the hinge pin. Plunger is used to lock and unlock the buckle and is spring loaded as shown in Figure 1.7.

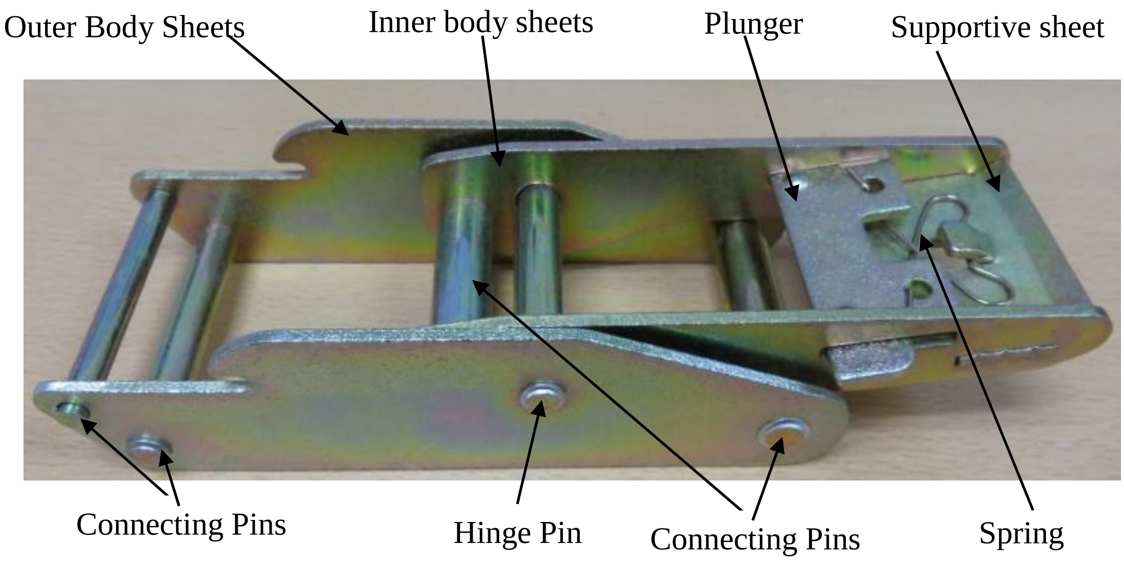


Figure 1.7: 5000 lb OCB – unlock mode

RL is identified as the ultimate load set for the OCB which had been introduced considering laboratory tensile test failure loads. If the OCB is loaded above RL, then the OCB must be replaced with new one. Rated load capacities are varied from 300 lb to 5000 lb according to the design and the size. The SWL is the maximum load expectation during the normal working conditions and is set normally as 2/3 to the RL which is 14.8 kN for the selected OCB. The selected OCB can be reused with SWL conditions for a minimum of two years period. [22]

Table 1.1: 5000 lb OCB requirements

Tensile Strength	SWL: 3333 lb (14.8 kN) RL: 5000 lb (22.5 kN)
Material	Carbon steel with heat treatments as required

Source: [22]

Some physical requirements can also be identified for the selected OCB design other than the structural requirements. The inner body sheet metal should have a minimum inner space to use relevant webbing straps, but to use it with the guidance of those sheet metal parts. The spring properties of the spring which support the plunger function and have a spring constant to easily operate the plunger with fingers. The supportive part should have a curved shape to support palm and unlock the OCB by pulling plunger back. All edges should be smooth edges.

2. CHAPTER 02: LITERATURE REVIEW

2.1. Optimisation and Optimisation Methods

The concept of optimisation is basic to what happens in daily lives, which is a desire to do better or be the best in one field or another. In engineering, the ultimate target now is to produce the best possible result with the available resources. The process of searching the best result is called "optimisation" and help from software tools are required to solve mathematical equations for achieving the desired results in a timely and economical fashion for designing new products in any field: aerospace, automotive, chemical, electrical, biomedical, agricultural, etc.

Design optimisation is a very general automated design analysis technique. In studying this technique, it is important to distinguish analysis and design. Analysis is the process of determining the response of the specified system to the certain combination of input parameters. For example, calculating stresses in the structure as a result of certain loads. Design on the other hand, means the process of defining a system. For example, designing a structure would mean selecting specific dimensions and location of the structural members that will allow the structure to withstand the specified load. Initial design can be subjected to the process of optimisation by means of analytical and numerical calculations till the optimum is achieved.

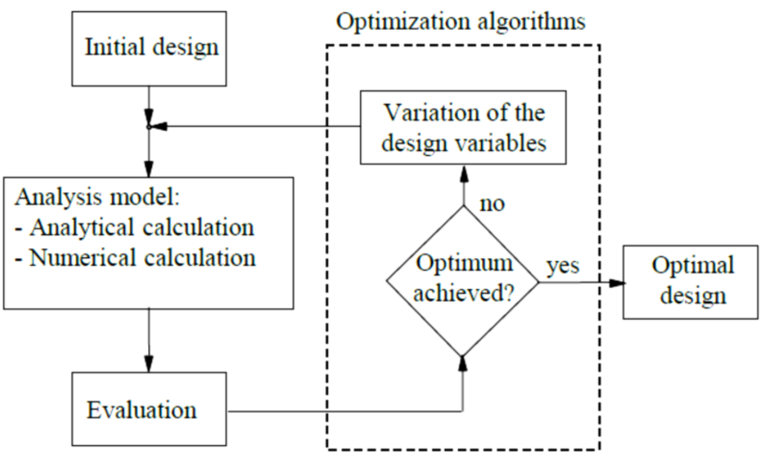


Figure 2.1: Design optimisation cycle
Source: [15]

Optimisation has become a necessity in the recent years to achieve an optimal design considering stress or strain, stiffness and weight etc. In earlier practices, dedicated codes are developed to achieve a specific optimised solution to a problem. For example, Bhat, Rao and Sankar (1982) used the method of feasibility directions to achieve optimum journal bearings for minimum unbalance response. [06]

Optimisation methods were wisely reviewed to find a suitable method for OCB optimisation. Structurally based methods were mainly considered as the study based only on structural analysis.

2.1.1. Optimisation by Evolution

Products are developed with the technological evolution and biological evolution. Most designs on the past have been optimised by an attempt to improve on an existing similar design. Survival of the resulting variations depends on the natural selection or user acceptance.

Often engineering optimisation is done by using a combination of judgment, experience, modeling, opinions of others, etc. Some engineers are very good at this. However, if there are many variables to be adjusted with several conflicting objectives and/or constraints, this type of experience-based optimisation can fall short of identifying the optimum design. [01]

2.1.2. Optimisation by Intuition

The art of the engineering is the ability to make good decisions, without being able to provide a justification. Intuition knows what to do, without knowing why one does it. The gift of intuition seems to be closely related to the unconscious mind. The history of technology is full of examples of engineers who made use of intuition to make major advances. Although the knowledge and tools available today are so much powerful, ultimately intuition continues to play an important role in technological development. [02]

2.1.3. Optimisation by Trial and Error Modeling

This refers to the usual situation in modern engineering design, where it is recognized that the first feasible design is not always the best. Therefore, the design model is exercised for a few iterations, in the hope of finding an improved design. However, this mode of operation is not true optimisation. Some refer to satisfying, as opposed to optimizing, to mean a technically acceptable job done rapidly and economically. Such a design should not be called an optimal design. [08] [09]

2.1.4. Optimisation by Numerical Algorithms

This is the area of current active development in which mathematically based strategies are used to search for the optimal. Calculations can be done either manually or using computers. Computer software is widely used for the complex calculations where accuracy is critical.

The purpose of numerical optimisation is rationally searching among alternative designs for the best design to meet the specified needs. The alternative designs of the same system differ from each other because some parameters of the system are not the same. The parameters that could be changed in the system while searching for the best design are called design variables. Design process can be defined as the process of finding the minimum or maximum of some characteristic, which may be called the objective function. For the design to be acceptable it must also satisfy certain requirements. These requirements are called design constraints. Optimisation automatically changes the design variables to find the minimum or maximum of the objective function, while satisfying all the required design constraints.

In order to employ this type of optimisation, several qualifications must be met. First, a quantitative model must be available to compute the responses of interest. Sometimes obtaining such quantitative models is not easy. Obtaining a valid, accurate model of the design problem is the most important step in optimisation. It is not uncommon for 90% of the effort in optimizing a design to be spent on developing and validating the quantitative model. Once a good model is obtained, optimisation results can often be realized quickly. [15] [20] [21]

Three main methods are possible.

- Size and Shape Optimisation
- Topography Optimisation
- Topology Optimisation

Size and Shape Optimisation

Shape optimisation means optimizing structural shapes by adjusting the surface shape of a 2D or 3D solid to minimize volume while satisfying stress and/or displacement constraints. Size optimisation consists of modifying size-related properties of structural elements such as shell thickness, beam cross-sectional properties, spring stiffness and mass to solve the optimisation problem. [15] [20]

Topography Optimisation

Topography optimisation is an advanced form of shape optimisation in which a design region for a given part is defined and a pattern of shape variable-based reinforcements within that region is generated using OptiStruct. The approach in topography optimisation is similar to the approach used in topology optimisation, except that shape variables are used rather than density variables. The design region is subdivided into a large number of separate variables whose influence on the structure is calculated and optimised over a series of iterations. The large number of shape variables allows the user to create any reinforcement pattern within the design domain instead of being restricted to a few. [15] [20]

Topology Optimisation

Topology is the mathematical study of shapes and topological spaces. Optimisation mathematically relates with the properties of space that are preserved under continuous deformations including stretching and bending, but not tearing or gluing. This includes such properties as connectedness, continuity and boundary.

Freeform optimisation (or topology optimisation) is a mathematical approach used in finite element analysis to determine the optimum material layout for a given design space which takes into any number of design constraints. By defining a design space it has to work in and applying boundary conditions such as predefined loads and fixture positions, topology optimisation can suggest the ideal layout of material to meet defined performance targets.

It is important to note that design proposals from a topology optimisation study will present a+n optimal layout of material distribution which may be at odds with the manufacturing processes employed. As such there is still a need for the engineer to interpret the output from the study into a final manufacture model that can be tested further. [15] [20] [26]

The Structural optimisation problem can be posed as;

Minimize or Maximize

$$F = F\ (x_1, x_2, x_3 \ldots\ldots x_n)$$

Subject to

$$C_1 = C_1\ (x_1, x_2, x_3 \ldots\ldots x_n) = 0$$
$$C_2 = C_2\ (x_1, x_2, x_3 \ldots\ldots x_n) = 0$$
$$.$$
$$.$$
$$.$$
$$C_n = C_n\ (x_1, x_2, x_3 \ldots\ldots x_n) = 0$$

and

$$\emptyset_1 = \emptyset_1\ (x_1, x_2, x_3 \ldots\ldots x_n) \geq 0 \quad (2)$$
$$.$$
$$.$$
$$\emptyset_n = \emptyset_n\ (x_1, x_2, x_3 \ldots\ldots x_n) \geq 0$$

Where: $x_1, x_2, x_3 \ldots\ldots x_n$ are the design variables, $C_1, C_2, C_3 \ldots\ldots C_n$ are equality constraints and $\emptyset_1, \emptyset_2, \emptyset_3 \ldots. \emptyset_n$ are the inequality constraints. The nature of the mathematical programming problem depends on the functional form of F, C and Ø. If these are linear functions of design variables, then the mathematical programming problem is treated as linear programming problem. On the other hand if any one of them is a nonlinear function of the design variable, then it is classified as nonlinear programming problem.

Typical objective functions could be related to cost, weight, reliability and so on. Essentially objective function is subjected to certain constrains arising from physical situations and limitations or from compatibility conditions on the individual variables.

Several benefits can be achieved through better optimisation.

- Reduce Development Time
- Create Efficient Designs
- Deliver Robust Designs
- Increase Communication

Structural optimisation is optimisation of a structure's shape, size, topology, topometry or topography to satisfy operating limits imposed on the response of the structure, and by further limits on the values that the structural parameters can assume. Structural optimisation methods apply optimisation algorithms to solve structural problems by means of finite element analysis. However, several numbers of equations will be set and totally large number of equations will have to be solved. Suitable computer software have to be selected and hence, in depth research was conducted on current trends in aerospace industry on optimisation. [15] [20]

2.2. Optimisation in Aerospace Industry

2.2.1. Topology Optimisation in A380 Airbus

Aggressive weight targets and shortened development time-scales in the civil aircraft industry naturally calls for an integration of advanced computer aided optimisation methods into the overall component design process. Airbus has in a number of recent studies used Altair's topology tool in an attempt to achieve lighter and more efficient component designs. Considered components include wing leading edge ribs, main wing box ribs, different types of wing trailing edge brackets as well as fuselage doorstops and fuselage door intercostal. The designs for most of these components are to some extent driven by buckling requirements but also by for example stress and stiffness requirements. Through collaborative partnerships with Altair Hypermesh, the software was developed to produce an innovative rib design, which resulted in an optimised weight saving over 500kg per aircraft. [12] [19]

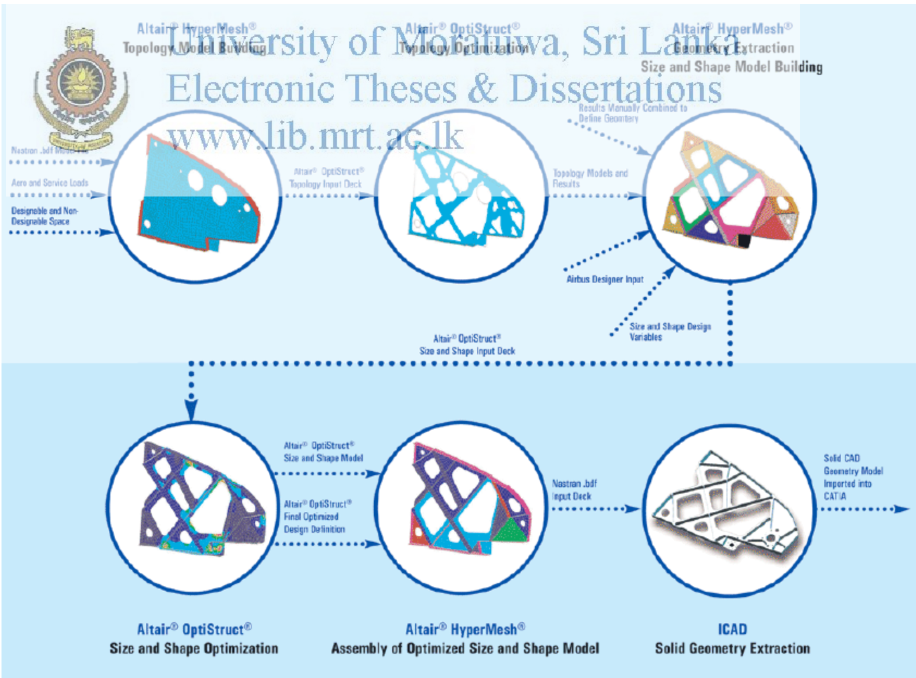


Figure 2.2: Topology optimisation on A380 parts (Original in color)
Source: [19]

Airbus as one of main biggest aircraft manufacture in the world, has selected Altair as a product development partner for ongoing aircraft design programs [12]. This is also given confidence to select the Altair Hypermesh OptiStruct as software facility for the analysis.

2.2.2. Topology Optimisation in Airbus Helicopters

Using OptiStruct topology and shape optimisation tools, Eurocopter created an innovative new design of a door support arm for the Fairchild Dornier 728 aircraft. Altair OptiStruct topology optimisation technology was deployed within Eurocopter's design process. This technology allowed Eurocopter to develop engineered design concepts taking into account performance and product objectives without having to develop, evaluate and iterate on multiple CAD design proposals. The initial door hinge design, provided by OptiStruct, maximized the stiffness for three load cases: door blocking, emergency opening and damper hit. In addition, draw direction constraints were included as part of the optimisation, yielding a design tailored to the specific method of manufacture. Secondary analyses further reduced the part mass by optimizing shapes and sizes of ribs for all load cases and a maximum allowable stress level. The company achieved a weight reduction of approximately 20 percent, using structural optimisation techniques as an integral part of the design process. [27]

2.2.3. Topology Optimisation in Honeywell Turbine Engines

Honeywell International, Inc. had been used Altair topology tool to gain considerable weight reduction in turbine engine designs. Structural strength based optimisation has been used to 10-30% weight reduction from several parts of the gas turbine engines. [28]

2.3. Topology Optimisation for OCB Optimisation

Literature review was carried out to find a suitable optimisation method for OCB optimisation process. After analysing the available methods, "Optimisation by Numerical Algorithms" was identified as the most actively and effectively used method at the moment. Investigation was continued to search on computer software (FEA) that can effectively perform the optimisation work. (refer to **Appendix A**).

Considering the design requirements, strength requirements, available FEA packages for optimisation process and the trends of the optimisation in aerospace industry, it was concluded that the Topology optimisation using Altair Hypermesh OptiStruct is the most suitable software tool for the 5000 lb OCB optimisation process. The important fact for the decision was that the aircraft manufacturers have already started the optimisation process with the Altair Hypermesh OptiStruct and the excellent results have been achieved.

OptiStruct has been developed in 2003 to perform linear structural optimisation and successfully applied for topology, topography, gauge and shape optimisations of automotive and airframe structures, e.g., Schuhmacher (2006), Taylor et. al., (2006) discussed the weight optimisation achieved in aircraft structures. With additional advances in mesh morphing techniques in HyperMesh, it has become somewhat easier in shape optimisation. [20]

3. CHAPTER 03: METHODOLOGY

The aim of this study is to formulate a mechanism for weight reduction of 5000 lb Over-Center Buckle (OCB) which is used in Aerospace safety equipment. Several objectives were set and well-structured methodology was followed to achieve a light weight design for the selected 5000 lb OCB. First objective of the project was to identify the relevant design specifications of existing 5000 lb OCB. Standards published by SAE Aerospace for safety straps and buckles were collected. In the meantime, a literature survey was carried out to select a suitable optimisation process for the OCB. Available optimisation methods and technologies were reviewed and thoroughly analysed to find a suitable optimisation procedure for OCB optimisation considering its load and strength characteristics. It was identified that the optimisation techniques considering numerical algorithms is the most actively used method to have weight reduction in various kind of parts especially in Automotive, Aerospace and commercial product industries. In addition, detail review was conducted regarding current trends in optimisation in aerospace industry to select specific numerical algorithm with a suitable computer software for the study. Optimisation processes are already being applied by the aircraft manufactures with combination of Altair Cooperation Inc. Size and shape, topography and topology optimisation procedures are commonly used in the field to achieve lighter designs in various structures and engines. Considering the load conditions and the required outcome, topology optimisation comes with Optistruct tool in Altair Hypermesh was selected for the study. Topology optimisation is concerned with material distribution and how the members within a structure are connected. It treats the "equivalent density" of each element as a design variable. This method will narrow the field of solutions to a set of feasible solutions considering manufacturing constraints.

Second objective was to develop a finite element methodology for the structural analysis of 5000 lb OCB. It was difficult to find some details related to the OCB and to prove those. Hence, experimental tests were required to conduct on OCBs to obtain characteristic load and deformation parameters of the OCB. All the experimental tests were conducted in accordance with SAE AS5385 Revision C which is the standard for

"Design Criteria and Testing methods for Cargo Restraint Straps". Few samples of 5000 lb OCBs were obtained and were carefully observed to understand the functionality of the OCB with its mechanisms and with the safety straps. Webbing feeding method, handhold requirements and lock / unlock mechanism were identified as key considerations of the 5000 lb OCB. Three samples were then measured and one was dissembled to check dimensions and features of each part. A CAD was generated using the measurements and 2D drawings (refer to **Appendix B**) were generated as reference for the design used for the study. Strength related tests were conducted using Testometric tensile test machine. As the first step, the measured OCB samples were loaded to the Safety Working Load (SWL) ± 0.1% and kept for 03 seconds and unloaded. The tested OCBs were again subjected to a measurement check. The difference in critical measurements before and after the test were observed and noted as deformations. Next set of tests were conducted to check the failure mode of the webbing and OCB interface which is similar to the operational conditions of safety straps. Three samples were prepared with polyester webbing parts keeping free hand hold at the free end and having permanent fix at fixed end. Samples were setup and loaded till failure occurred. All samples were failed from the webbing at free end and buckles were permanently deformed. As still buckle did not fail, next step was to check the buckle failure load. Doubled webbing layers were used for both ends and tests were repeated as the same way. Buckle failure occurred this time at the fix end pin and the sheet metal interface. However, these failure loads were showed that the OCB has far better strength compared to the rated load 5000 lb. Accordingly, experimental rests were proved that the 5000 lb OCB is an over designed metal part.

The CAD model of the OCB was simplified removing some cosmetic shapes, such as fillets, chamfers and tiny shapes, merging permanently fixed parts, such as outer body sheet metals and connecting pins, and removing some parts such as fixed end webbing part, washers to use fixing pins to sheet metal parts. The CAD geometry was simplified to three parts instead of more than 20 parts in the original design. These steps are mainly to reduce the complexity of the analysis and hence to avoid having inaccuracy simulation results. The simplified model in CAD software was then saved as *.STP format and was imported to Hypermesh software as geometries. OptiStruct solver was selected in the HyperMesh. Basically, two materials were considered for the analysis. The first material collector was created using AISI 1044 hot rolled material properties and is for all the buckle parts. (refer to **Appendix B**). The other material collector was created for the webbing part using dyneema fibre properties. Three component collectors were then created for three separate parts of the buckle (Inner Body, Outer Body and Supportive Part). A separate component collector was also created for the webbing part.

Geometries were converted to mappable geometries doing internal edits in HyperMesh software. The aim was to have brick elements in meshed model of the OCB. Brick elements such as Pentahedral (an element having 6 or 15 vertices, 9 edges, bounded by 2 triangular and 3 quadrilateral faces) and Hexahedral (an element having 8 or 20 vertices, 12 edges, bounded by 6 quadrilateral faces) elements would minimise the element distortion. The three metal parts of the OCB and the webbing part were separately meshed to the relevant component collectors as required. A mesh element check was conducted to check quality of the meshed model and compared those values with the acceptable range published for OptiStruct solver. After several attempts, all relevant values were achieved in the acceptable range. The next step was to create contact surfaces between the parts where it really necessary. Slide and Freeze contact surfaces used as required. As the last step of the FEM preparation, boundary conditions were setup. Fixed support boundary condition was given to the pin where the fixed end webbing goes through. Load was applied to the webbing part to the outward direction from the plunger side. Point loads were given to keep uniform load over the cross section of the webbing part.

The FE model was then required to validate before using it for optimisation. Three steps were followed. Firstly, the deformation results in FE model at SWL were compared with the experimental results. Secondly, the deformation shapes and values of critically loaded pins were compared with the experimental results. Thirdly, von-mises stress of FEA for the load which the buckle failure occurred were compared with the buckle failure mode. All these steps were proved that the FEM is acceptable to use it for strength analysis and the buckle would more closely give the realistic results. Hence, the model was continued to develop an optimisation problem with topology optimisation.

The model was further processed following necessary steps for topology optimisation. The aim was to reduce the total weight of the OCB without reducing the strength of the buckle. Responses which are the measurements of system performance, were created to define the constraint and the objective function. As volume is proportional to the weight of a part, Volfrac response was created and constraint was defined to limit the volume reduction to a maximum of 30%. Compliance response was created to define objective function which is the function whose least value is searched by the solver during optimisation. The objective function was set to minimize the compliance of the OCB model. The prepared model was then solved using the topology optimisation scheme. Analysis was completed after running the process with several iterations and the message stating that the optimisation had converged with satisfaction of all the constraints applied. Results were analysed using HyperView software and the last iteration was considered for the results. A colour variation of the elements could be viewed according to the element densities of each mesh element. The solver has calculated an equivalent density for each element, where 1 is equivalent to 100% material, while 0 is equivalent to no material in the element. Considering the objectives and responses had been setup, the elements which are below 0.1 element density, were neglected. As a result, the elements which are above the 0.1 element density, were derived as a geometry. The derived geometry was then opened with the original OCB model in the same window to visually identify the areas that can be neglected. Analysing the both geometries carefully and identifying other physical requirements, a light weight design was proposed with step by step modifications for the 5000 lb OCB. A new FEM was again setup for the proposed OCB design and subjected to tensile strength analysis for SWL. The von-mises stress results received from the new FEM were analysed and compared with the von-mises stress results of the original design which had been kept as reference in the study. Comparing results in both cases, it was identified that first proposed design gave closer results to the original model results and hence, the optimised model was considered as feasible design. Nearly 7% weight reduction could be achieved from the proposed design using proposed optimisation method.

4. CHAPTER 04: EXPERIMENTS AND RESULTS

Various kind of experiments was conducted to obtain characteristic load and deformation parameters of the 5000 lb OCB. As the first step of the experimental tests, a detail measurement check were conducted on OCB samples. Few OCBs were disassembled and pins, sheet metal parts and other parts were carefully observed to have details of each parts. The dimensions were used to build a 3D CAD model using Co-Create 17.0 software which is for FE analyses. Engineering 2D drawings were also created from the 3D model for the 5000 lb OCB assembly and each part using CATIA V5 software. The drawings were created as references of the 5000 lb OCB which is used for this study.

Then the 5000 lb OCB samples with polyester webbing were subjected to various tensile strength tests using a 50 kN Testometric tensile testing machine. The purpose of the experimental tests was mainly to check the load characteristics of the 5000 lb OCB. All the testing were conducted in accordance with SAE AS 5385 revision C standards which is the standard used to follow when qualifying safety straps and relevant metal parts to use those in aerospace industry. Three dimension checked samples from previous step were taken and test samples were prepared. Polyester webbing parts were used by stitching permanently a webbing part to the fixed end of the OCB and keeping the free hand hold at the free end. Two webbing bollards were used to setup the samples in the tensile test machine (refer Figure 4.1). Once the specimen is set up within the test machine bollards, the test machine data display is programmed according to the requirement of the test. The distance between the bollards of the all tests were kept between 550mm to 600mm range to make the test consistent. Tests are then initiated by moving the top bollard vertically upwards at the rate of 75mm/min until the required load reached or the sample failure occurred. The machine was programed to stop in the range of 14.8 ± 0.1 kN (3333Lb) for the deformation test samples and to continue till the sample failure occurs for sample failure tests. Either way, the bollard movement was stopped and the loads and failure modes (if applicable) were recorded.

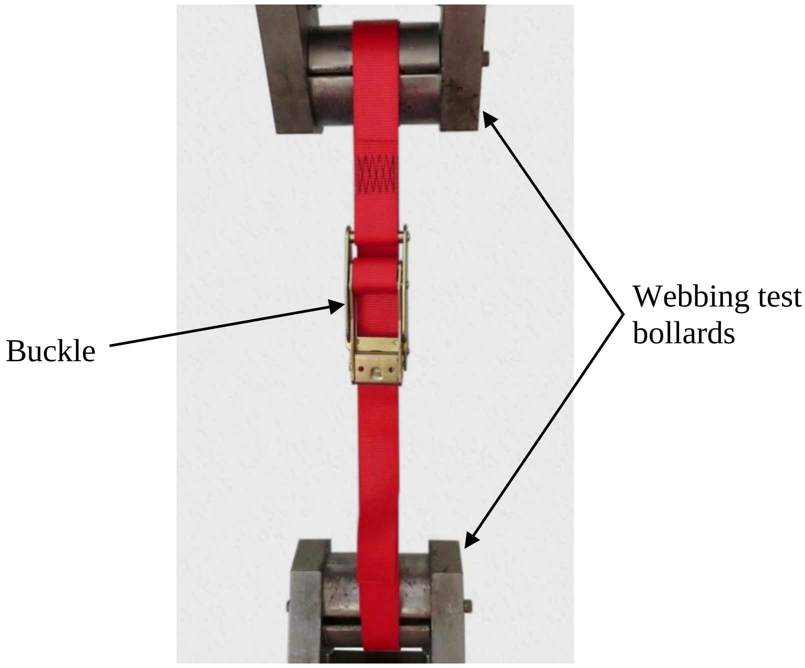


Figure 4.1: Tensile strength test setup – OCB (Original in color)

Mainly there kinds of tests were conducted to experimentally check the buckle design as mentioned before.

4.1. Deformation Check for SWL

The purpose of the deformation check was to examine whether the buckle parts are deformed in the plastic region for the maximum load of SWL (14.8 kN) or not. The dimension checked OCB sample with polyester webbing specimens, was loaded about SWL, kept for 3 seconds and then unloaded. The unloaded buckle was again subjected to a dimension check for checking critical dimensions of the OCB (refer to **Appendix C** for balloon marked dimensions). Dimensions before and after test were compared and were tabulated.

Table 4.1: Deformation results for SWL of original OCB

Sample	Balloon Ref. [Appendix C]	Drawing Dimension (mm)	Measured Dimension (mm)		Difference (mm)
			Before	After	
1	A	5 (Minimum)	5.18	5.12	-0.06
	B	Ø9 (Minimum)	9.15	9.14	-0.01
	C	Ø9.5 (Minimum)	9.52	9.51	-0.01
	D	5 (Minimum)	6.05	6.00	-0.05
	E	Ø12 (Minimum)	12.60	12.60	0.00
	F	Ø12.7 (Minimum)	12.80	12.81	0.01
	G	167	166.26	166.30	0.04
	H	42	42.50	42.48	-0.02
	H	45.5	45.72	45.76	0.04
	K	55	56.52	56.48	-0.04
2	A	5 (Minimum)	5.05	5.02	-0.03
	B	Ø9 (Minimum)	9.51	9.49	-0.02
	C	Ø9.5 (Minimum)	9.54	9.53	-0.01
	D	5 (Minimum)	6.12	6.03	-0.09
	E	Ø12 (Minimum)	12.62	12.59	-0.03
	F	Ø12.7 (Minimum)	12.82	12.80	-0.02
	G	167	166.54	166.59	0.05
	H	42	42.52	42.51	-0.01
	J	45.5	45.82	45.88	0.06
	K	55	56.38	56.40	0.02
3	A	5 (Minimum)	5.25	5.18	-0.07
	B	Ø9 (Minimum)	9.30	9.31	0.01
	C	Ø9 (Minimum)	9.45	9.38	-0.07
	D	5 (Minimum)	6.40	6.32	-0.08
	E	Ø12 (Minimum)	12.68	12.65	-0.03
	F	Ø12.7 (Minimum)	12.83	12.83	0
	G	167	166.82	166.88	0.06
	H	42	42.42	42.42	0
	J	45	45.80	45.82	0.02
	K	56	56.51	56.50	-0.01

Measured values could be considered as very small compared to the relevant dimensions. Hence, the differences in dimensions could be neglected. The test was proved that the OCB design can be repeatedly used for the SWL conditions without having plastic range deformations.

4.2. Tensile Strength Test for Webbing / Buckle Interface

Three samples were prepared with webbing parts stitched in the fixed end and with free hand holds at the plunger end (refer to the Figure 1.4 and 1.5). The purpose of this test was to check the OCB for tensile strength while keeping the conditions similar to the operational conditions. One layer of webbing has been used for each ends.

Table 4.2: Webbing / buckle interface tensile test experimental data

Specimen	Breaking strength		Failure Mode
1	30.41 kN	6836.4 lb	Webbing failed at free end (Figure 4.2)
2	28.97 kN	6512.7 lb	
3	29.42 kN	6613.9 lb	
Min.	**28.97 kN**	**6512.7** lb	

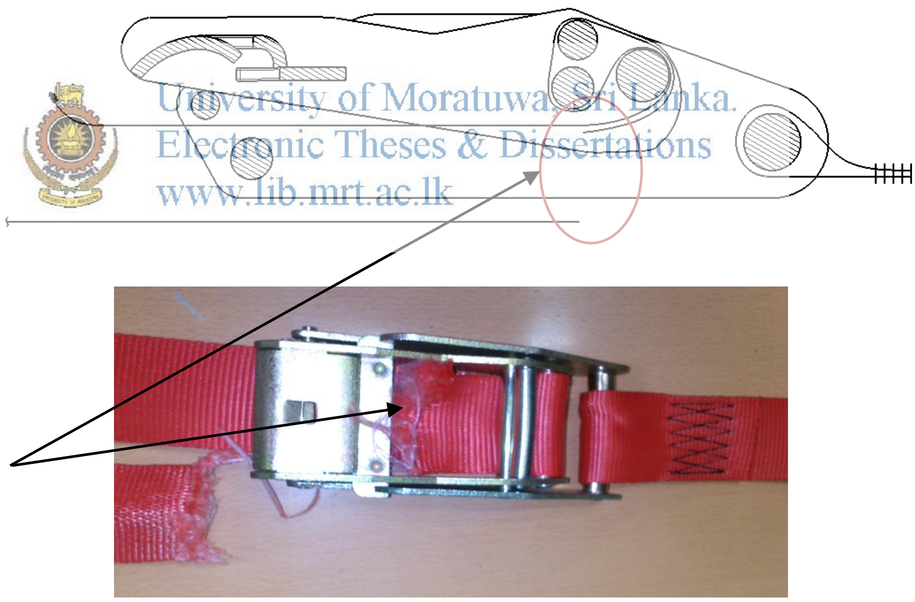


Figure 4.2: Webbing / buckle specimen failure mode (Original in color)

All the specimens failed from the free end webbing part closer to the three pins of the inner body where the friction is generated between webbing layers to hold the free end. The OCBs have been permanently deformed. The failure was occurred when the load is averagely reached to 130% of the rated load. Still the failure occurs from the webbing and the OCB did not fail. This experimentally proves that the buckle webbing interface fails from the webbing before the OCB failure occurs. Hence, tensile load failure tests were repeated using double layers of webbing to strength webbing parts and to check the buckle failure.

4.3. Tensile Test Results for Buckle Failure

The 5000 lb OCB was tensile tested using double layers of webbing and tests were continued till failure occurred.

Table 4.3: Buckle failure tensile test experimental data

Specimen	Breaking strength		Failure Mode/s
1	39.32 kN	8839.5 lb	Buckle failed (Figure 4.3)

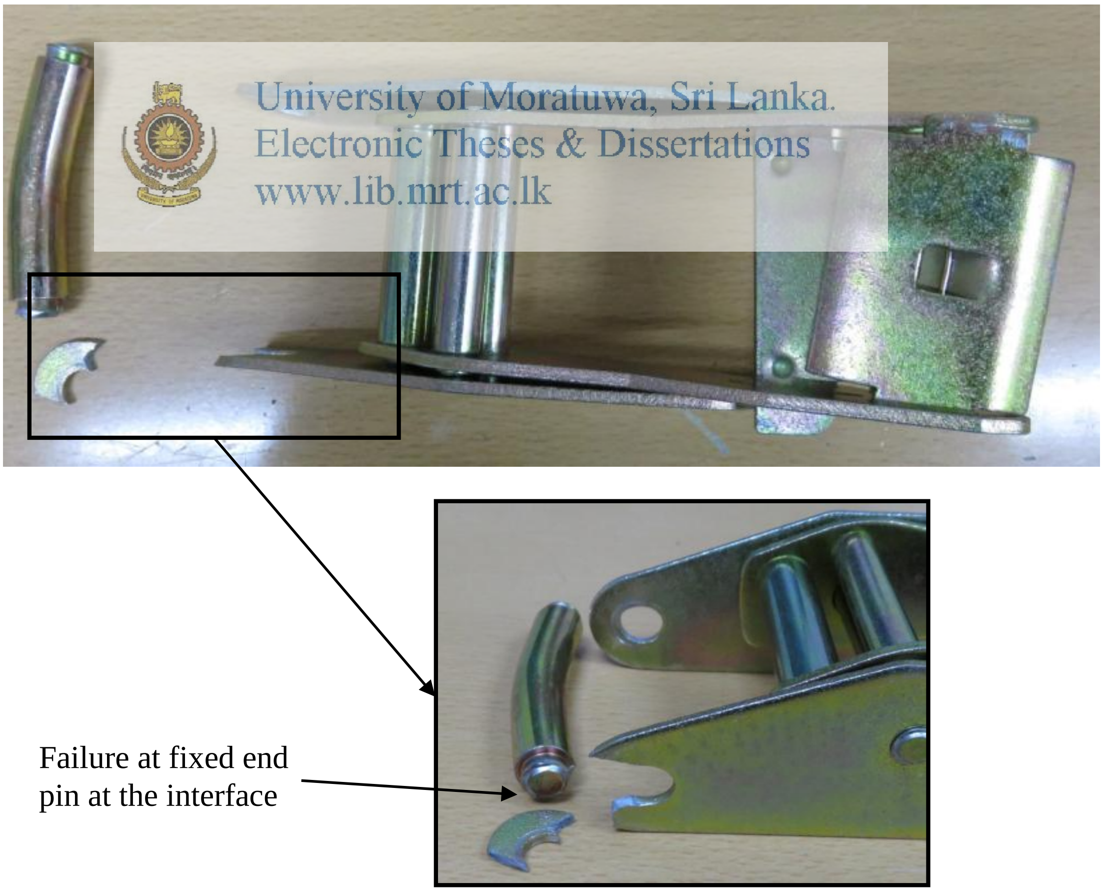


Figure 4.3: Failure mode – Test of buckle till failure

The samples were this time failed from the outer body sheet metal part from the fixed end pin interface as shown in the Figure 4.3. The buckle failure was occurred when the load is passed 175% compared to the rated load (22.25 kN). Force against Time graph generated from the test machine is attached in Appendix D.

Observing the results obtained from the tensile strength tests of the selected 5000 lb OCB, it was proved that the 5000 lb OCB is an over-designed model for the purpose it is used with. Though the RL (5000 lb) is the ultimate load of the OCB, the OCBs failure occurred when it passed the 8000 lb limit. Further, the 5000 lb OCB design used in aerospace industry was clearly identified as a design which can be optimised further having light weight design for it.

5. CHAPTER 05: FEM SETUP, ANALYSIS AND RESULTS

5.1. Finite Element Model (FEM) Setup

Functionality of the over-center buckle (OCB) was carefully observed before setting up a Finite Element Method (FEM). Looking to the operation of the buckle, the webbing fixed end is identified opposite to the plunger side. The webbing is rolled over the front most pin and stitched to make it fixed. From the plunger side, other webbing goes through pins as showed in Figure 1.5 and one end is kept as a free hand hold. The webbing feeding technique will support to keep the friction required between webbing layers to avoid slippage when the OCB is getting tensioned.

FEA is a numerical technique calculating approximate solutions of complicated equations. It does this by generating a mesh, grid of nodes, over the entire model which is based on the CAD model. The density of the grid varies throughout the model, having higher densities at fillets, corners, and high stress areas. The mesh contains the structural and material properties, which define how the structure reacts to the conditions implemented within the analysis. At each node an equilibrium equation is generated and the equations are solved to obtain the results in the analysis. Hence, quality mesh will reduce the number of nodes and the solving time accordingly. The 3D model of the over-centre buckle was simplified as required by removing non-critical parts, merging permanently fixed parts and removing cosmetic features of the OCB design. The number of parts of the OCB design was reduced to 03 (inner body, outer body and supportive part) instead of more than 20 parts of the original design. With the removal of fixed end webbing, the number of webbing parts also reduced to one. These steps were reduced the complexity of the mesh model with less number of nodes and less number of contacts. As a result, an accurate result could be achieved with less number of elements and nodes with lesser solving time.

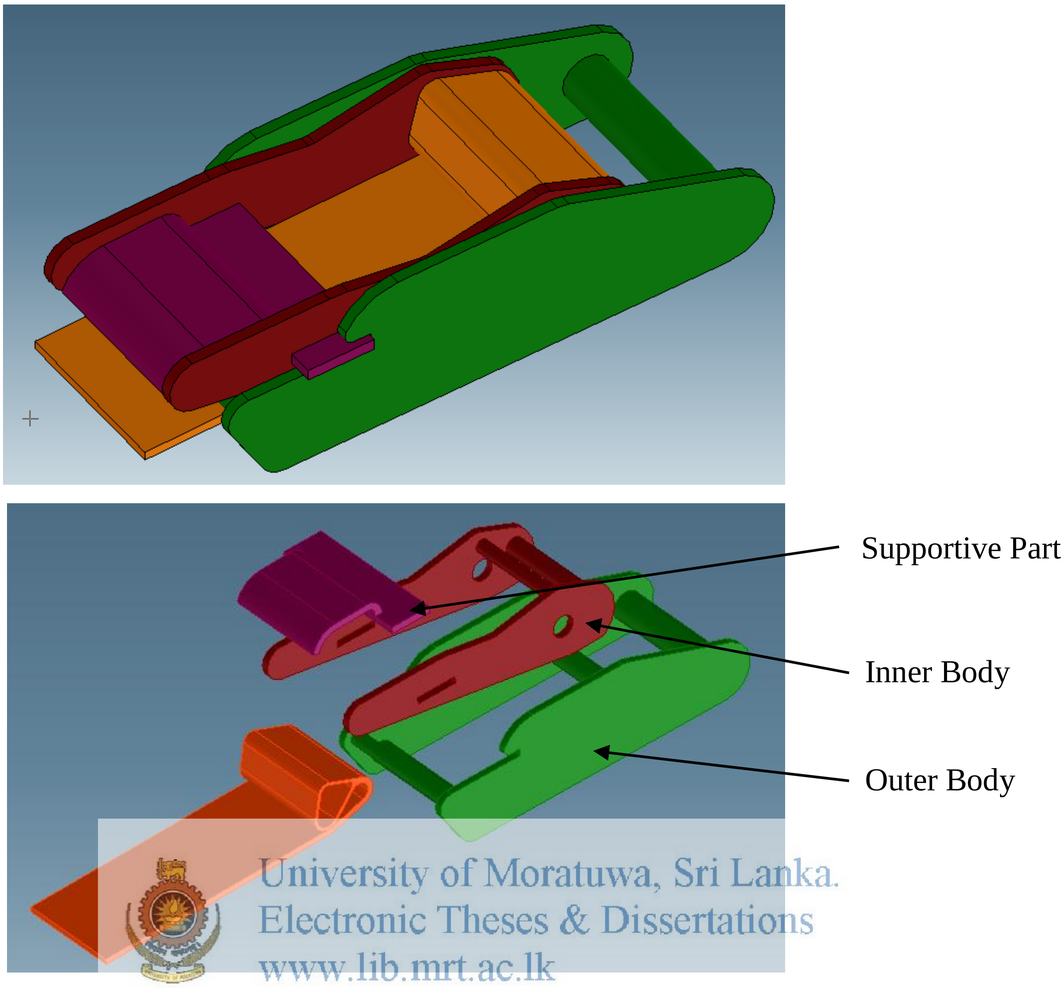


Figure 5.1: Simplified CAD geometry of OCB (Original in color)

The simplified CAD model then imported to the Altair HyperMesh as geometries. OptiStruct solver was selected as the solver in the HyperMesh software. Two material collectors were buckle parts and the webbing part. Dyneema material was considered for the FEA as it has lower elongation properties.

Table 5.1: Properties of the material used

Material Name	Density (kg/m^3)	Tensile Strength (MPa)	Yield Strength (MPa)	Modulus of Elasticity (GPa)	Shear Modulus (GPa)	Poisson's Ratio
Hot rolled AISI 1044	7870	620	415	200	80	0.29
Dyneema	975	3600	-	130	45	0.46

Source: [09] [10]

Property collectors were then created including the material collectors. STEEL SOLID property was used for the metal buckle parts and DYNEEMA property created for the webbing part. The most important step of a FE analysis is to have quality mesh elements as much as possible. “Solid Map” function was used to have brick element in the meshed model. However, as the OCB geometries are not simple mappable geometries, the geometries were required to be edited in the Hypermesh software internally. [13] [14] [15]

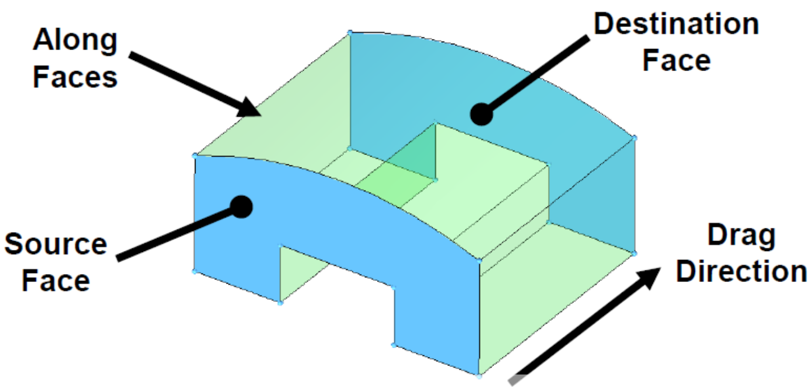


Figure 5.2: Mappable solid geometry sample

Source: [13]

‘Solid Edit’ function was used to trim and merge geometry parts as required to convert the complex buckle parts to mappable parts. No any complete separation occurs between edited parts as the solid parts are edited within the Hypermesh software. The elements in the common faces of two or more edited parts share same nodes and interfaces keeping the connection among the elements in the different parts. Geometries were then meshed keeping element sizes to have at least three layers of the direction of the sheet metal thickness to avoid problems in analysis convergence. As the sheet metal parts are 3mm in thickness, the element sizes are restricted to 1mm to have required minimum element layers in the sheet metal parts. After several attempts, better mesh could be achieved for the OCB.

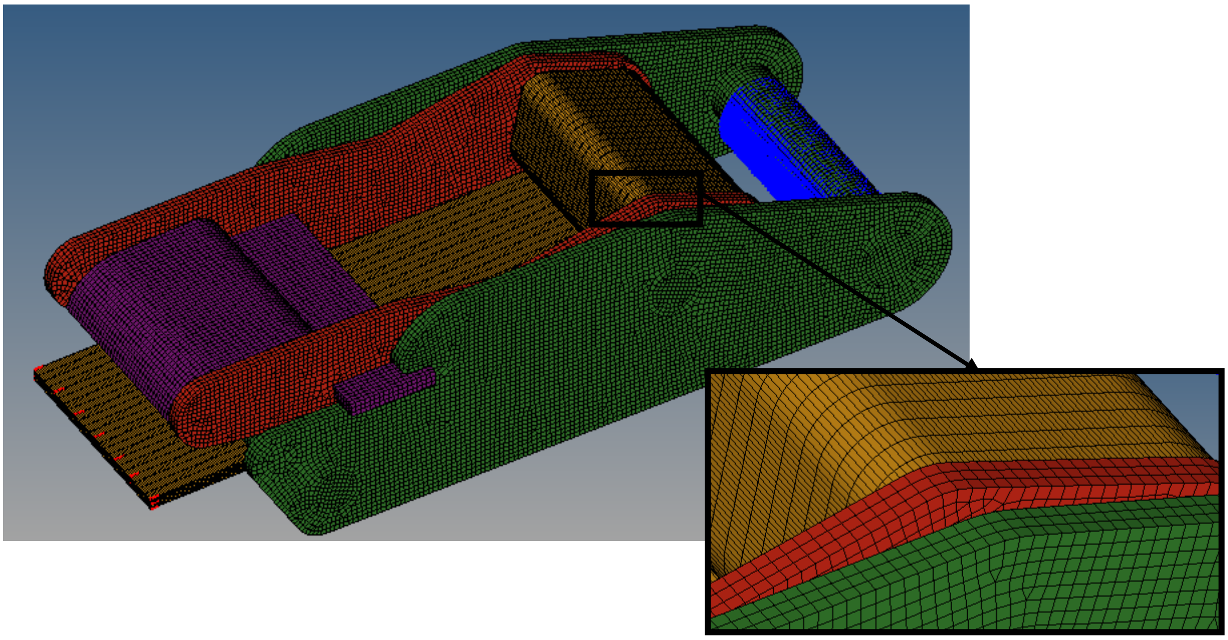


Figure 5.3: Meshed model – OCB (Original in color)

5.2. Mesh Quality of the Meshed Model

Mesh elements of the metal parts and the webbing part was a mix of Hexahedral and Pentahedral elements which are extracted from 2D quad and tria elements. This combination was given better mesh quality with complex shapes and boundaries. Hexahedral or more famously called as the brick element has 8 or 20 nodes (depending on the order of the element) and 6 quadrilateral faces and pentahedral has 6 or 15 nodes (depending on the order of the element), 2 triangular faces and 3 quadrilateral faces. Hexahedral elements give most accurate and fasted results and pentahedral elements resolve boundary layer efficiently. Also better numerical behavior can be achieved in problems and meshes comprised of hexahedrals and pentahedrals are easier to visualize than meshes comprised of pentahedrals. [17]

Geometry parts (inner body, outer body and webbing) were separately meshed into relevant component collectors and before moving to the next step, mesh quality of the model was checked in order to evaluate suitability of the mesh model for analysis. There were several attempts changing parameters to have a better mesh. Check Element function was used to get the minimum and maximum values of the model related to various kinds of checks as shown in Figure 5.6. The reason for the element quality check is complex but, is related to the fact that quality and the solution, by definition, is approximate. In the formulation of finite elements a local parametric coordinate system is assumed for each element type and how well the physical coordinate systems, both element and global, match the parametric dictates element quality.

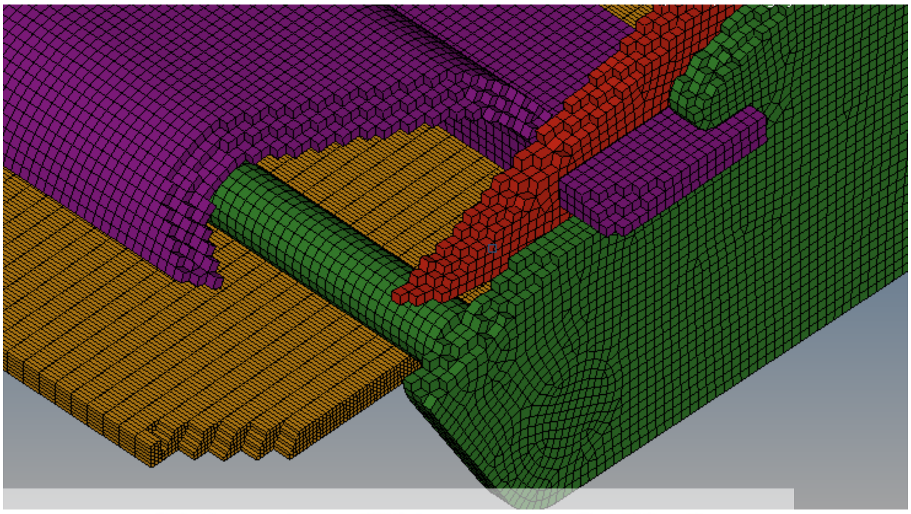


Figure 5.4: Hexa and Penta elements – meshed model (Original in color)

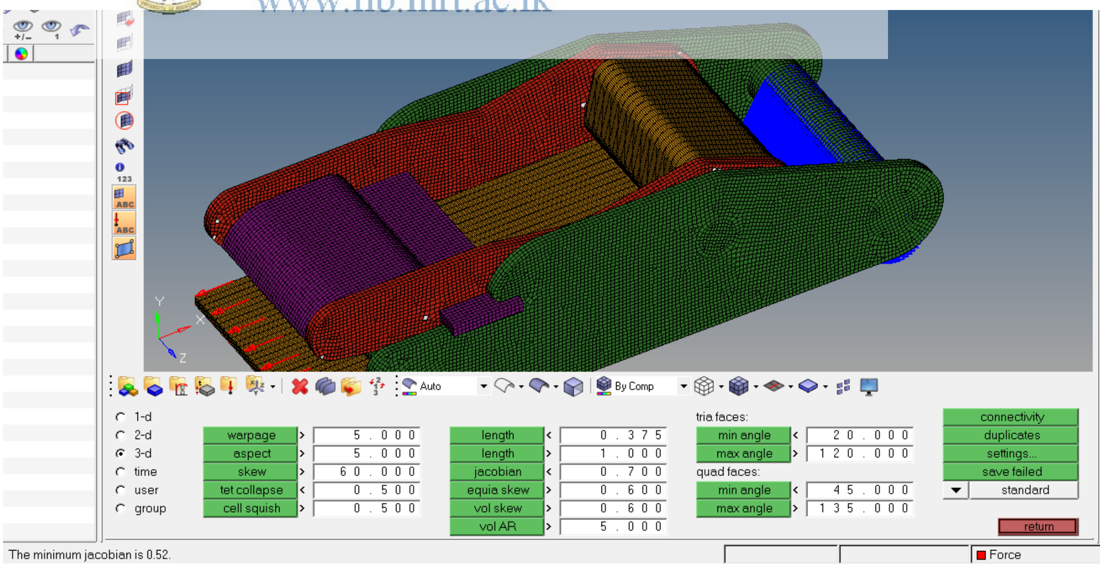


Figure 5.5: Mesh elements check (Original in color)

The values received from element quality check of the model were compared with the acceptable ranges and values for each check as shown in Table 5.2. The acceptable ranges and values are valid only for the Hypermesh OptiStruct meshing and has been published by Altair Engineering Inc. [13]

Table 5.2: Mesh quality check values

Check Type	Mesh Value	Ideal Value	Acceptable Range / Value
Warp angle	Max. 0.05°	0°	< 30°
Jacobian	Min. 0.52	1	> 0.5
Skew Angle (Tria face)	Min. 26.68° Max. 93.19°	-	20° < 0° < 120°
Skew Angle (Quad face)	Min. 48.41° Max. 134.42°	-	45° < 0° < 135°

Source: [13]

Warp angle defined as the angle between the normal to two planes formed by splitting the quad element along the diagonals. The maximum angle of the two possible angles is reported as the warp angle. The maximum wrap angle for the mesh model is 0.05° which is extremely closer to the ideal value as shown in the Table 5.2.

In simple terms, the jacobian is a scale factor arising because of the transformation of the coordinate system. Elements are transformed from the global coordinates to local coordinates which is defined at the centroid of every element, for faster analysis. The determinant of the jacobian relates the local stretching of the parametric space required to fit it onto global coordinate space. Hypermesh evaluates the determinant of the jacobian matrix at each of the element's integration points (also called Gauss points), and reports the ratio between the smallest and the largest. The minimum jacobian for the mesh model is 0.52° which in the acceptable range as shown in the Table 5.2.

Skew in trias is calculated by finding the minimum angle between the vector from each node to the opposing mid-side and the vector between the two adjacent mid-sides at each node of the element. Ninety degrees minus the minimum angle found is reported. Skew in quads is calculated by finding the minimum angle between two lines joining opposite mid-sides of the element. Ninety degrees minus the minimum angle found is reported. The skew check is performed in the same fashion on all faces of three-dimensional elements. All elements in the meshed model fall within the acceptable ranges. [13]

5.3. Contact Surfaces for the Meshed Model

5000 lb OCB is an assembly which includes several sheet metal parts and solid cylindrical pins. Some parts are permanently fixed. Some parts can slide over another part. As an example, complete Inner Body Sheets can slide around the Hinge Pin (refer to Figure 1.7). When the buckle is in operation, two parts of webbing is used as shown in Figure 1.5. All the parts of inner body and outer body and the webbing part in the FEM are required to have contact surfaces in the interfaces. As some of the parts had already been merged, number of contact surfaces required was lesser. Contact surfaces were created on the contacting interfaces as required and Contact Groups were then created to introduce the type of the contact. Correct contact type is needed to be applied to get results more close to the actual conditions.

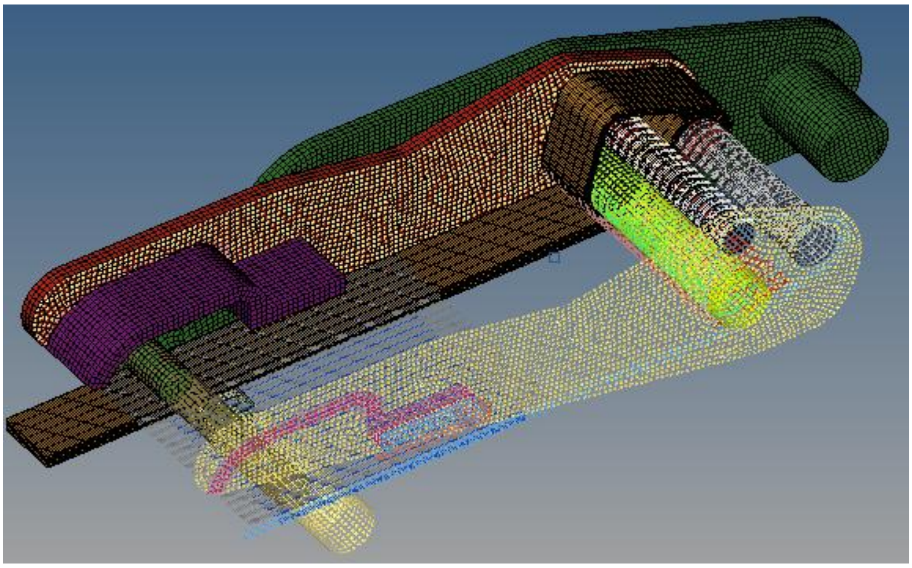


Figure 5.6: Contact surfaces on FEM (Original in color)

All the contact surfaces are kept as FREEZE except the contacts are listed in the Table 5.3. Freeze contacts are enforced zero displacements on the contact interface. As slide contacts are assumed to have small slide with respect to each other, the model is still can be analyzed in static linear range.

Table 5.3: Contact surfaces for sliding contacts

Interface	Type	Figure
Webbing / Inner body sheets	Slide / Open Gap	Figure 5.7
Webbing / Webbing	Slide / Open Gap	Figure 5.8
Webbing / Connecting Pins	Slide / Open Gap	Figure 5.9

Figure 5.7: Contact surfaces between inner body / webbing (Original in color)

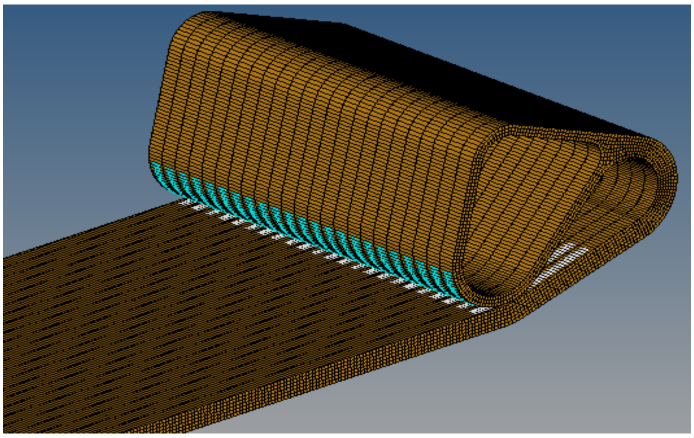


Figure 5.8: Contact surfaces between webbing / webbing (Original in color)

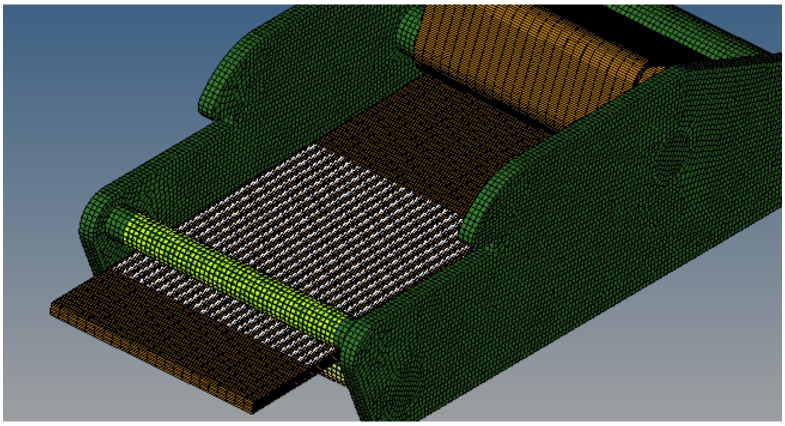
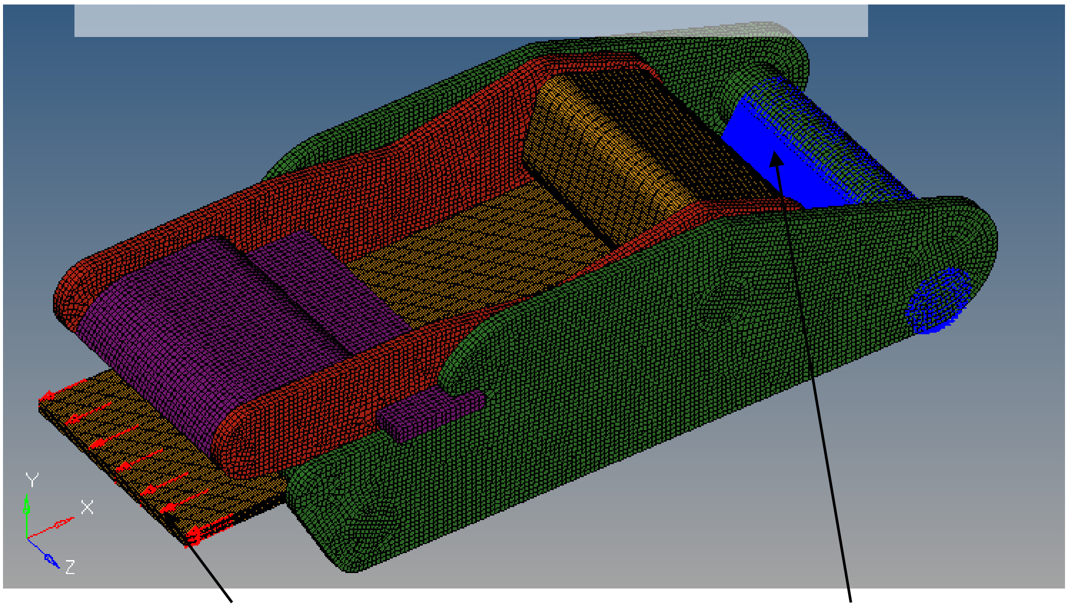


Figure 5.9: Contact surfaces between webbing / pins (Original in color)

5.4. Boundary Conditions of the Meshed Model

In the operational condition and tensile test setup in tensile test machine, two parts of webbings are used as fix end and free end as shown in Figure 1.4 & 1.5. The fixed end webbing was removed in this model to reduce complexity and 6 DOF fixed constraints were given to the contact area of the front most connecting pin where the relevant webbing part contacts. Load was applied opposite to the X direction by means of equal point loads to the webbing as shown in Figure 5.10.



Distributed Points Loads

Fixed Support

Figure 5.10: Boundary condition for the FE Model (Original in color)

5.5. Finite Element Model Validation

The FE model developed in the previous steps, was required to validate having comparison with the experimental results.

5.5.1. Deformation Check for Model Validation

Deformation check results at the SWL was to be taken into account as the first step of the model validation. Total of the distributed point loads were set to SWL which is 3333Lb (14.8 kN) and the FEM was solved. The deformation results were compared with the experimental data obtained in Section 4.1. The direction of the small deformation was considered than the negligible deformations occurred during the experimental test and the theoretical analyses for the SWL.

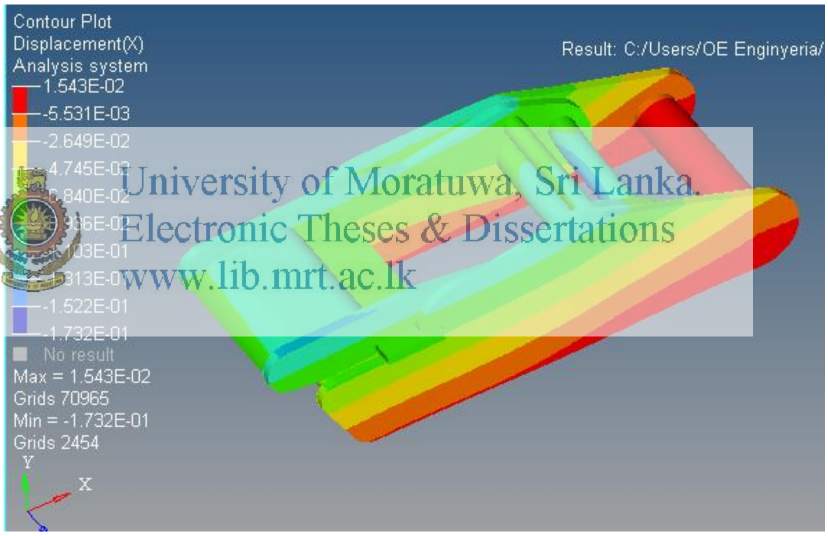


Figure 5.11: Displacement contour to the X direction (Original in color)

Deformation occurred in the critical places and the direction (refer to **Appendix C**) was noted and the comparison was tabulated in the Table 5.4. In some cases, the dimensions from several places were recorded and got the average. For others, got the minimum or the maximum compared to the place of the deformation. All the dimension taken from the meshed model is to be considered as approximate dimensions as the limitation of the taking measurements from a FE model.

The experimental values were taken as an average of the deformation check done for the samples. Observing the results, it is obvious that the theoretical values and the experimental values are in an agreement. Especially, the direction of the deformation occurred considering decrease (-) and increase (+) are corresponded.

The gap between the pins at the middle (dims with balloon marks A and D in Appendix C) is assumed to be reduced when the buckle is at loaded condition.

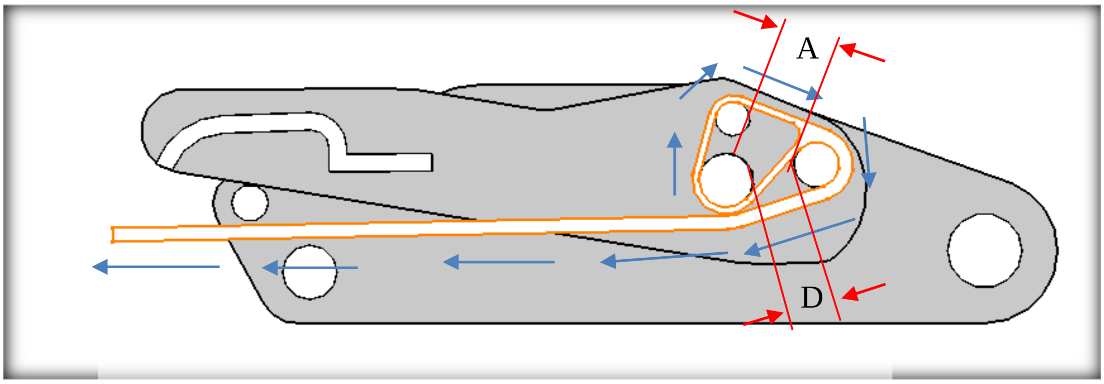


Figure 5.12: Critical load bearing pins (Original in color)

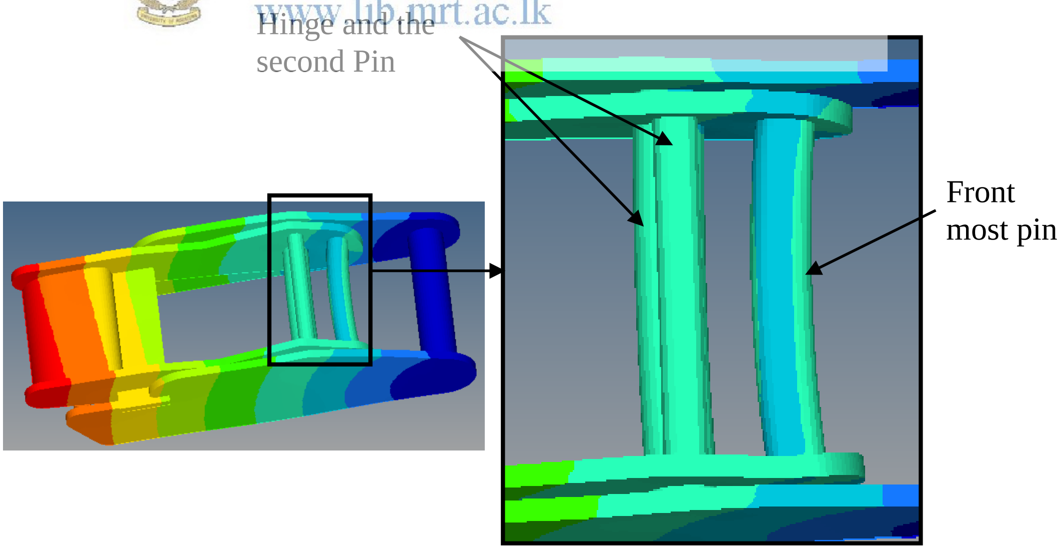


Figure 5.13: Deformation (8x) of the load bearing pins (Original in color)

Table 5.4: Deformation check (experimental vs theoretical) - model validation

Drawing Balloon Ref. (refer to Appendix C)	Drawing Dimension (mm)	Deviation from Experimental test (mm)	Deformation from FEA (mm)
A	5 (minimum)	-0.05	-0.05
B	Ø9 (minimum)	-0.01	0.03
C	Ø9.5 (minimum)	-0.03	0.00
D	5 (minimum)	-0.07	-0.05
E	Ø12 (minimum)	-0.02	0.01
F	Ø12.7 (minimum)	0.00	0.00
G	167	0.05	0.09
H	42	-0.01	-0.01
J	45.5	0.04	0.03
K	55	-0.01	-0.02

The results given from FEM was similar to the expected condition and was similar to the experimental results obtained in the Section 4.1. Hence, it was proved that the critically loaded pins act as expected and hence, the FEM was reasonable to use for analysis.

5.5.2. Buckle Failure Mode Analysis for Model Validation

The OCB failure occurred in the experimental test (refer to the Section 4.3) was compared with the static structural results given by the Finite Element Model for the same load. Load of 39.32kN, which is the load that the buckle failed in the experimental tensile test (refer to Table 4.3), was applied to the FE model and linear static analysed. Linear static analysis is an approximation for this analysis as the failure load is in the non-linear range. The analysed results were observed with the HyperView and the elements which have stresses above the yield stress of the material were removed. Most of the elements removed from the outer body sheet metal part were closer to the pin which the fixed end webbing involved. The experimental failure closely is coincided with the theoretical results obtained from the FE model.

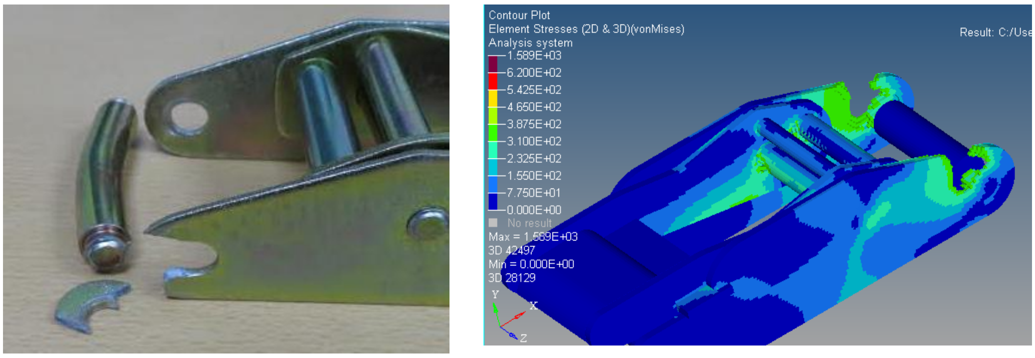


Figure 5.14: OCB failure - experimental vs theoretical (Original in color)

Considering the above listed comparisons between experimental results and FEA results, it can be concluded that the proposed Finite Element Model is giving the results more closely to the experimental values. As a result, the Finite Element Model prepared for the analysis of this thesis is justified.

5.6. Static Structural Tensile Test Results - Original OCB

The validated FE model was subjected to the SWL which is assumed to be the maximum load expectation in operational conditions of the buckle. This was taken as 2/3 of the RL for the selected 5000 lb OCB design. With the SWL, OCB could be re-used with no plastic deformation occurred. [22]

SWL	=	5000 x 2/3 = 3333 lb
No of point Loads	=	21
Force for one point	=	3333 / 21
	=	158.72 lb (706 N)

The load set as 706N separately for each 21 point loads and the FE model analysed. This is mainly to give uniform load over the cross sectional area of the webbing.

5.6.1. Von-Mises Stress Results (Original Model) - SWL

Von Mises stress is considered to be a safe checking principle for design engineers. The von-mises stress is often used in determining whether an isotropic and ductile metal will yield when subjected to a complex loading condition.

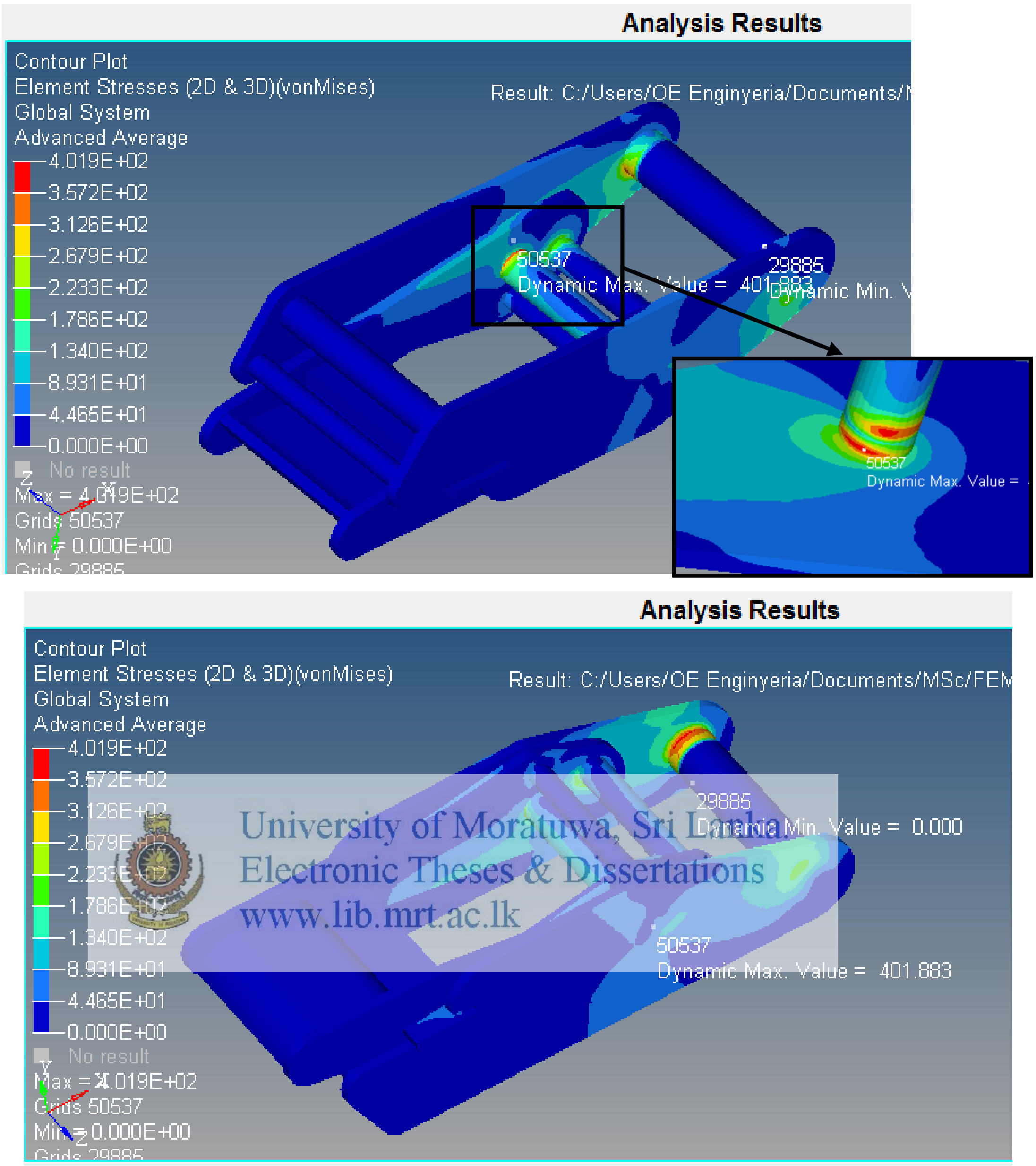


Figure 5.15: Von-mises contour for SWL - Original OCB (Original in color)

Advance average method was used for von-mises contour to view the average stress of elements. Advanced average is transformed the vector results into a consistent system and then each component is averaged separately to obtain an average tensor (or vector). The invariants are calculated from this averaged tensor. Basically stresses were accumulated to the front areas of the inner body and the outer body where load flow lines could be expected intuitionally. The maximum stress, which is 401Mpa, appears on the hinge pin between inner body sheet metal and outer body sheet metal.

High stress areas appear closer to pin / sheet metal interfaces in the front area and are still below the yield stress (415Mpa) of the material but closer. No considerable permanent deformation could be expected from the FEA results and the results correspond with the experimental data in Section 4.1. The FEA results for the SWL has been justified the FE model once more.

5.7. Topology Optimisation

The ultimate concern of this study is to find a light weight design for the 5000 lb OCB without losing the existing strengths. The optimal design configuration is obtained by checking the distribution of the load in throughout the buckle. According to the general idea of the topology optimisation, the design domain is discretized by finite elements. The material of each element is considered as a kind of porous material that is made of the solid materials, and the equivalent property of this kind of material depends on the property of the material and the porous fraction. Thus, the topology optimisation can be accomplished by designing the density of each element. The solver calculates an equivalent density for each element, where 1 is equivalent to 100% material, while 0 is equivalent to no material in the element. The solver then seeks to assign elements that have a low stress value a lower equivalent density before analyzing the effect on the remaining structure. In this way extraneous elements tend towards a density of 0, with the optimum design tending towards 1.

Finite element based topology, sizing and shape optimisation tools are typically used as part of a two-phase design process. Firstly, a topology optimisation is performed to obtain a first view on an optimal configuration for the OCB as an initial design with optimal load paths. Next, the suggested configuration is interpreted to form an engineering design and this design is then optimised using detailed topology optimisation methods with real design requirements. The ultimate target of this topology optimisation study was to minimize the weight of the buckle limiting to the 30% of its initial mass/volume. [07] [13] [14] [15]

The necessary steps were followed to setup an optimisation problem from FEM used for normal analysis and can be summarised as follows;

1. Distinguish design space / non-design space
2. Set responses
3. Set optimisation constraints
4. Set the objective
5. Analyse for optimisation

5.7.1. Design Space / Non-Design Space

The design space of the buckle, in other words part of the model which are required to optimise, is defined with help of property collectors. STEEL SOLID property collector was set as design space including all the parts of the OCB for optimisation. The reason for this was to optimise the buckle considering all the volumes of the buckle.

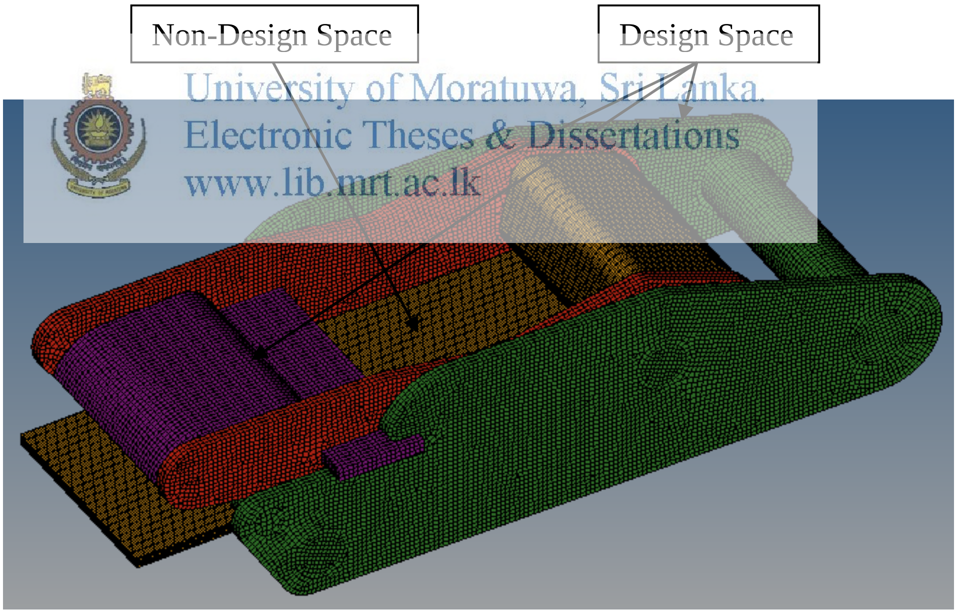


Figure 5.16: Design and non-design spaces in OCB model (Original in color)

Free end webbing part was kept in the non-design space as the webbing part is not required to optimise. If there is no separate property collectors like design space and non-design space, then all the parts will be optimised and the results then will conflict with the requirement.

5.7.2. Responses

Responses are the measurement of system performance which interested in during optimisation problem, like volume, mass, etc…

Two responses were created for the problem to support defining objective function and constraint.

- Volfrac response
- Compliance response

5.7.3. Optimisation Constraints

A restriction placed on a problem by limiting the values that selected response functions of the system can take and that must be satisfied for the design to be acceptable. Whilst they can be expressed as equalities, constraints are usually expressed as inequalities.

The weight reduction was limited to the 30% of the initial buckle weight. This was set as a volume fraction in this problem and in the response panel, the volume constraint is created as "VolFrac" and the upper bound was set to 0.3.

5.7.4. Objective Function

Objective function is the goal of the optimisation process and is the function whose least value is searched for during an optimisation. It represents the single most important property of a design, and it's associated response is a function of the design variables, e.g. Mass, Stress, Displacement, Moment of Inertia, Frequency, Center of Gravity, Buckling factor, etc. [15]

The objective of the OCB optimisation was set as to minimize the compliance. Minimizing compliance means improving stiffness.

5.7.5. Visualization of Results

Optimisation process was run after setting all the steps mentioned above. The buckle optimisation ended after 12 iterations with the promising messages listed below.

- Optimisation has converged
- Feasible design (all constraints satisfied)

Optimisation variation against iteration was viewed. It clearly shows that the optimisation has become stable after several iterations.

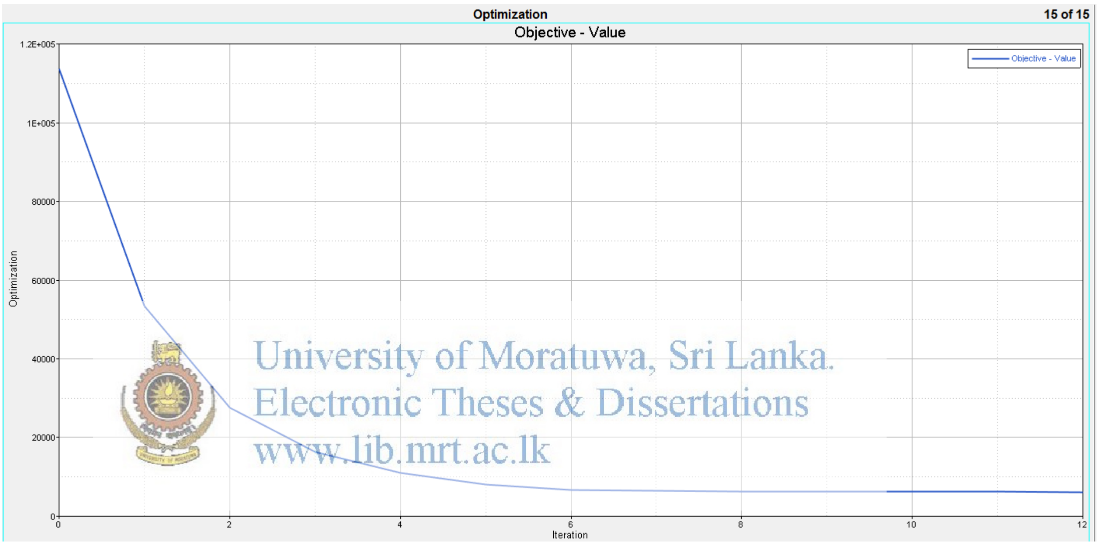


Figure 5.17: Compliance Vs Iteration number graph

The topology optimisation results were available with relevant *.h3d file to open with HyperView for visualization of results. When the file is opened in HyperView, the element density distribution could be viewed in colours varied from blue to red.

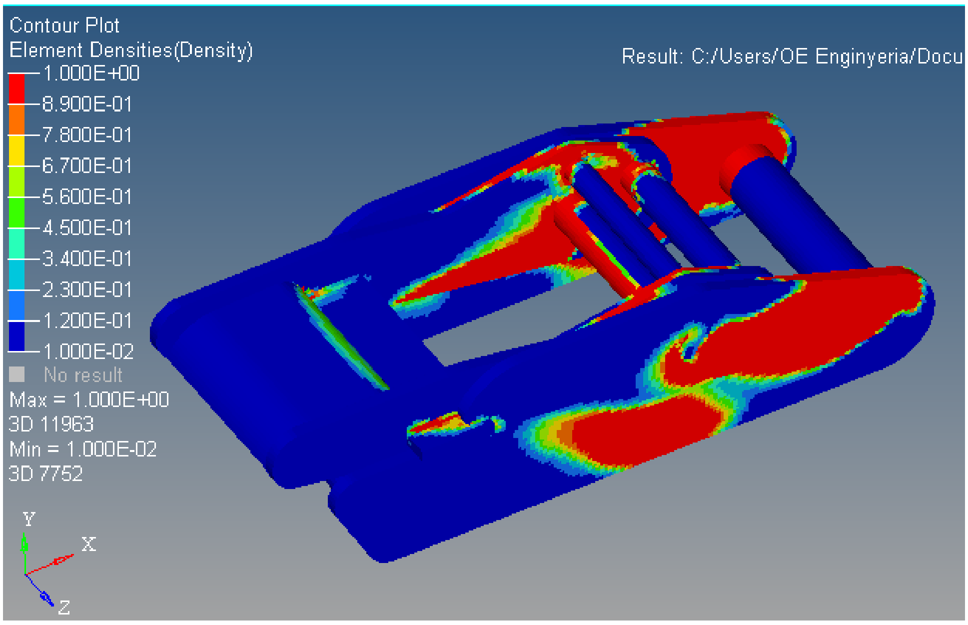


Figure 5.18: Load density value variation (Original in color)

Red color indicates the regions with element density one or more closer. Those areas could be identified as the regions which are structurally critical for the design. Blue color indicates the regions with element density zero or more closer. Those areas could be identified as non-critical areas for the design. According to the rest of values, colors are appeared as in the legend. The webbing was not kept in the design space, hence it was excluded from the results.

5.8. Proposed Model for 5000 lb OCB

According to the responses and objective were set, the elements with a density value below 0.1 were neglected, which implies that their respective contribution to the overall stiffness of the structure were also neglected. This is counterbalanced by assigning all elements with densities greater 0.1 the new standardized density value one. This in turn implies that their stiffness is equally increased. The geometry areas, where the density value falls above the element density 0.1 were derived in *.IGES format as geometries from the OSSMOOTH panel. OSSmooth is semi-automated design interpretation software, facilitating the recovery of a modified geometry resulting from a structural optimisation, for further use in the design process and FEA re-analysis. The recovered geometry was opened with the original model to get a clear idea about the areas.

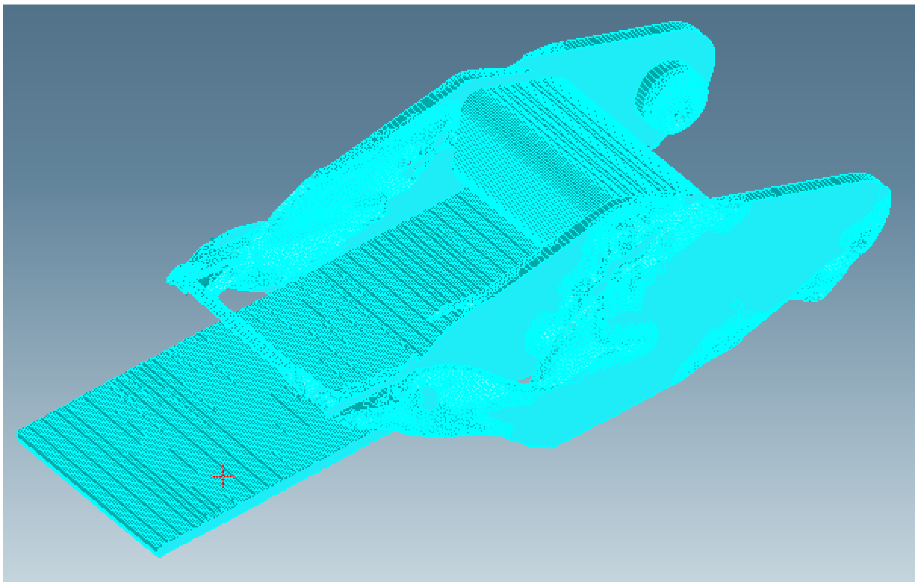


Figure 5.19: Recovered optimised geometry (Original in color)

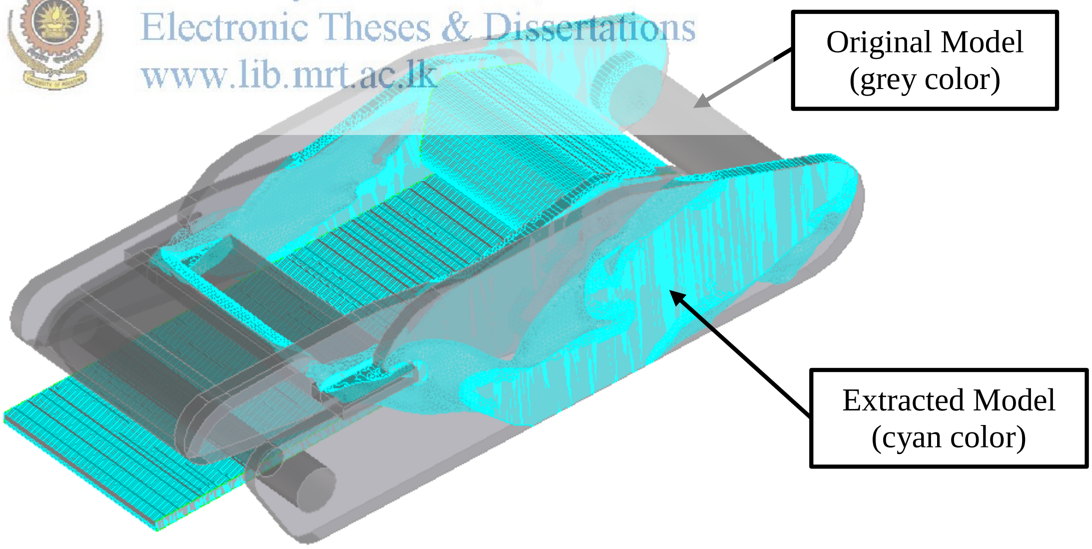
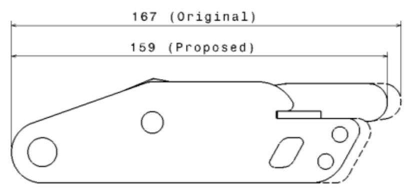
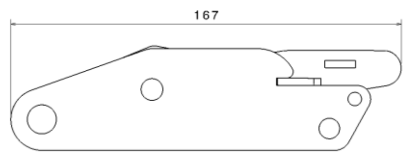
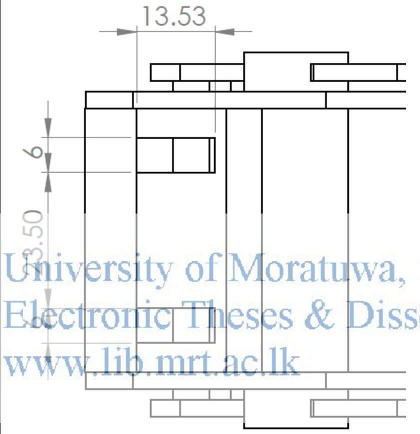
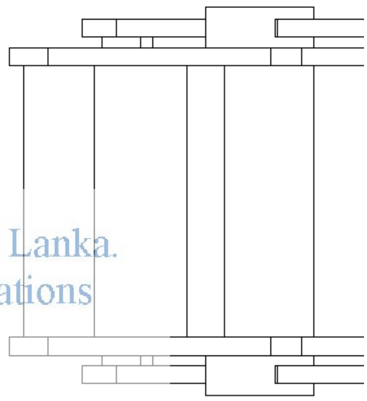
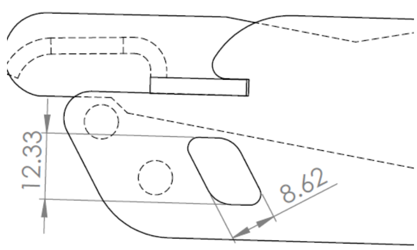
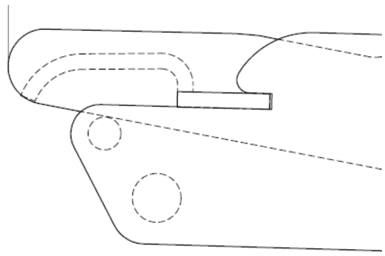
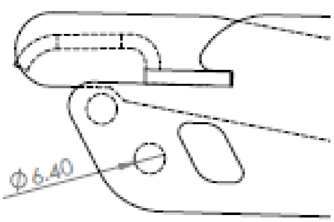
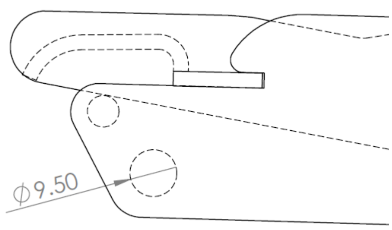


Figure 5.20: Optimised geometry vs original geometry (Original in color)

After having comparison carefully between both geometries and opening both in same CAD window, the structurally non-critical areas were identified. However, the obtained results are related to the structural requirements. Hence, considering the other physical requirements, step by step modification was done for the existing OCB buckle design. The changes proposed to the optimised model were tabulated in Table 5.5 having comparison to the original OCB design.

Table 5.5: New proposed geometry vs the original model

No	Changes done	Proposed Model	Original Model
1	Length of the sheet metal parts were reduced		
2	2 slots of 6 x 13.5 mm removed from supportive sheet metal part		
3	8 x 12mm parallelogram area removed from both outer body sheets		
4	Diameter of one connecting pin reduced from 9.5mm to 6.4mm.		

1. Length of the sheet metal parts of the inner body as well as outer body were mainly reduced in this step. Reduction was carried out from the plunger side of the buckle as those areas were fallen below to the threshold, hence elements from that area could be neglected.

2. Involvement of the supportive part was minimal according to the load distribution results. No changes were done to the curved area of the supportive part as the palm is to be placed in the curved area when pulling out the plunger for buckle unlocking. Likewise, considering other requirements, two slots as shown in Figure 5.21 were removed at sides.

3. Almost all the elements in the sheet metal parts closer to the plunger or back side area are appeared in blue color and below the threshold. Therefore, considering other requirements of the area, a possibility of removing slots was considered. A parallelogram shape slot was removed from the outer body sheet metal parts as shown in Figure 5.21 after having several analyses.

4. The two connecting pins at the back side had not been considerably involved to the load distribution according to the optimisation results. The main purpose of these pins is to guide the free end webbing and hence, the diameter size of the bigger pin from those two was reduced from Ø 9.5mm to Ø 6.4mm which is the smaller pin size. This diameter size was taken having several FE analyses and similar to the lowest pin size in the model. Refer Appendix D for more clear drawing with changes.

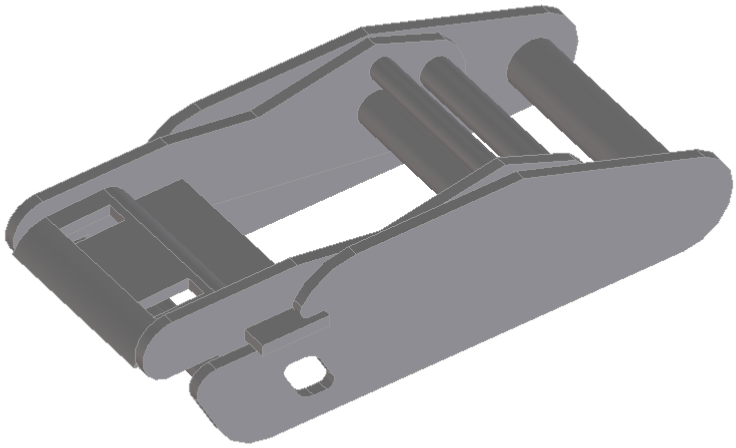


Figure 5.21: Proposed model for the 5000 lb OCB

5.9. Static Structural Tensile Test Results – Optimised OCB

The proposed model was imported to HyperMesh software and a new FE model was setup similar to the FE model developed for the original OCB design. All the conditions including merging elements, geometry simplification, creating contact surfaces and boundary conditions were kept as it is in the original model. Mesh sizes and types were also kept same and the element quality was also in the acceptable ranges. Then, the new FEM was used for static structural analysis..

5.9.1. Von-Mises Stress Results (Optimised Model) – SWL

The new model was subjected to the SWL and checked for von-mises stresses. Again, the advance average method was used for von-mises contour to view the average stress of elements in the optimised model results. Basically stresses were accumulated to the front areas of the inner body and the outer body similar to the original design FE analysis results in Section 5.6.1.

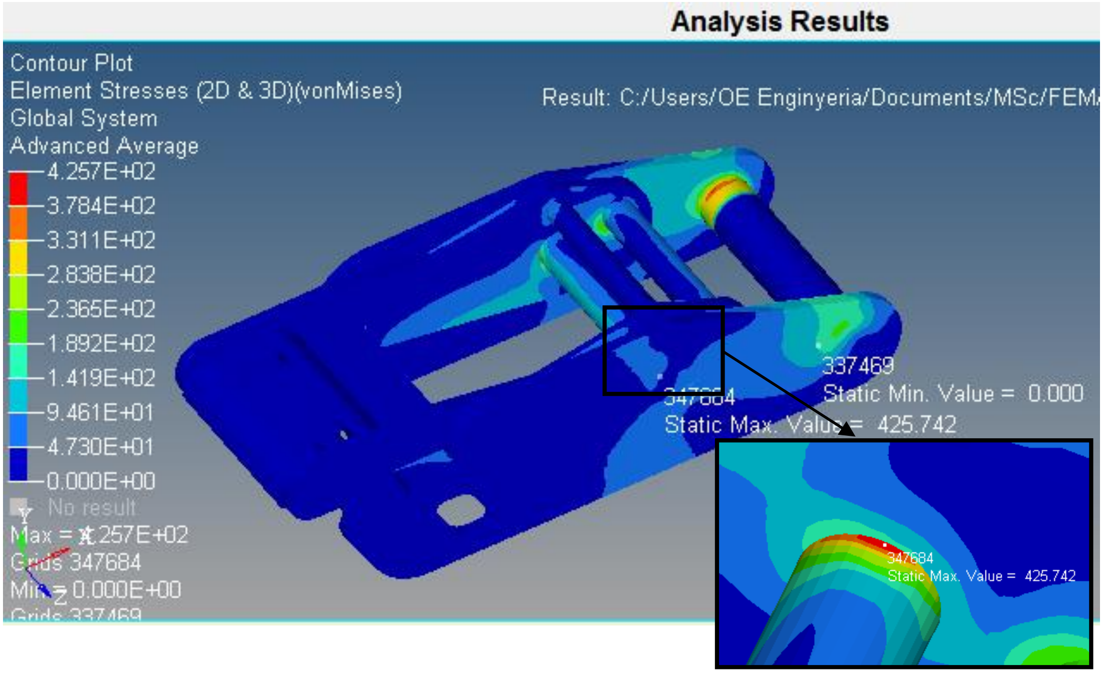


Figure 5.22: Von-mises contour for SWL - Optimised OCB (Original in color)

The maximum stress, which is 425.7Mpa, appears on the hinge pin inside of the inner body closer to the sheet metal. High stress areas appear closer to pin / sheet metal interfaces in the front area similar to the original design results and are little bit above the yield stress (415Mpa) of the material but closer. No considerable permanent deformation could be expected from the FEA results of the optimised model as well due to negligible amount of elements are above the yield limit.

6. CHAPTER 06: DISCUSSION, CONCLUSIONS AND FUTURE WORK

6.1. Discussion

The literature review demonstrated that there is a widespread range of studies relevant to optimisation techniques. Selection of a suitable method depends on the complexity of the design, the boundary conditions of the model and the availability of the technology. Optimisation has become famous in several fields where special attention is required. The word "Optimisation" for the aerospace industry came with the necessity of economical aircraft designs. Although aircraft manufactures have recently entered to this area with new series of economical aircrafts, most of bottom layer suppliers' designs are still seemed to be bulky, and need to be optimised. 5000 lb OCB is also one of identified bulky design being used in aerospace industry.

Selection of a metal fitting to the study was challengeable as lack of design specifications and other details due to the confidentiality of information pertaining to the field. 5000 lb OCB was selected from available metal fittings which are normally used in the other industries. But the scope was limited to the aerospace industry. It was identified that the achieving of a light-weight design, is a research and development process combining the theoretical studies and experimental results. As a start, a Finite Element Methodology could be developed for tensile load application on the OCB and a light-weight design was introduced without changing the functionality of the OCB letting to continue the optimisation process further.

FE model was firstly setup and analyzed with first order elements to make it easier. Once the setting up processes is completed, the elements were converted to the second order as the higher order elements always give better results. Having the analyses for the both first and second order elements, a comparison of results between first order elements and the second order elements in a same FE model was conducted. Second order elements are those with one mid-side node compared to first order elements. These types of elements offer the benefits of ease of modeling and a higher degree of accuracy per element. Hence, it can be proposed to use the second order elements to have more accurate results.

In the validation process of FE model, the idea was to compare conditions between experimental tests done using tensile test machines and theoretical tests done using FE analyses. Graphs were obtained from the test machines for each test to compare with the graphs from FE analysis. With the selection of static analysis for the FE Analysis, this idea couldn't be achieved. As there is no variation of the force with the time in the static analyses, no graph generates with time variation in FEA results. Hence, the FE model validation from this way was failed. However, better validation could be achieved with comparison of experimental and theoretical results.

From the proposed first optimised model, approximately 7% of weight reduction could be achieved. Numerically it is 41 grams. At least four safety straps are used to support a pallet and 180 pallet positions are included in the A380 aircraft design (refer to Appendix F). If A380 is taken as example and assuming all the straps are used with over center buckles, then at least 720 OCBs are needed for the full working condition. Then the weight saving is 29.5kg per aircraft. As 1kg of weight saving will save 183.6$ per year, then the cost saving is more than 5000$ per year. As this amount is only for one metal fitting of the straps and nets, and for least requirements of the safety straps for a pallet, this could be the minimum saving. As this is one of complex parts in the field and the optimisation procedure has been successfully introduced, this would be a beginning step for optimisation for this kind of metal fittings in the aerospace sector. If all metal fittings which are used in the aircraft could be optimised, that would be a considerable weight reduction compared to the total weight of an aircraft.

6.2. Conclusions

This paper attempted to formulate a mechanism for a strength based weight reduction on standard 5000 lb Over-Center Buckle (OCB) which is used in aerospace industry. Currently, weight reduction on these kind of metal fittings, is being achieved by introducing lightweight materials, following trial and error methods and by intuition while the structural optimisation is one of the most extensively researched areas in design field.

Experimental tests were successfully used to prove that the 5000 lb OCB is an over-design model and for the FEM validation. This made reasonable on formulating optimization mechanism for the 5000 lb OCB. The available technology for optimization was successfully used for an assembly in research environment with short academic time scales with the several limitations. Performing topology optimization for an assembly was successfully achieved in the study. The study showed that the load flow lines are concentrated in a specific area of the buckle and hence, a significant weight reduction from rest of the area is possible without touching the functionality of the OCB. As a quantitative analysis, the weight reductions of the proposed OCB design was closer to 7% (41g) and an improved design can be achieved through further R and D process with theoretical and experimental tests. The study opens a path to consider values of optimization in all fittings in the aerospace safety industry to have a considerable total weight reduction of an aircraft.

6.3. Future Work

The study was limited to static structural analysis when determining the proposed design as there was a time limitation and a lack of information regarding fatigue behavior of the OCB. As the next step of this continuous process, fatigue and bending moments can be taken into account. Static analysis gives better results as long as parts behave in a quasi-static manner. However, non-linear analysis considering multi-body dynamics could be considered to arrive at a more accurate result.

Optimisation process can be continued as a research and development process to ultimately arrive at a better optimised model. Consequently, the proposed design can be manufactured from the original 5000 lb OCB supplier and subjected to the same qualification check with same conditions mentioned in the SAE (Society of Automotive Engineers) aerospace standards. If the OCB load characteristics are still far than expected, optimization process can be continued further. The process could be continued till the optimal model is achieved.

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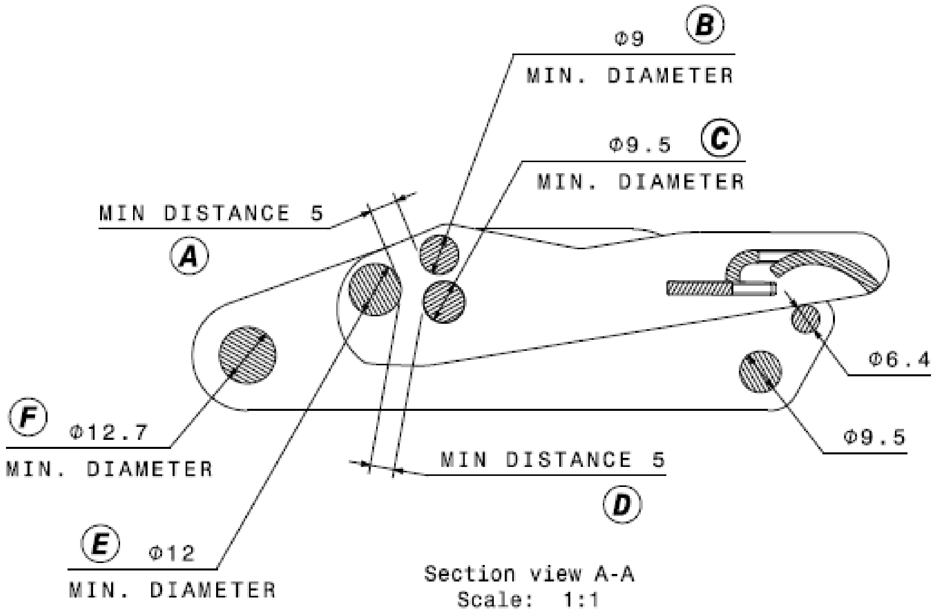
Appendices

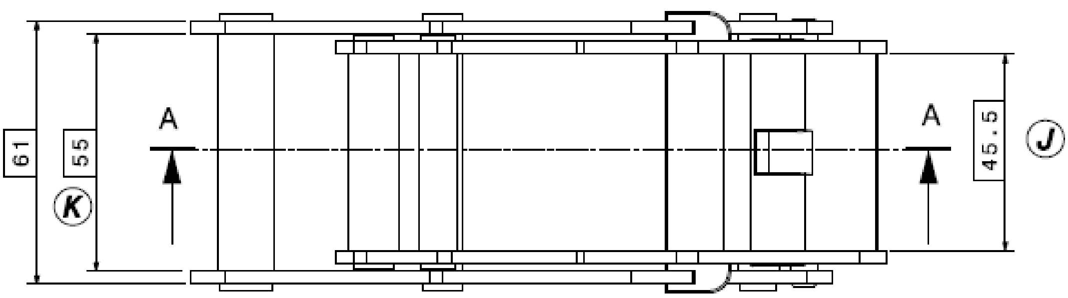
Appendix A: FEA Packages available for Analysis purposes

- Altair HyperStudy, OptiStruct, solidThinking Inspire
- ANSYS Adjoint Solver, Optimetrics
- Autodesk Optimisation for Inventor
- CD-adapco STAR-CCM+ /Enabling Optimate+
- Cenaero Minamo
- Collier Research HyperSizer
- COMSOL Multiphysics Optimisation
- Concepts NREC TurboOPT II
- DATADVANCE MACROS, pSeven
- DecisionVis ExplorerDV
- Dynamic Design Solutions FEMtools Optimisation
- Dynardo optiSLang
- ESI Group Virtual Performance Solution Optimisation
- ESTECO modeFRONTIER
- Exa PowerFLOW Optimisation Solution
- FEA-Opt SmartDO
- FRIENDSHIP SYSTEMS CAESES/FRIENDSHIP-Framework
- Function Bay Recur Dyn /Auto Design
- iChrome Nexus
- LIONlab LIONsolver
- LSTC LS-OPT
- MSC Nastran Design Optimisation
- NISA Software CSIL NISAOPT
- Noesis Solutions Optimus
- Optimal Solutions Sculptor
- Phoenix Integration ModelCenter
- PIDOTECH PIAnO
- PTC Creo BMX (Behavioral Modeling Extension)
- Quint OPTISHAPE-TS
- RBF Morph
- Red Cedar Technology (a CD-adapco company) HEEDS MDO, HEEDS NP
- Siemens PLM NX Nastran Optimisation, Femap with NX Nastran Optimisation, LMS Virtual.Lab Optimisation
- Sigma Tech IOSO
- SIMULIA Isight, SEE, Tosca
- SolidWorks Simulation Structural Optimisation
- Vanderplaats R&D GENESIS, DOT, BIGDOT, VisualDOC
- Virtualpyxis Virtual.PYXIS
- Within Technologies (an Autodesk company) Enhance

Appendix B: 5000 lb Over-center Buckle Engineering Drawings

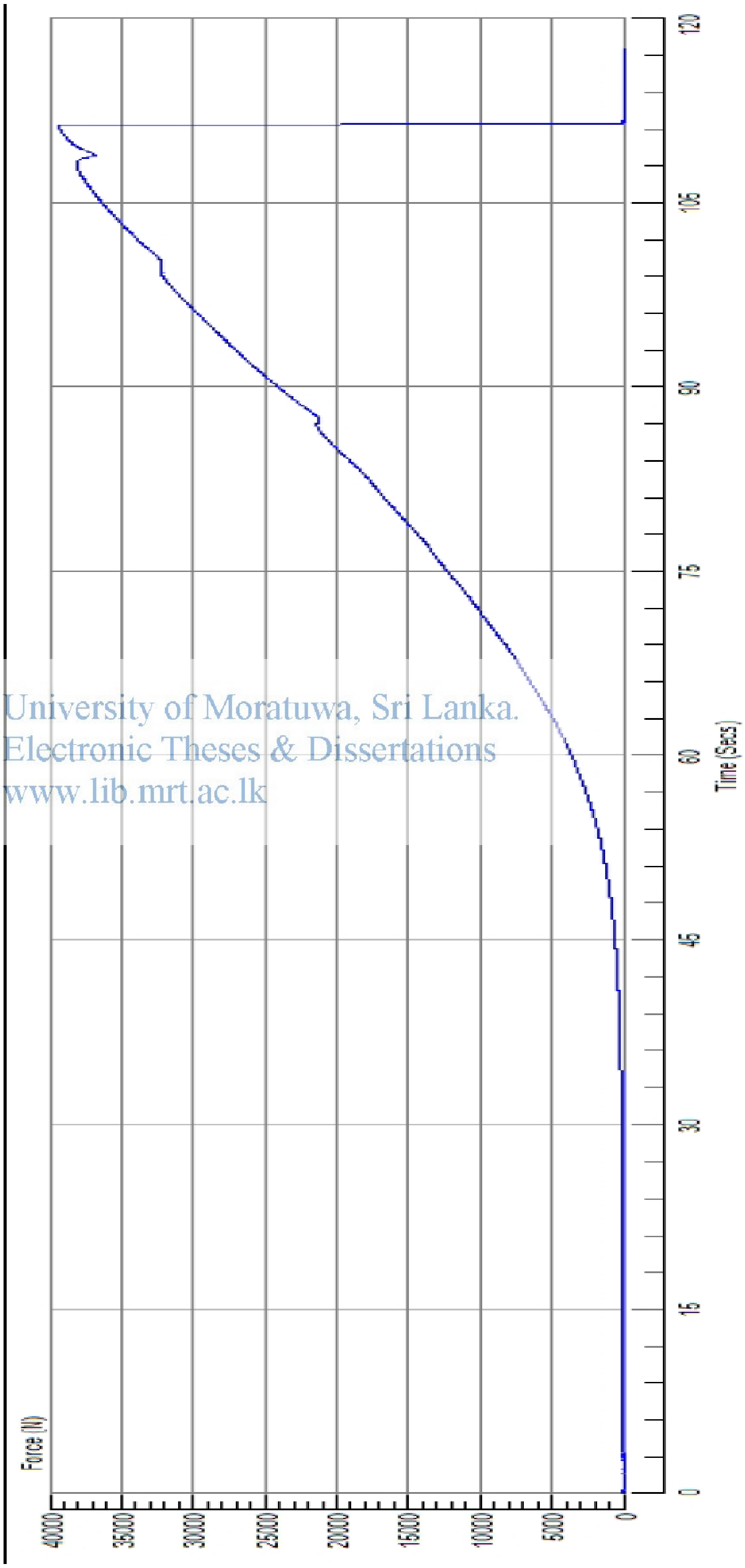
Appendix C: Balloon Marked Dimensions for Deformation Check



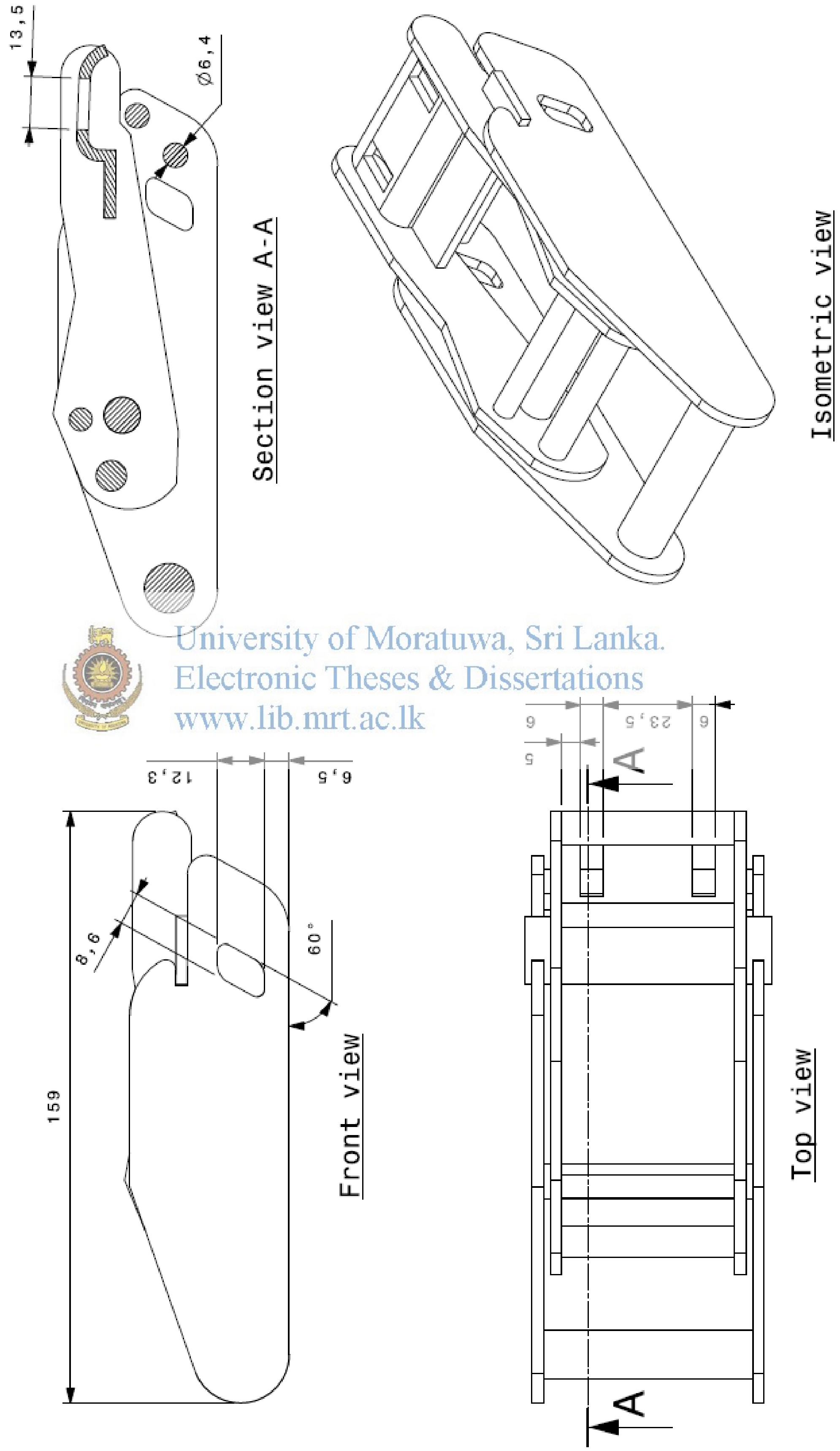


Appendix D: Time vs Force Graph – Test for OCB Failure

Test No	Force @ Peak (N)
1	39320.000



Appendix E: Changes in the Optimised OCB Design



Appendix F: Estimated Cost Savings for Airlines

Sorce: [23]

Estimated cost savings for ABC Airlines

	Aircraft type	Quantity	Pallet positions	Total pallet positions	Weight savings (kg)	Fuel savings ($k)
Passenger	A330	22	6	132	1,320	242,352
	A340	15	6	90	900	165,240
	A380	18	10	180	1,800	330,480
	777-200	22	6	132	1,320	242,352
	777-300	55	8	440	4,400	807,840
Freighter	747F	8	36	288	2,880	528,768
	777F	4	37	148	1,480	271,728
Total		**144**	**109**	**1,410**	**14,100**	**2,588,760**

Assuming a 10kg per net weight saving
Fuel cost $1,025/tonne and a 1kg weight saving = $183.60 per year
Estimated 4,500 flying hours per year (average 12.3 hours per day)